EXPERIMENTAL AND FINITE ELEMENT ANALYSIS OF CONTACT BEHAVIOR IN LUBRICATED ROLLING CONTACT

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DECLARATION

This work has not previously been accepted in substance for any degree and is not being concurrently submitted in candidature for any degree.

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STATEMENT 1

This thesis is the results of my own investigations, except where otherwise stated. Other sources are acknowledged by giving explicit references. Bibliography/references are appended.

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LIST OF ABBREVIATIONS

SYMBOL	DESCRIPTION
2D	Two-dimensional
3D	Three-dimensional
AISI	American Iron and Steel Institute
ANSYS	American computer-aided engineering software
ASTM	American Society for Testing and Materials
CNT	Carbon nanotube
cof	Coefficient of friction
EP	Extreme pressure
FE	Finite element
FEA	Finite element analysis
FEM	Finite element method
RBD	Refined bleached and deodorized
SKF	Svenska Kullagerfabriken (Swedish Ball Bearing Factory)
TEHL	Thermal elastohydrodynamic lubrication

ABSTRAK

Pelincir seperti minyak dan gris digunakan secara meluas dalam galas unsur bergolek untuk mengurangkan geseran antara dua permukaan logam. Pelincir membentuk lapisan nipis antara permukaan logam untuk menahan sesetengah beban yang dikenakan oleh galas. Pelinciran yang tidak betul boleh menyebabkan tekanan kontak tinggi dan ubah bentuk pada galas dan akhirnya akan mengakibatkan kegagalan. Tujuan projek ini adalah untuk mengkaji pekali geseran yang dihasilkan oleh gris berasaskan minyak sawit yang baru dibangunkan dan untuk menyiasat ciri-ciri kontak pada galas roller yang dilincirkan dengan analisis elemen hingga. Hasilnya kemudian dibandingkan dengan dua gris komersil yang tersedia untuk menilai prestasinya. Terdapat empat jenis gris yang diuji secara eksperimen, iaitu gris berasaskan minyak sawit formulasi A dan B, gris gred makanan dan gris galian. Pekali geseran masing-masing ditentukan menggunakan penguji empat bola berdasarkan prosedur ujian standard ASTM D2266. Analisis elemen hingga model galas separa roller 2D dan 3D dilakukan dalam keadaan statik dengan menggunakan perisian simulasi ANSYS. Model elemen hingga ini dibuat berdasarkan geometri galas roller dan simulasi dijalankan dalam keadaan tanpa geseran dan dilincirkan. Hasil kajian menunjukkan bahawa gris minyak sawit formulasi A mempunyai pekali geseran paling rendah, diikuti dengan gris minyak sawit formulasi B, gris galian dan gris gred makanan. Ini menunjukkan bahawa gris berasaskan minyak sawit berpotensi untuk digunakan dalam aplikasi sentuhan bergolek disebabkan ciri geseran yang rendah. Analisis elemen hingga dengan pelbagai pekali geseran pelincir menunjukkan bahawa tekanan von Mises maksimum dan ubah bentuk untuk kontak geseran adalah lebih tinggi daripada kontak tanpa geseran. Untuk analisis kontak geseran 2D, nilai serupa diperoleh dengan tekanan von Mises pada 400.69 MPa dan ubah bentuk pada 3.4033×10^{-4} mm. Sementara itu, untuk model galas separa roller 3D, hasilnya menunjukkan terdapat sedikit penurunan pada tekanan setara dan ubah bentuk antara roller dan gelang dalaman dengan peningkatan pekali geseran gris. Hasilnya dibandingkan dengan kaedah analitik berdasarkan teori kontak Hertzian, di mana nilai tekanan setara dan ubah bentuk untuk kontak tanpa geseran menggunakan kaedah 2D adalah lebih dekat dengan nilai teori daripada yang menggunakan kaedah 3D. Hasil kajian menunjukkan bahawa perbezaan kecil pekali geseran gris tidak mempengaruhi sentuhan bergolek.

Hasil kajian juga menunjukkan bahawa gris terbiodegradasi yang baru dibangunkan mempunyai prestasi yang serupa dari segi geseran sentuhan bergulir dan ciri-ciri kontak dalam keadaan bahawa galas beroperasi dalam keadaan biasa.

ABSTRACT

Lubricants such as oil and grease are widely used in rolling element bearings to reduce the friction between two metal surfaces. Lubricant forms a thin film between the surfaces to partially sustain the load exerted by the bearings. Improper lubrication may cause high contact stress and deformation to the bearings and will eventually lead to catastrophic failure. The purpose of this project is to investigate the coefficient of friction produced by newly developed palm oil-based grease and to investigate the contact characteristics in lubricated roller bearings by finite element analysis. The result is then compared with two available commercial greases to evaluate the performance. There were four types of grease that were tested experimentally, which were palm oil-based grease formulations A and B, food grade grease and mineral grease. The respective friction coefficients were determined using four-ball tester following the standard testing procedure ASTM D2266. The finite element analysis of 2D and 3D partial roller bearing models were conducted in static condition using the ANSYS simulation software. The finite element model is developed based on the roller bearing geometry and the simulation is carried in a frictionless and lubricated condition. The finding shows that palm oil grease formulation A had the least coefficient of friction, followed by palm oil grease formulation B, mineral grease and food grade grease. This indicates that the palm oil-based grease has the potential to be applied in rolling contact applications due to low friction characteristics. Finite element analysis with various lubricant coefficients of friction shows that the maximum von Mises stress and total deformation for frictional contact are higher than the frictionless contact. For the 2D friction contact analysis, similar values were obtained with von Mises stress at 400.69 MPa and deformation at 3.4033×10^{-4} mm. Meanwhile, for 3D partial roller bearing model, the results showed there was a slightly decreasing trend on the equivalent stress and total deformation between the roller and the inner ring with the increase in the coefficient of friction of the grease. The results were compared with the analytical method based on Hertzian Contact Theory, where the values of equivalent stress and total deformation for frictionless contact using 2D method were closer to the theoretical values than that using 3D method. The finding shows that the small difference in grease coefficient of friction did not affect the rolling contact. The finding also shows that the newly developed biodegradable grease has a similar performance in terms of rolling

contact friction and contact characteristic in a condition that the bearing is operating in normal condition.

CHAPTER 1
INTRODUCTION

1.1 OVERVIEW

Rolling element bearings is a mechanical device that provides rotational movement to carry the load to another device connecting to it. It consists of several rolling elements in the form of balls or rollers, which are placed in between the two bearing rings known as races. The relative motion between the inner and outer races causes the rolling element to roll with very little rolling resistance. Rolling element bearings generally reduces the friction between moving parts and it is widely used in various applications, especially in industrial machineries.

To minimize friction and prevent excessive stress and deformation, lubrication usually applies in metal-metal contact applications such as roller bearings. Different types of lubricants such as oil, grease and so on are designed for specific operating conditions. Generally, lubricants are used in rolling element bearings due to its unique characteristics such as maintaining a stable viscosity over a wide range of temperatures, providing good film strength which able to support loads and provides a barrier against moisture and contaminant.

Recently, biodegradable greases are started to be considered as lubricants for industrial and transportation applications, due to growing concern over environmental sensitivity. This research emphasizes on substitution of mineral-based oil grease with vegetable-based oil grease and palm-based oil grease for more environmentally friendly lubricants for roller bearing application. As an example, palm-oil has been tested widely in the lubrication industry and potentially to be used as a lubricant due to the advantages of palm-oil over conventional lubricants, which are inexpensive, non-toxic and easily decomposed. In the current world state, pollutions are all over the place and become a reason for a demand of biodegradable grease in lubricants, as well as health and safety issues emerging due to changes in economic and supply factors [1].

1.2 PROJECT BACKGROUND

The cycle of rolling contact consists of three phases, which are the running-in phase, the steady state phase and wear-out phase. Running-in, also known as breaking-in, is the change in the geometry of the sliding surfaces and in the mechanical properties of the surface layers of the material during the initial sliding period [2]. Generally, it is a process of moving parts wearing against each other to produce small adjustment that will settle them into a steady state for the rest of the working life. Although wear is usually undesirable, running-in wear is encouraged rather than avoided [3]. This process can be found in many applications, including the breaking-in of a brand-new engine.

During the running-in phase, when two new metal surfaces are loaded for the very first time, there may be slight initial misalignment caused by the difference in roughness between the two metal surfaces. This creates ‘high spots’ on almost all surfaces, which contributes to higher surface roughness. In addition, the high rotational speed of the roller bearings produces high heat caused by the frictional forces acting between the rollers and the races. Without lubrication to the bearings, the contact areas between the rollers and the races will experience significantly high stress and deformation, which then causes the bearings to fail. SKF stated that improper lubrication accounts about 80% of bearing failures [4].

To solve this issue, research and development efforts have been carried out in order to determine the appropriate type of lubrication to the bearings. The main function of lubrication in rolling element contact is to form a layer of other material between the two metal surfaces, which reduce the frictional force between them and prevents wear. Different types of lubricants with different properties such as viscosity and friction coefficient are developed for different applications based on the operating load and speed. Proper lubrication to the bearings is crucial to ensure the durable and reliable operation of the bearings. This also helps to reduce initial costs and prevent costs associated with the premature bearing failures. In recent years, biodegradable lubricant started to be developed due to the rising concern of the pollution produced by conventional mineral grease.

Researches have been done to investigate the performance of the biodegradable grease include the viscosity and friction of newly developed vegetable oil-based lubricant. Several studies have been conducted to investigate the contact behavior in rolling contact, however the studies are limited to dry contact. The contact stress and deformation in lubricated roller contact is rarely reported. Most of the researchers had proposed several types of finite element model of rolling contact were developed to analyze the stress and deformation acting between the rollers and the races.

In this project, biodegradable grease has been produced from palm oil as the base oil. The friction coefficient of the new developed grease is measured and compared with commercially available grease. The contact characteristics of the roller bearings include the friction and contact stress produced by these greases are studied. Experimental work is carried out to investigate the friction coefficient of the grease. The value is then applied in finite element analysis to evaluate the contact condition in terms of contact stress and deformation. The finite element model is developed considering rolling contact in roller bearing by using simulation software.

1.3 PROBLEM STATEMENT

During the rolling of bearings, metal-to-metal contact especially in rough surfaces contributes friction and heat. Over time, the metal contact area could be under high stress and the metal surfaces begin to wear and deform. Eventually, the bearings can fail due to severe stress and deformation. To maintain the lifespan of the bearings, several types of lubricants can be applied to the bearings in order to minimize friction and prevent wear. Hence, the stresses and deformation acting on metal contact area could be managed with the aid of lubrication.

1.4 PROJECT OBJECTIVES

The objectives of this project are focused in this study to address the issues stated in the problem statement, which are

1. To investigate the coefficient of friction produced by the newly developed grease.
2. To investigate the friction performance of different types of grease experimentally.
3. To investigate the contact characteristics of lubricated roller bearings by finite element analysis.

1.5 SCOPE OF WORK

The scope of this work is divided into two. First is the experimental work to produce biodegradable grease and measure its coefficient of friction. There are four types of grease lubricant is tested, which are palm oil grease formulation A, palm oil grease formulation B, food grade grease and mineral grease. The friction is measured using four ball test. The second is the finite element analysis of the contact behavior in lubricated contact. The finite element model is developed for roller bearing using ANSYS software. The data obtained from the experiment is applied in the simulation. From the simulation, the stress and deformation are obtained and analyzed to observe the contact characteristics of lubricated roller bearings.

1.6 ORGANIZATION OF THESIS

This thesis is divided into five main chapters, which are introduction, literature review, methodology, results and discussion, as well as the conclusion. Chapter 1 (Introduction) briefly explains the background of this work includes the importance of lubrication in rolling contact, specifically in roller bearings. Chapter 2 is the literature review, which summarizes various approaches and studies that have been conducted by other researchers related to the contact analysis of bearing using finite element method. Chapter 3 explains the methodology that is applied in this research project, which can be divided into two parts. The first part is the experimental setup for grease synthesis and tribology test using Four-ball tester, while the second part is the simulation of the roller bearing model in both 2D and 3D methods using ANSYS. Chapter 4 discusses the results from the experimental part mentioned above along with the results from the simulation model such as stress and deformation. Lastly, Chapter 5 concludes the research's outcomes that fulfil the objectives of the project as well as the future implementation.

CHAPTER 2
LITERATURE REVIEW

2.1 LUBRICATION AND FRICTION COEFFICIENT

Various studies have been carried out on the effects of different lubricants on the friction characteristics on metal surfaces. Matthias [5] reviewed on the ultra-low wear systems during the running-in of lubricated metal-metal contacts. The researcher found that the initial coefficient of friction is the highest and steadily decreases with time during the running-in.

Sutaria et al. [6] carried out ASTM performance tests on the four-ball tribotester using five different blending ratios of oil samples. They found out that as the blending ratio of oil with based oil increases, the viscosity of oil decreases, which resulting in decreasing coefficient of friction and increasing wear scar diameter.

Yong et al. [7] conducted an experiment using four-ball tester based on ASTM D4132 B standard to obtain the friction and wear characteristics of natural oil-based lubricants with glycerin and oleic methyl ester with carbon nanotubes (CNT) as additive. They found that lubricant with CNT as additive had lower friction coefficient compared to that with refined glycerin.

Syahrullail et al. [8] performed an experiment to evaluate the effect of load on the tribological performance of refined bleached and deodorized (RBD) palm olein and paraffinic mineral oil using four-ball tester according to the standard ASTM D4172. They concluded that RBD palm olein had better friction reduction and wear resistance than paraffinic mineral oil.

Another similar study was also performed by Chiong Ing et al. [9] who investigated the influence of the normal load on friction and wear performance for RBD palm olein and compared with paraffinic mineral oil at various normal load using four-ball tester in accordance with the standard testing procedure ASTM D4172. By using the friction torque data, the coefficient of friction could be calculated according to IP-239. They also concluded that RBD palm olein had lower coefficient of friction but had larger wear scar compared to paraffinic mineral oil at various normal loads.

Joysula et al. [10] studied the extreme pressure (EP) behavior of grease using four-ball tester according to the standard ASTM D2596. It was done to understand the seizure prevention caused by EP lubricants, which is affected by several factors such as actual load, sliding speed, lubricant temperature and friction. In addition, the fictitious low friction lubricant could decrease in surface roughness at higher extreme pressure or high load.

2.2 NUMERICAL SIMULATION OF CONTACT ANALYSIS

Several studies have also been carried out on the contact analysis of metal-to-metal surfaces, especially the contact characteristics of rolling element bearings using finite element analysis (FEA) to determine the Hertz contact stress and other properties. Liu et al. [11] studied on the contact characteristics of a roller bearing with horizontal and slant subsurface cracks. From their simulated results, they concluded that the contact deformation, contact width and the stress between the roller and the race increase with the radial load on the horizontal subsurface crack. Similarly, the contact stress increases with crack depth and crack width.

Liu et al. [12] built a finite element model to investigate the plastic deformation at the spall edge of a roller bearing with various radial load and rotor speed. The results show that the maximum contact stress, the total deformation and the plastic deformation have increasing trends with the increase of radial load and rotor speed. Also, the simulation also shows that the longer the spall length, the higher the maximum contact stress as well as the total deformation and plastic deformation.

Yin et al. [13] analyzed the contact deformation and stress distribution on the lubrication film of cylindrical roller bearings in both experimental and numerical simulation methods. The results show that the deformation of the bearing surfaces can be accelerated by the non-uniform loading distribution. Gopalakrishnan and Ruban [14] also studied the contact analysis of roller bearing using ANSYS. Through this simulation, the stress, strain, contact penetration and frictional stress on the roller bearing were determined. The solution from ANSYS also shows that there was good consistency between the finite element solution and the Hertzian theory solution.

On the other hand, Scari and Pedro Jr. [15] evaluated the tapered roller bearing using a non-linear finite element (FE) analysis with both complete and simplified bearing model. They compared the contact stress results at both tapered roller and the outer raceway. The results show that the outer raceway presents higher contact stress than the tapered roller. In addition, they concluded that the simplified FE model is more suitable for this analysis subjected to thrust loads compared to the complete FE model as the simplified model shows a better correlation with the analytical method.

Edwin [16] studied the contact force between the roller and the race in a roller bearing with a numerical model for mechanical event simulations. The numerical model of a cylindrical roller bearing provided with the spatial and time distribution of stress and strain values and was validated by the analytical model presented by Harris-Jones without any significant differences.

Dong et al. [17] developed a numerical solution of thermal elastohydrodynamic lubrication (TEHL) of tilted roller pairs. They claimed that the von Mises stress acting on the tilted roller increase as the tilting angle increases. Also, they found that the distributions for both film pressure and von Mises stress of the TEHL contact are different from the Hertzian contact results.

Bryant et al. [18] studied a finite element analysis of rough surface line contact between a cylinder and rigid plane. The model showed that multiple contacts in contact with a rigid plane has influenced the contact pressure and the residual deformation is limited to the asperity contact area. Li [19] developed a mathematical model and numerical model for contact analysis of ball and roller bearings. The contact stress on the ball and roller models in the FE software were different than calculated stress using Hertz theory as the Hertz theory did not consider the total deformation of the ball and roller while the software considered the total structural deformation.

Purushothaman and Thankachan [20] conducted the Hertz contact theory validation using finite element analysis (FEA) between two rollers. The analytical calculation was also checked using a Hertz stress calculator software and the 2D model was developed by using ANSYS. The results of maximum shear stress obtained between FEA and analytical solution only had the difference of about 0.61% while the results of contact patch width obtained between FEA and analytical calculation had the difference of 2.46%.

Sulka et al. [21] conducted a simple static structural analysis of ball bearing using FEM software ANSYS Workbench. The results showed that the maximum contact pressure was 3706.1 MPa, which was close to the value using the analytical method of 3663.62 MPa. Tang and Sun [22] studied the contact analysis of deep groove ball bearing (SKF 6020) using APDL language in the finite element software ANSYS. The simulates results showed good consistency with the Hertzian theory solution, with the maximum contact stress in simulation was 8599 MPa while the theoretical value was 8572 MPa.

The contact stress and deformation in dry contact were presented, however, the simulation with lubricated roller contact is rarely reported. Researches on the rolling contact friction have been started for the railway application. Toumi et al. [23] developed a 3D finite element model of the frictional rolling in wheel-rail contact and found that the normal contact solution in elasticity from different FE methods is in good agreement with Hertzian theory for the maximum contact pressure and the contact shape. The effect of rolling friction on the surface crack [24] and rolling contact fatigue and thermal cracks damage [25] of the railway track model also has been studied. Kudra and Awrejcewicz [26] developed the approximate models of coupled friction and rolling resistance in the case of elliptic contact and concluded that generally rotational motion of the deformation zone can also have some influence on the contact stress distribution.

2.3 SUMMARY

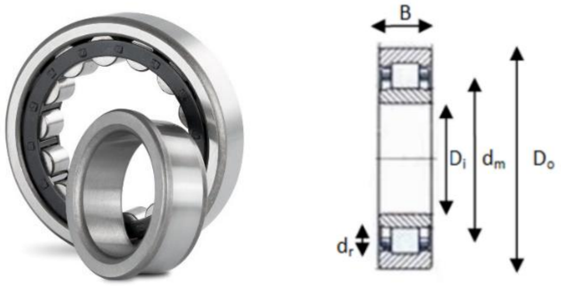
Most of the researchers have focused on building the finite element model of rolling element bearing in order to study the stress and deformation acting on the contact points between the rollers and the races, with some of those numerical results are verified with Hertzian contact theory. There are also researchers who have conducted various tests to determine the coefficient of friction of several types of lubricants. However, the study on the effects of different lubricants to the contact stress and deformation of the roller bearings is very limited. Hence, this project will be carried out to fill the gap that exists between the researches in order to investigate the contact characteristics of lubricated roller bearings.

CHAPTER 3
METHODOLOGY

The methodology of this project could be divided into two major parts. The first part was the experimental work, which consists of the synthesis of palm-oil based grease and the tribology test of different types of grease using Four-ball tester to determine their coefficient of friction. The second part of this project was the simulation of the finite element model of lubricated roller bearing was conducted in static condition based on the Hertzian Contact Theory in 2D and 3D. The values of friction coefficient for the respective grease were applied as part of the setup of the simulation using commercial software ANSYS. From the simulation, the stress and deformation will be obtained and analyzed to observe the contact characteristics of lubricated roller bearings.

3.1 ROLLER BEARING SPECIFICATIONS AND MODELLING

In this project, a detachable cylindrical roller bearing (NU205E.TVP2.C3) was selected for this study. It consists of an outer ring, an inner ring and 13 cylindrical rollers which are hold by a cage to prevent the rollers from coming into contact with each other. The rings and rolling elements of this bearing are made of low-alloy, thorough hardened chromium steel (No. 1.3505, DIN 100Cr6, AISI 5210) [27]. It is designed to support high radial loads and is widely used in various machineries such as gearbox. The specifications of this bearing as well as the properties of the material were listed in Table 3.1 and was then obtained and transferred into a finite element model.



(a) (b)

Figure 3.1: (a) NU205E.TVP2.C3 cylindrical roller bearing with detachable inner ring (b) Schematic cross section of the bearing

Table 3.1: Specifications of cylindrical roller bearing NU205E.TVP2.C3

Properties	**Values**
Number of rollers	13
Roller diameter (d_r)	7.5 mm
Inner bore diameter (D_i)	25 mm
Outside ring diameter (D_o)	52 mm
Pitch diameter (d_m)	38.5 mm
Width (B)	15 mm
Weight	0.14 kg
Young's Modulus of rings and rollers (E)	210 GPa
Poisson's ratio of rings and rollers (v)	0.30
Length of the roller (L)	9 mm

The geometry of the roller bearing is modelled using Computer-Aided Drawing application Solidworks. The 3D model is then used for the simulation.

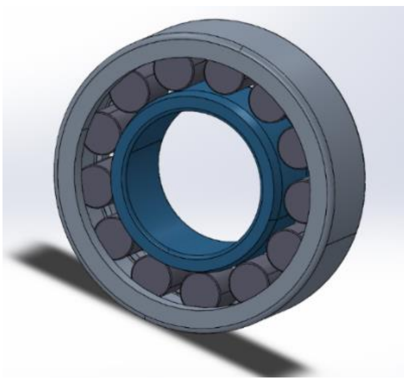


Figure 3.2: 3D model of the cylindrical roller bearing (NU205E.TVP2.C3)

3.2 HERTZIAN CONTACT THEORY

Hertzian contact stress refers to the localized stresses which develop as two curved surfaces come in contact and deform slightly under imposed load. The contact stress and deformation are functions of the normal contact force, the radii of curvature of both bodies, the Young's modulus and the Poisson's ratio of the two bodies.

In this project, a cylindrical roller bearing (NU205E.TVP2.C3) will be used for the simulation. Both the rollers and the races are in a cylindrical shape, hence the contact of these two elements is considered to be the Hertzian contact of the cylindrical surface. The type of contact is a line contact initially before load application, then it becomes a "rectangle" contact once the bodies are deformed under load.

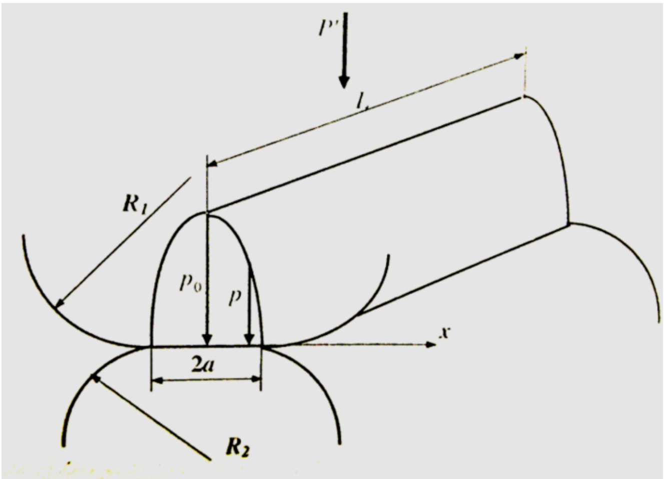


Figure 3.3: Schematic diagram of Hertzian contact of two cylindrical surfaces

3.2.1 Governing Equations

The effective radius of curvature, R is given by [28]

$$\frac{1}{R} = \frac{1}{R_1} + \frac{1}{R_2} \tag{3.1}$$

where

R_1: radius of the roller

R_2: radius of the inner race

The effective Young's modulus, E^* on the Hertzian contact is determined by

$$\frac{1}{E^*} = \left(\frac{1 - v_1^2}{E_1} + \frac{1 - v_2^2}{E_2}\right) \tag{3.2}$$

where

E_1, E_2: elastic moduli associated to the roller and the inner race respectively

ν_1, ν_2: Poisson's ratios associated to the roller and the inner race respectively

The contact radius between the two cylindrical surfaces under load is given by

$$a = \left(\frac{4PR}{\pi E^*}\right)^{\frac{1}{2}} \tag{3.3}$$

where

P: load per unit length (N/m)

The maximum Hertzian contact stress, P_o is given by

$$P_o = \frac{2P}{\pi a} = \left(\frac{PE^*}{\pi R}\right)^{\frac{1}{2}} \tag{3.4}$$

and the interference, δ is given by

$$\delta = \frac{P}{\pi E^*}\left[\ln\left(\frac{L^2 \pi E^*}{2RP} + 1\right)\right] \tag{3.5}$$

3.3 EXPERIMENTAL SETUP

For the experimental part of this project, there are four different types of grease were tested to determine their respective friction of coefficient (cof). The first two types of grease were commercially available, which were Biogrease Marlin-9 Food Grade Grease and SKF LGGB 2/0.4 Biodegradable Bearing Mineral Grease. Meanwhile, the other two types of grease were palm oil based, in which palm oil was synthesized with other chemicals to form a biodegradable and sustainable grease. They were labelled as palm-oil grease formulation A and B respectively.

(a) (b)

Figure 3.4: (a) Food grade grease and (b) mineral grease samples

3.3.1 Synthesis of Grease

In the grease synthesis experiment, cooking oil was used as the base material, along with lithium stearate as thickener and copper nanoparticles as additives [29]. Each of the chemicals were carefully weighed separately based on the compositional formulation given in Table 3.2. The contents were mixed together in a beaker. Then, the mixture was heated to a temperature of 90-120 °C while being stirred using magnetic stirrer for about 4 hours. After that, the heating plate was turned off, but the stirring was continued until the contents were cooled to room temperature. The grease was covered with a plastic wrap and left overnight to check for its stability. The steps were repeated for a different composition formulation. The grease samples were then taken to conduct further experiments.

Table 3.2: Composition formulations of grease samples

Material	Composition (%)	
	A	**B**
Cooking oil	93.5	87.6
Lithium stearate	5	11
Copper nanoparticles	1.5	1.4

3.3.2 Tribology Test Using Four-Ball Tester

A tribology experiment was carried out to determine the friction coefficient values of four different types of grease, which are palm oil grease formulation A, palm oil grease formulation B, food grade grease and mineral grease. This experiment will be following the standard testing procedure ASTM D2266 with the test parameters shown in Table 3.3. This experiment will be conducted using four-ball tester, which was proposed by Boerlage [30].

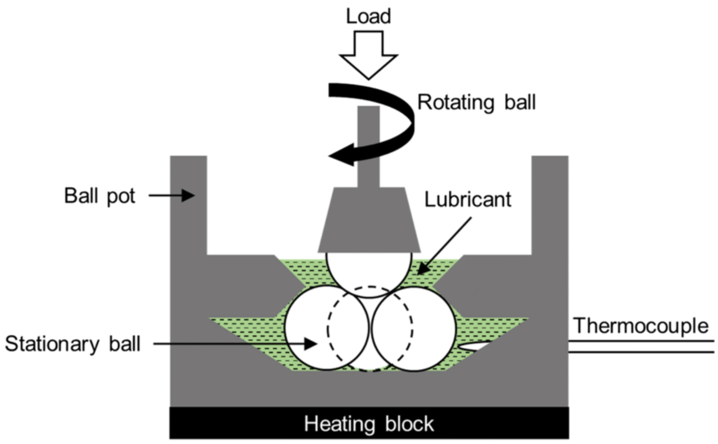


Figure 3.5: Schematic diagram of the four-ball tester [31]

Table 3.3: Test parameters according to ASTM D2266

Parameter	Condition
Load applied	40 kg (392 N)
Duration	60 min
Temperature	75±2 °C
Speed	1200±60 rpm

Four standard steel balls were used in this experiment, with three balls held firmly in a pot containing the grease and one ball on the top pressed against them, as shown on the schematic diagram in Figure 3.5. Before the steel balls can be installed, they were cleaned using acetone and wiped dry to make sure the steel balls were free of contaminants.

Three steel balls were placed into the ball pot and the assembly was tightened by using a torque wrench. Meanwhile, the top steel ball was locked inside the collector and fixed onto the spindle. The pot was then filled with the test grease. The pot was installed onto the machine and the load was gently applied. The grease was heated to 75 °C via built-in heater. Then, when the set temperature was reached, the motor was started. After an hour, the machine was stopped and the assembly is removed from the machine. The grease in the cup was removed. The components in the assembly were removed, cleaned with acetone and the experiment was repeated with three other types of grease.

The results from the four-ball tester were recorded using the data acquisition system and generated in the form of Microsoft Excel spreadsheet in computer, which consist of normal load and friction torque at intervals of one second. The average friction coefficient for each test was calculated according to IP-239, which is expressed as follows [9]:

$$\mu = \frac{T\sqrt{6}}{3Wr} \tag{3.6}$$

where μ is the coefficient of friction, T is the friction torque in kg mm, W is the applied load in kg and r is the distance from the center of the contact surface on the lower balls to the axis of rotation, which was determined to be 3.67 mm. Hence, the equation to calculate the friction coefficient can be simplified as follows:

$$\mu = 0.22248\frac{T}{W} \tag{3.7}$$

3.4 FINITE ELEMENT ANALYSIS OF ROLLER BEARING

The finite element analysis of lubricated roller bearing is performed based on the Hertzian Contact Theory in both two-dimensional (2D) and three-dimensional (3D) models. Generally, for both scenarios, the steps to conduct the finite element analysis (FEA) using ANSYS Static Structural is summarized on the flowchart in Figure 3.6.

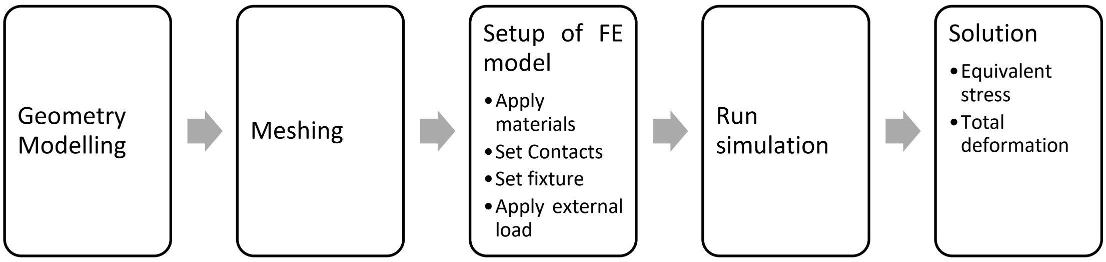


Figure 3.6: Flowchart of finite element analysis of roller bearing in ANSYS

3.4.1 2D Analysis of Roller Bearing

The 2D model of the cylindrical roller bearing (NU205E.TVP2.C3) could be simplified by studying the model using only one inner ring and one roller on top of it, and only half of inner ring and roller were included in the simulation to simplify and reduce the computing power and memory required. For this study, both inner ring and roller were considered to be in contact as per plane stress theory criteria to improve the accuracy of the results [20].

Based on the actual dimensions of the roller bearing, the roller was sketched with a radius of 3.75 mm. As for the inner ring, it was modelled with its inner radius of 12.5 mm and outer radius of 15.75 mm. The properties of the roller bearing material, which is low-alloy, thorough hardened chromium steel (AISI 5210) was created on the engineering data in ANSYS and its material properties were updated. Then, both inner ring and roller were assigned with this custom material for finite element study.

On the setup of the simulation, the contact between the edges at the inner race and the roller was set at frictionless. The frictionless contact was selected as a reference since the Hertzian Contact Theory assumes that the contact between two metal bodies is frictionless. The boundary conditions of the bearing were specified as shown in Figure 3.7, with normal load of 100 N on the roller acting downwards. The displacements of both inner ring and roller were fixed in the x-direction and only allowed in the y-direction.

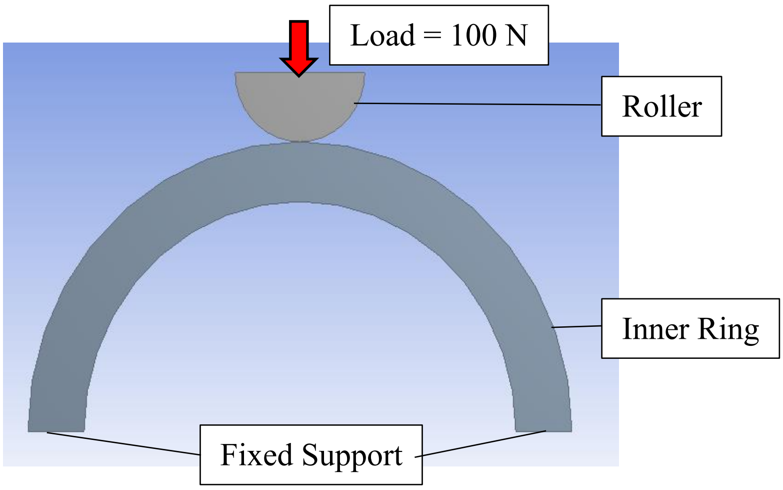


Figure 3.7: 2D model of the partial roller bearing and its boundary condition for simulation

The next step was the meshing of the bearing model, with element size of 0.4 mm using all triangles method. The contact area between the inner race and the roller was given a much finer mesh sizing of 0.0055 mm. The generated mesh was consisting 132788 elements as shown in Figure 3.8.

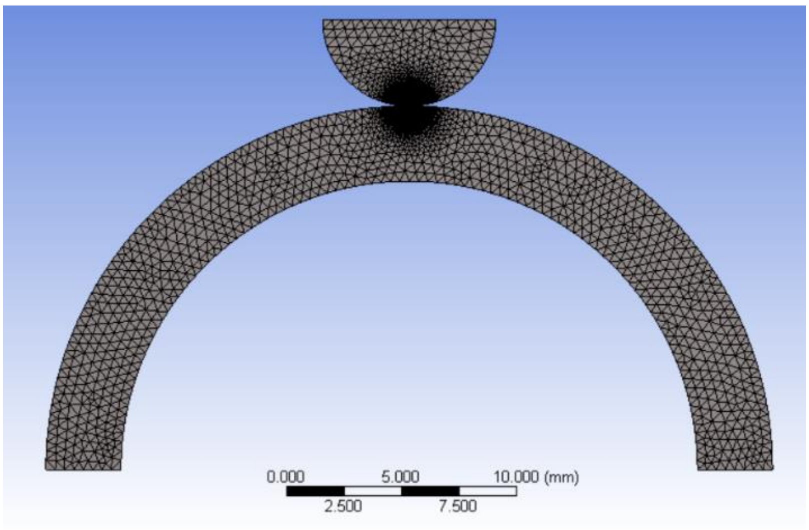


Figure 3.8: Meshing of the 2D roller bearing model

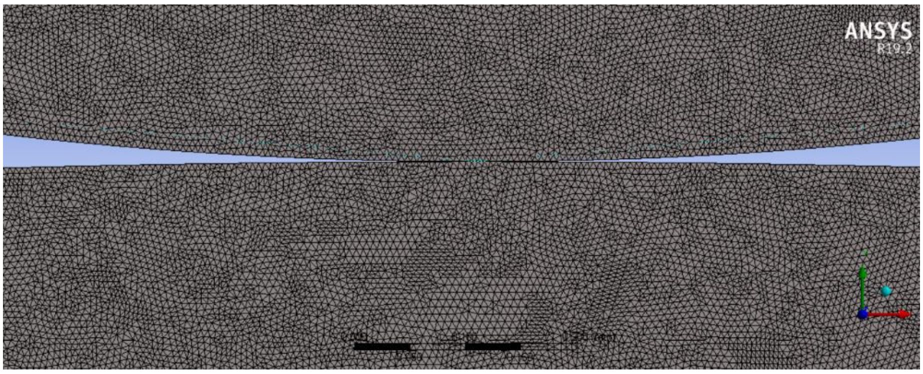


Figure 3.9: Detailed mesh at the contact area between roller (top) and inner ring (bottom)

After setting the boundary conditions, the simulation was run and solved. The equivalent stress and total displacement of the bearing model were displayed in contour form and the results were recorded. The simulation was then repeated by changing the contact from frictionless to frictional, with the coefficient of frictions (μ) of four grease samples were included as part of the initial setup for the simulation.

3.4.2 3D Analysis of Roller Bearing

The 3D model of the cylindrical roller bearing (NU205E.TVP2.C3) was imported into ANSYS Structural Static. The model could be simplified by studying the model using only one inner ring and one roller on top of it to simplify and reduce the computing power and memory required. The simulation was performed with the assumption of the radial load exerted by the bearing was applied directly on top of only one cylindrical roller, and only the non-conformal contact between the inner race and the roller was taken into consideration.

On the setup of the simulation, the 3D model was assigned to the actual material of the bearing (AISI 5210). The contact between the surfaces at the inner race and the roller was set at frictionless. The boundary conditions of the bearing were specified as shown in Figure 3.10, with normal load of 100 N applied directly to the roller. The displacements of both inner ring and roller were fixed in two direction and only allowed in the radial direction.

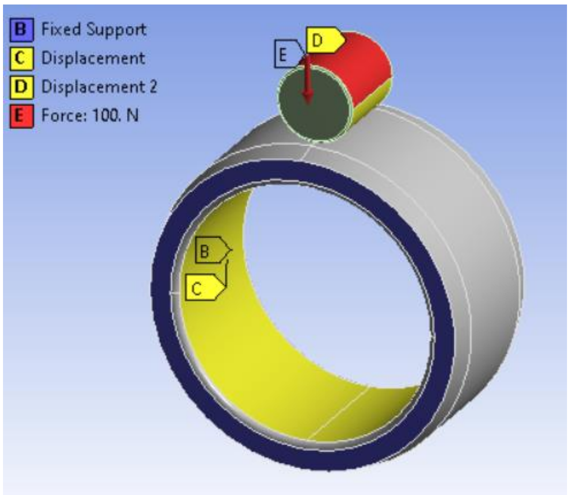


Figure 3.10: Boundary conditions of 3D partial roller bearing model

The next step was the meshing of the bearing model, with element size of 1.0 mm using automatic method. In addition, the contact region between the inner race and the roller was assigned as contact sizing with element size of 0.25 mm. The generated mesh was consisting 60302 elements as shown in Figure 3.11.

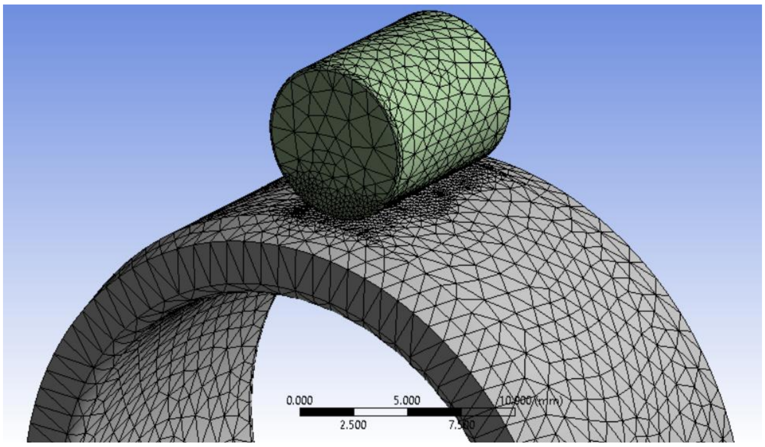


Figure 3.11: Meshing of the 3D roller bearing model

After setting the boundary conditions, the simulation was run and solved. The equivalent stress and total displacement of the bearing model were displayed in contour form and the results were recorded in the next chapter. The simulation was repeated by changing the contact from frictionless to frictional, with the coefficient of frictions (μ) of four grease samples included as part of the conditions for the simulation.

CHAPTER 4
RESULTS AND DISCUSSION

4.1 EXPERIMENTAL RESULTS

Based on the Four-ball tester experiment, the results of normal load and friction torque across time were plotted as shown in Figures 4.1 to 4.4. From there, the average normal load and frictional torque for each grease samples were acquired and the average coefficient of friction (cof) of each grease samples can be calculated based on Equation 3.6. The summary results from this experiment were tabulated in Table 4.1.

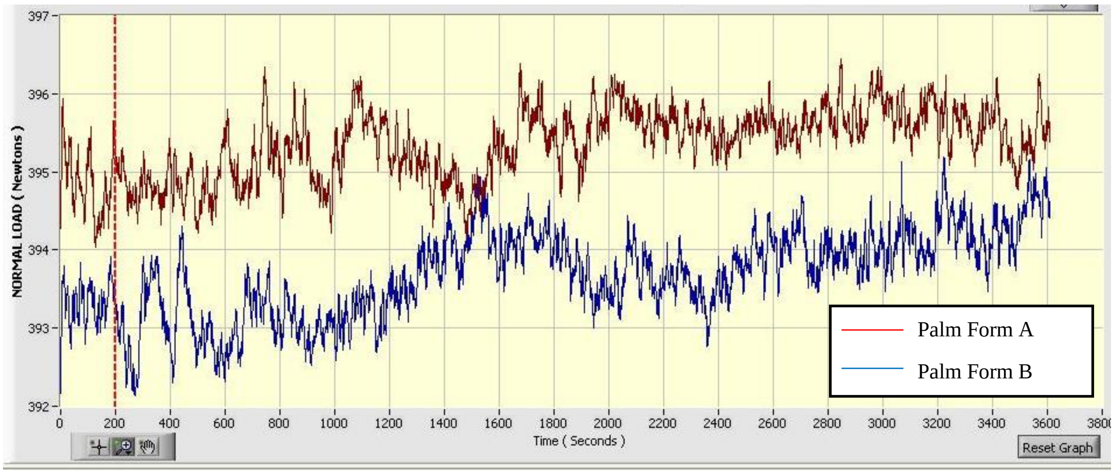


Figure 4.1: Graph of normal load against time for palm oil grease formulations A and B

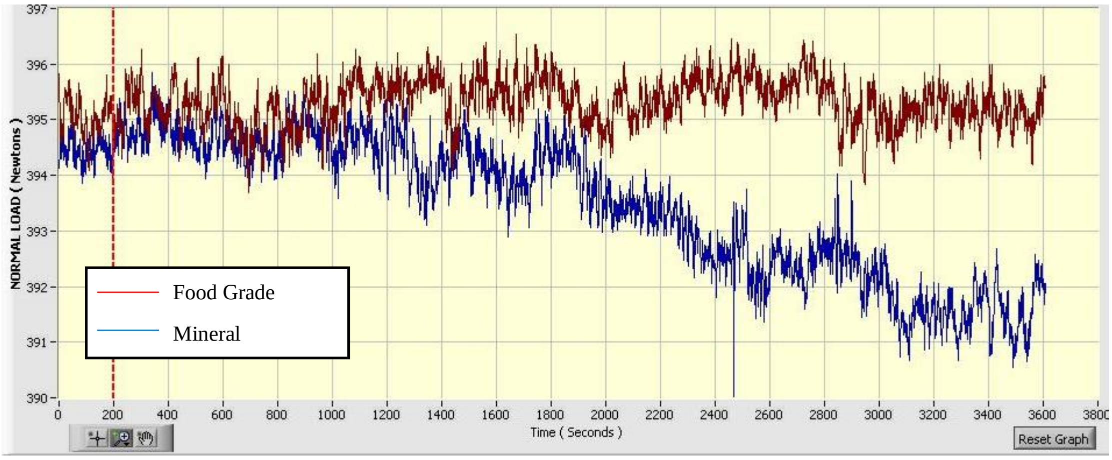


Figure 4.2: Graph of normal load against time for food grade grease and mineral grease

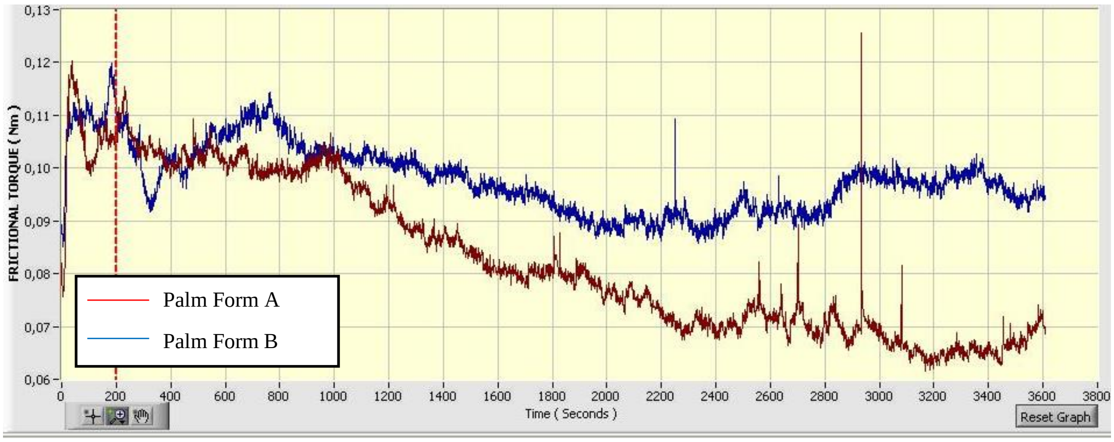


Figure 4.3: Graph of frictional torque against time for palm oil grease formulations A and B

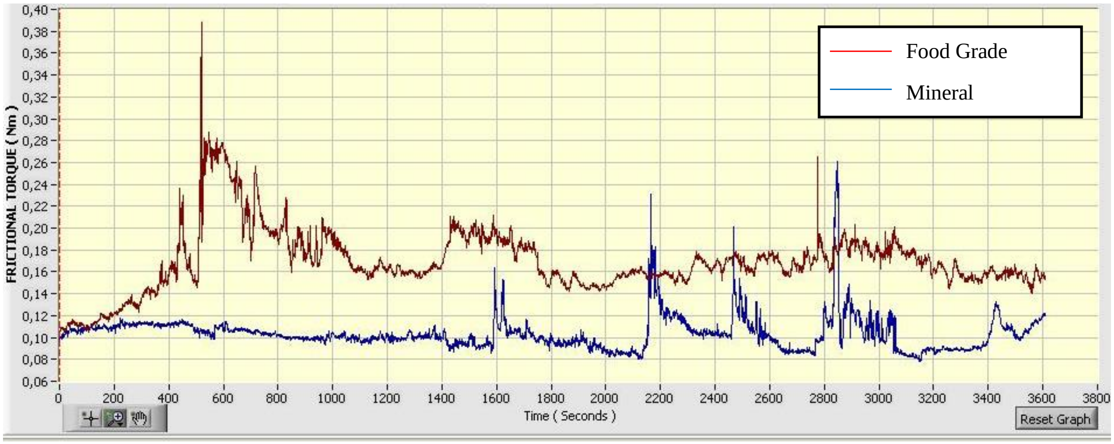


Figure 4.4: Graph of frictional torque against time for food grade grease and mineral grease

Table 4.1: Average coefficient of friction of four different grease from tribology experiment

Grease	Avg. Normal Load (N)	Avg. Frictional Torque (Nm)	Coefficient of Friction
Palm oil form A	395.35	0.0832	0.046842
Palm oil form B	393.68	0.0979	0.057769
Food grade	395.31	0.1693	0.096311
Mineral	393.50	0.1036	0.063654

Based on the results in Table 4.1, it could be observed that the average normal load recorded for all four grease samples had almost the same values, ranging from 393.5 N to 395.3 N. Meanwhile, it could be also observed that food grade grease recorded the highest average frictional torque of 0.169 Nm, followed by mineral grease, palm oil grease formulation B and finally palm oil grease formulation A, which recorded the lowest average frictional torque of 0.083 Nm.

The coefficient of friction for food grade grease was determined to be the highest at 0.096311, followed by mineral grease at 0.063654, palm oil grease formulation B at 0.057769 and finally palm oil grease formulation A at 0.046842. Interestingly, an increment in the average frictional torque also increased the average coefficient of friction. In other words, the average coefficient of friction was directly proportional to the frictional torque at constant normal load. The findings show that the newly developed palm oil-based grease has a good performance and has the potential to be applied in rolling contact applications due to low friction characteristics.

4.2 RESULTS OF FINITE ELEMENT ANALYSIS

4.2.1 Mesh Convergence Test

The mesh convergence test, also known as the grid independence test, was performed before running the simulation of the roller bearing model. It is important in numerical simulation analysis in order to provide results from simulation which are independent of numerical grids.

For the simulation of 2D partial roller bearing, the mesh was created from relative coarse size to a much finer mesh, with mesh number ranging from 2900 to 243374 elements. In this case, the mesh of 132788 elements was selected for the 2D simulation as the changes of both stress and deformation were almost constant beyond that. In addition, the simulation with this mesh number yield much faster computation time as compared to that with 243374 element size.