An Experimental Investigation on Friction and Scuffing Failure of Lubricated Point Contacts

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An Experimental Investigation on Friction and Scuffing Failure of Lubricated Point Contacts

A thesis submitted in partial fulfillment of the requirements for the degree of Master of Science in Engineering

By

Sai Goutham Soma
B.Tech., Jawaharlal Nehru Technological University, 2013

2015
Wright State University
I HEREBY RECOMMEND THAT THE THESIS PREPARED UNDER MY SUPERVISION BY Sai Goutham Soma ENTITLED An Experimental Investigation of the Surface Roughness on Scuffing Failure of the Lubricated Point Contacts BE ACCEPTED IN PARTIAL FULFILLMENT OF THE REQUIREMENTS FOR THE DEGREE OF Master of Science in Engineering.

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ABSTRACT

Soma, Sai Goutham. M.S.Egr., Department of Mechanical and Materials Engineering, Wright State University, 2015. An Experimental Investigation on Friction and Scuffing Failure of Lubricated Point Contacts.

Scuffing failure is a catastrophic thermal failure mode induced by the extreme surface temperature of the contacting components. As two mechanical elements roll against each other with high sliding motion, the frictional heat flux elevates the surface temperature to exceed the critical limit, resulting in the welding of the surfaces. The relative motion then tears the welded surfaces apart, causing the damage. Scuffing failure has been an important failure mode for rolling machine elements such as bearings and gears in aerospace applications, owing to the very high operating speeds. Recently, this failure mode has extended to the automotive field, where the power density of the transmission system has been continuously increasing. Employing a two-disk contact set-up, this experimental study investigates the scuffing load carrying capacity of two new alloys, which are paired with the lubricant of the Mil-PRF-23699 turbine fluid, in comparison to the baseline material of AISI 5120, under different speed conditions. The variations of the friction coefficient with the sliding are also quantified for all the three alloy-lubricant pairs under various load and speed conditions. It is observed that the new materials failed to improve the scuffing performance with the lubricant and the operating conditions considered. It is suggested to use a different additive package in the lubricant, which may be more effective for the formation of the protective tribo-film along the new material surfaces. It is also suggested to extend the operating speed range for higher speed applications.
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Dedicated to my parents
CHAPTER 1

INTRODUCTION

1.1 Background and Motivation

Scuffing failure is a catastrophic thermal failure mode induced by the extreme surface temperature of the contacting components. As two mechanical elements roll against each other with high sliding motion, the frictional heat flux produced at the contact interface can be sufficiently large to elevate the local surface temperature, which is the sum of the bulk temperature and the flash temperature, to exceed the critical scuffing limit [1,2]. This very high surface temperature results in the welding of the two surfaces. The continued relative motion, however, tears the surfaces apart, resulting in the scuffing damage. Scuffing failure has been an important failure mode for rolling machine elements such as bearings and gears in aerospace applications, owing to the very high operating speeds. Recently, this failure mode has extended to the automotive field, where the power density of the transmission system has been continuously increasing due to the demand for high power while compact size.

The frictional heat power, $Q$ is determined by two factors, i.e. the friction force, $F$ and the sliding velocity, $u_s$ as
The main causes for the overwhelming frictional heat between the contacting surfaces can be categorized into three groups. The first is the very harsh operating conditions, such as the heavy load, the high sliding velocity, and the high lubricant temperature. When the normal load is large, the resultant contact pressure and tangential shear distributions within the contact zone are large as well, leading to the significant heat production. The sliding velocity of the contacting surfaces influences the friction power in two aspects: (i) the direct impact through Eq. (1.1) and (ii) the lubricant shear-thinning owing to the fluid non-Newtonian behavior when the shear rate (sliding) is high. The latter effect reduces the lubricant fluid film thickness and even breaks down the hydrodynamic lubrication film, resulting in the metal-to-metal contact (boundary lubrication) and the increased friction. The lubricant temperature affects the viscosity (higher temperature leads to lower viscosity) and consequently the lubricant film thickness. Under the high lubricant temperature condition, the lubricant viscosity is low and the established hydrodynamic fluid film may not be able to separate the two surfaces fully. As a result, direct metal-to-metal contacts take place. In addition, the lubricant temperature impacts the bulk temperature of the contacting components, which constitute an important portion of the total surface temperature [2].

The second group of the potential causes for the large frictional heat flux includes the high roughness amplitude, and any wear and fatigue debris in the lubricant. Due to the finishing processes of machine components (grinding and shaving in the production of gears for instance), significant tool marks are left on the surfaces and forms the roughness profiles. The roughness peaks produce localized asperity contacts between the two surfaces, and dictate the localized heat flux and consequently the flash temperature rise [2]. The debris in the lubricant act as stress raisers the similar way as
that of the surface asperities, leading to extreme contact pressure and shear, and thereby the significant local heat flux. Another possible mechanism of debris caused scuffing failure is that the large amount of debris block the inlet of the contact zone, such that the lubricant cannot be entrained hydrodynamically into the contact, resulting in starvation and the sudden lubrication film collapse [3]. The last group of the causes is the depletion of the lubricant additives and the loss of the protective tribo-film produced through the chemical reaction between the metal surfaces and the lubricant additives. Different additives in the lubricant can produce tribo-films with different properties, such as low wear and low friction. The low wear characteristic can reduce the wear debris in the lubricant and consequently reduce the chance of debris induced scuffing failure. And the low friction characteristics can achieve low boundary friction when metal-to-metal contacts occur. As the reactions between the metal surfaces and the additives continue, the latter can be used up after certain amount of service life. As a result, the beneficial effects of the tribo-film is lost.

The objective of this study is to experimentally evaluate the scuffing resistance performance of two heavy duty aerospace materials, namely Material A and Material B, with material 5120 under different rolling-sliding combinations and contact loads, using a twin-disk contact set-up that has been newly developed. The specimens are finished through circumferential grinding, such that the roughness lay directionality is parallel to the rolling and sliding direction. A typical turbine fluid MIL-PRF-6399 is used as the lubricant. The direct contribution of this study will be a data base for the limiting contact pressure and sliding velocity of scuffing failure for the materials and lubricant considered, providing a highly demanded design guideline for the elimination of scuffing failure in machine elements such as rolling element bearings and gears. In addition, a set of experiments will be performed to measure the friction coefficient
under various operating conditions, which can be used for power loss and mechanical efficiency evaluations.

### 1.2 Literature Review

The extreme local surface temperature, which is the sum of the surface bulk temperature, and the instantaneous flash temperature induced by the local frictional heat flux, is the direct cause of the scuffing initiation. Enthoven and Spikes [3] measured the surface temperature distribution of a steel ball that was in contact with a sapphire flat disk, using an infrared microscope. It was found both the temperature at the inlet and the maximum temperature of the entire contact zone increased sharply within a 0.6 second time window right before the scuffing failure. The inlet surface temperature increased from 110 °C to 250°C; and the maximum surface temperature was raised from 170 °C to 420 °C, where scuffing occurred. For the contact of both steel surfaces, the temperature measurements within the contact zone is difficult to perform. Therefore, only the surface bulk temperature was available in the twin-disk type of contact set-up such as in Refs. [2, 4-6]. The estimation of the flash temperature was then carried out using various prediction models, such as the well-known closed-form Blok’s formula [1].

The very early flash temperature modeling activity of Blok [1] assumed smooth surface condition and uniform heat flux across the contact, which is usually not true for actual machine elements whose surface roughness is prominent. It was shown by Ling [7] that any asperity contact can significantly elevate the surface temperature. This observation has been well confirmed by the modern thermal elastohydrodynamic lubrication (EHL) models for rough surfaces, which are capable of finding the detailed
temperature distributions of both the lubrication fluid film and the contact surfaces in a deterministic way. Computer generated Gaussian roughness profiles were considered in the works of Lai and Cheng [8] and Qiu and Cheng [9], and the resultant solid surface flash temperatures were evaluated without solving the energy equations of the fluid and the bounding solid surfaces. Incorporating the energy equations into the EHL governing equations for a line contact, Cioc et al [10] numerically determined the flash temperatures of both the fluid and the solid surfaces under full film lubrication condition. Considering the more realistic mixed lubrication circumstance where the hydrodynamic fluid film and the asperity contacts coexist and share the normal load, Deolalikar et al [11] determined the flash temperature distributions within the contact by treating the fluid regions and the asperity contact regions separately. Zhu and Hu [12] and Wang et al [13] used a unified EHL governing equation system to solve for the temperatures in a numerically more stable way. Li et al [2] and Li [14] employed a novel discretization scheme for the fast and accurate solution of the thermal mixed EHL governing equations. It was shown for a specific lubricant-steel pair, there exists a critical scuffing limit temperature, exceeding which, the scuffing failure occurs.

The effects of operating conditions (normal load, rolling velocity, sliding velocity and lubricant temperature), surface conditions (roughness amplitude and lay directionality, coating, etc.), and different lubricants (viscosity properties, additives, etc.) have been the topics of discussion when scuffing comes into play. The works such as Refs. [2,4-6,8,15] showed the scuffing load-carrying capacity decreases when the sliding increases, using a twin-disk contact set-up. Considering the contact of gears, Hohn et al. [16] showed that the increase in lubricant temperature not necessarily reduces the scuffing resistance. Although the lubricant viscosity and consequently the
film thickness becomes smaller when the temperature is increased, the EP additives within the lubricant can perform better to form a protective tribo-film.

The surface roughness has always been an important factor influencing scuffing failure. The extensive experiments performed by Patching et al. [4], Alanou et al. [5], Nakatsuji and Mori [17], Shon [18], and Liou [19] examined the impacts of the surface roughness amplitude on scuffing failure. These experiments paved to a conclusion that the reduction in the surface roughness amplitude promotes the scuffing performance. Another group of studies focused on the roughness lay directionality on scuffing failure. Ichimaru et al. [20] found improved scuffing resistance when the longitudinal surface roughness texture was replaced by the transverse roughness orientation. This experimental observation was confirmed by the computational studies of Li [14] and Horng et al. [21], which also demonstrated the weak scuffing resistance of oblique roughness texture patterns in comparison to the transverse roughness texture. To improve the scuffing load capacity, the studies such as Alanou et al. [5], Snidle et al. [6], and Shon [18] implemented hard coatings onto the surface, showing the effectiveness in friction reduction and scuffing performance enhancement in comparison to the ground surfaces without any coating. This observation was found to be also valid under the starved lubrication condition [18].

The lubricant properties, including viscosity and additives, have evident impacts on the scuffing performance of a contact pair. For lubricant with larger viscosity, a thicker lubrication film can be achieved. However, it is noted that the higher viscosity also leads to higher frictional heat, which may offset the benefit of a thicker fluid film. The non-Newtonian effect, i.e. the shear-thinning effect, when the shear strain rate is high is another important lubricant property [22]. Since gears often operate under high sliding condition, the shear-thinning behavior is significant away from the
pitch line. The reduced lubrication film thickness, thereby, leads to the elevated frictional heat between the contacting surfaces and scuffing failure. The additives in lubricants is a popular research topic for the reduction of scuffing failure. The additives react with the metal surface, forming a chemical layer (also referred as the tribo-film), sulphide layer on the steel surface (FeS) for instance, which acts as a barrier between and prevents direct metal-metal contacts. The studies such as [20,23] examined the influences of different lubricant additives on scuffing failure. Using a twin-disk set-up, Ichimaru et al [20] concluded that the scuffing resistance of a steel could be improved by implementing additives to form tribo-films that are capable of insulating the surface from the frictional heat or low friction.

The lubricant properties on the molecular scale can also be important. Askwith [24] and Klamann [25] showed high polarity lubricant molecules get adsorbed on the surfaces easily whereas low polarity molecules do not. Therefore, a lubricant with high polarity molecules can get adsorbed on the surfaces and form a nanoscale thickness of lubrication film more effectively, reducing the boundary lubrication friction. Lee and Chen [26] proposed a scuffing failure mechanism based on the physisorption behavior of lubricants. The surface temperature of asperities along with the lubricant pressure affect the rate of desorption, to which the frictional temperature is proportional. The hydrodynamic pressure, however, is proportional to rate of adsorption.

1.3 Thesis Objectives

In view of the experimental studies on scuffing in literature, the scuffing load carrying capacity of the two alloys (Material A and Material B) considered in this study, which are paired the lubricant of the turbine fluid of Mil-PRF-23699, is missing. The
corresponding friction coefficient measurements are neither available. Therefore, the specific technical objectives of this study are summarized as below:

- Investigate the scuffing load carrying capacity for the combination of Material A alloy, which is case-hardened, and Mil-PRF-23699 lubricant, under different speed condition.
- Investigate the scuffing load carrying capacity for the combination of Material B, which is case-hardened and Mil-PRF-23699 lubricant, under different speed condition.
- Quantify the friction coefficient for both the alloy-lubricant pairs under various load, speed, and sliding conditions.

The experiments will be carried out using a newly developed twin-disk contact set-up. The contact specimens will have the roughness lay direction to be parallel to the rolling and sliding direction.

1.4 Thesis Outline

The outline of this thesis is listed below:

- Chapter 2: The newly developed twin-disk contact set-up and the test specimens will be introduced in detail. The procedure of the experiments, including both the test and inspection processes, will be described.
- Chapter 3: The experimental test matrix will be constructed. The test results will be documented and discussed.
- Chapter 4: The research activity will be summarized. Conclusions and recommendations for future work will be provided.
CHAPTER 2

TWO-DISK TEST METHODOLOGY

2.1 Two-Disk Tribo-meter Description

The two-disk contact set-up as shown in Fig. 2.1 is used in this study for the measurement of friction and scuffing failure. Using the schematic layout of Fig. 2.2 as the illustration, this tribo-meter consists of two identical belt driven spindles, each of which is powered by an independent 3-phase 11.2 kW (15 HP) motor to allow different surface velocity combinations of the roller-disk contact pair. The connecting shaft between the belt and the spindle is supported by two pairs of high speed angular contact ball bearings assembled in the back-to-back configuration to be able to accommodate large radial and axial loads in either direction. A high speed spindle grease is applied in the bearing cavities to eliminate the need for the additional bearing lubrication system required for lubrication oils. The bearing assembly is packaged in the bearing housing for each of the shaft. For the measurement of the frictional torque produced within the roller-disk contact, the connecting shafts are equipped with the high precision torque meters between the bearing housing and the belt drive. In this design, the spindle is capable of operating within the rotational speed range of $500 \leq \Omega \leq 10,000$ rpm with
Fig. 2.1  (a) Test compartment of the two-disk tribo-meter, and (b) close-up view of the roller-disk contact pair fully installed.
Fig. 2.2  The schematic layout of the two-disk tribo-meter used in this study.
the accuracy of ±1 rpm in either direction (CW or CCW). The frictional torque measurement range is between 0 and 10.5 Nm (92.9 inlbf) with the accuracy of ±1 %.

As shown in Fig. 2.1 (a), the lower spindle and its bearing housing is attached to the bed of the test compartment and cannot move. The upper spindle and its bearing housing, however, are designed to be able to translate in the horizontal direction along the sliding slots through the pneumatic actuation (The rotational axis is in the vertical direction). This arrangement produces the changeable center distance between 20 mm and 75 mm for the roll-disk contact pair to allow different diameter specimen combinations for the imitation of the contact geometry of, for instance, the different mesh position along the line-of-action of a gear pair. The normal load between the contact pair is provide by the pneumatic cylinder that is attached to the upper bearing housing. The load range is from 300 to 4,500 N with the accuracy of ±5 %. It is measured using a force transducer.

The lubrication system of the two-disk tribo-meter has a 10 liter (2.64 gallon) insulated lubrication reservoir capable of supplying temperature controlled (electrically heated and water cooled) oil to the test specimen at the flow rate ranging from 0 to 0.5 liters per minute (0 to 0.13 gpm). The oil is directed to the specimen interface using a flexible lubrication jet. An in-line cartridge type filter with replaceable elements is used for easy replacement. Guarding and shielding is provided to contain the oil within the testing area and protect operators in the event of test article deterioration. A 1.5 kW in-line oil heater and a 1 kW oil-water heat exchanger control the oil temperature within the range of ambient +10 °C to 150°C. Two thermocouples, one located in the lubricant reservoir and one located at the manifold, to which the flexible lubrication jet connects, are used to monitor the lubricant temperature. The manifold lubricant temperature is assumed to be the same as that of the lubricant supplied at the contact
interface, and used to control the temperature. The lubrication system is plumbed using materials compatible with typical industrial and aerospace oils, and includes no yellow metals.

The two-disk tribo-meter is controlled by a touchscreen PC using National Instruments hardware and a custom LabVIEW application. The system is programmed to be able to operate in several modes including manual control and automatic modes as

1. Constant spindle speeds with load varied in a step-wise or continuous way
2. Constant load with spindle speeds varied in a step-wise or continuous way
3. Block duty cycles (series of spindle speeds and load set points)

2.2 Test Specimens

The contact pair consists of two cylindrical specimens, namely the smaller roller and the larger disk, as shown in Fig. 2.1 (b) in the assembled condition. The roller has the face width of 7.62 mm and the outer diameter of $d_1 = 31.75$ mm as shown in Fig. 2.3. There is no crown applied for the roller. As for the disk, it has the face width of 6.35 mm and the outer diameter of $d_2 = 57.15$ mm as displayed in Fig. 2.4. In order to eliminate any edge loading condition, the axial circular crown of 76.2 mm radius is implemented for the disk. To mount the roller-disk pair onto the test rig, specific fixtures for the roller and disk are required. Figures 2.5 and 2.6 show the assemblies of the roll-fixture and the disk-fixture, respectively. Both the roller and the disk are shrink-fitted on their respective fixtures. A compression washer and a lock nut is then used to securely fasten the roller specimen. For the disk, only a lock nut is used. The assembled roller-fixture and disk-fixture are then mounted onto the lower and the upper
spindle, respectively, using four bolts positioned 90° away from each other circumferentially, as shown in Fig. 2.1 (b).

The roller-disk specimen pairs are made from three different steels. The baseline steel is the AISI 5120 alloy. The other two are referred as new material A, and new material B as shown in Fig. 2.7. All the three material specimens have the surface hardness of 60 HRC. The purpose of this study is to investigate any improvement in terms of friction and scuffing failure of these two new materials in comparison to the AISI 5120 alloy. All the specimen surfaces are first ground and then polished to have the root-mean-square (RMS) surface roughness amplitude of $R_q = 0.16 \mu m$. Figures 2.8 and 2.9, 2.10 and 2.11, and 2.12 and 2.13 show the example roller and disk surface roughness profiles measured along the circumferential direction, using a contact stylus type surface roughness profiler located at The Ohio State University. For these measurements, the Gaussian filter is set to have the upper cut-off of $L_c = 0.25$ mm and the lower cutoff of $L_s = 0.0025$ mm to filter out both the low-frequency surface waviness and the high-frequency noise.

2.3 Test Setup

The two-disk tribo-meter is controlled through the touch screen interface as shown in Fig. 2.14, where a test is ready to be started. When the machine runs, the target operating conditions, including the rotational speed, normal load, and lubricant temperature (manifold thermocouple), as well as their corresponding actual values are displayed on the screen for easy comparisons. The instantaneous frictional torque measurements are also shown. The detailed data is recorded on an USB drive, which allows the quick data transfer.
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Fig. 2.4  Engineering drawing of the disk specimen.
Fig. 2.5  (a) Roller-fixture assembly, and (b) roller-fixture dissembled.
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Fig. 2.8 Measured example surface roughness profiles for the roller specimens of AISI 5120.
Fig. 2.9 Measured example surface roughness profiles for the disk specimens of AISI 5120.
Fig. 2.10  Measured example surface roughness profiles for the roller specimens of new material 
A.
Fig. 2.11  Measured example surface roughness profiles for the disk specimens of new Material A.
Fig. 2.12  Measured example surface roughness profiles for the roller specimens of new Material B.
Fig. 2.13  Measured example surface roughness profiles for the disk specimens of new Material B.
2.3.1 Friction Measurement with Continuous Sliding Sweep

The tribo-meter has two distinct operating modes, namely the friction measurement with continuous sliding sweep, and the rolling contact with block operating condition. The first operating mode of friction measurement is designed to evaluate the friction variation with the continuous sweep of the slide-to-roll ratio, \( SR \), from -1 to 1 under the constant rolling velocity, \( u_r \), and normal force, \( W \). This slide-to-roll ratio is defined as the ratio of the sliding velocity, \( u_s \), to the rolling velocity, i.e.

\[
SR = \frac{u_s}{u_r}
\]  
(2.1)

Where

\[
u_s = u_1 - u_2
\]  
(2.2a)

\[
u_r = \frac{(u_1 + u_2)}{2}
\]  
(2.2b)

with \( u_1 \) and \( u_2 \) representing the roller and disk tangential surface velocities, respectively. These surface velocities are related to their respective rotational speeds, \( \Omega_i \ (i = 1, 2) \) through

\[
u_i = \frac{\pi d_i \Omega_i}{60}
\]  
(2.3)

Substituting Eq. (2.3) into Eq. (2.2), the sliding and rolling velocities are rewritten as

\[
u_s = \frac{\pi}{60} (d_1 \Omega_1 - d_2 \Omega_2)
\]  
(2.4a)

\[
u_r = \frac{\pi}{120} (d_1 \Omega_1 + d_2 \Omega_2)
\]  
(2.4b)

Utilizing Eqs. (2.1) and (2.4), it is found
\[ d_2 \Omega_2 = \frac{2 - SR}{2 + SR} d_1 \Omega_1 \]  

(2.5)

Substituting Eq. (2.5) back into Eq. (2.4b), the rolling velocity is arrived lastly as

\[ u_r = \frac{\pi}{30} d_1 \Omega_1 \frac{1}{2 + SR} \]  

(2.6a)

Or

\[ u_r = \frac{\pi}{30} d_2 \Omega_2 \frac{1}{2 - SR} \]  

(2.6b)

Thus, the minimum and maximum allowable rolling velocities for the friction measurement mode are

\[ \begin{align*}
    u_r^{\text{min}} &= \frac{\pi}{30} d^{\max} \Omega^{\min} \\
    u_r^{\text{max}} &= \frac{\pi}{90} d^{\min} \Omega^{\max}
\end{align*} \]  

(2.7a, 2.7b)

where \( d^{\min} \) and \( d^{\max} \) correspond to the smaller and the larger of \( d_1 \) and \( d_2 \), respectively, and \( \Omega^{\min} \) and \( \Omega^{\max} \) are the minimum and maximum rotational speeds.

Given \( d^{\min} = 0.03175 \) m and \( d^{\max} = 0.05715 \) m for the contact pair considered, and \( \Omega^{\min} = 500 \) rpm and \( \Omega^{\max} = 10,000 \) rpm, the limiting rolling velocities become

\[ \begin{align*}
    u_r^{\text{min}} &= 2.99 \text{ m/s} \\
    u_r^{\text{max}} &= 11.08 \text{ m/s}
\end{align*} \]

It is noted that positive \( \Omega_i \) corresponds to the counterclockwise rotation as viewed facing the spindle nose.

Except for the automatic slide-to-roll ratio sweep, the operator is required to input the contact parameters, including the diameters of the contact pair, the lubricant temperature, the rolling velocity that has to stay between \( u_r^{\text{min}} \) and \( u_r^{\text{max}} \), the normal force, and the operating time. Figure 2.15 shows an example setup for a friction measurement operating under the lubricant temperature of 50 °C, rolling velocity of 5.5 m/s, and normal force of 1095 N within a 5 minute time duration, using the contact
pair defined by Figs. 2.3 and 2.4. The coefficient of friction is computed from measured frictional torque, $T$, according to

$$
\mu = \frac{2T}{dW}
$$

(2.8)

### 2.3.2 Rolling Contact with Block Operating Condition

For the second mode of rolling contact, the operator is allowed to define the total number of blocks, which corresponds to the input of number of stages in Fig. 2.16. Here, a total of 14 blocks are implemented. Except for the diameters of the contact pair and the lubricant supply temperature that is set at 80 °C in Fig. 2.16, the slide-to-roll ratio, rolling velocity, normal force and number of contact cycles need to be defined separately for each of the blocks. Figure 2.16 (a) and (b) shows the first block (stage 1) and last block (stage 14) contact parameters for an example scuffing test, where only the normal force varies with the block. When the scuffing failure occurs, the frictional torque shoots up due to the thermal welding of surfaces that are in relative motion. The tribo-meter will automatically stop, when the measured frictional torque exceeds the preset value of 10 Nm as shown in Fig. 2.17.

For an arbitrary combination of $d_1$, $d_2$ and $SR$, the allowable range of $u_r$ is given by

$$
u_{r,\text{min}} = \frac{\pi}{30} \max \left\{ \frac{d_1\Omega_{\text{min}}}{2 + SR}, \frac{d_2\Omega_{\text{min}}}{2 - SR} \right\}$$

(2.9a)

$$
u_{r,\text{max}} = \frac{\pi}{30} \min \left\{ \frac{d_1\Omega_{\text{max}}}{2 + SR}, \frac{d_2\Omega_{\text{max}}}{2 - SR} \right\}$$

(2.9b)
It is noted the other rolling contact measurements, including contact fatigue (micro and macro pitting) and wear under the duty cycle condition can also be performed using this operating mode.
Fig. 2.14  Control screen showing ready for testing.
Fig. 2.15  Test parameter setting up for friction measurement with continuous SR variation.
Fig. 2.16 Test parameter setting up for staged (block duty cycle) rolling contact (a) stage# 1 (first stage), and (b) stage# 14 (last stage).
Fig. 2.17  Test stop criteria.
3.1 Contact Parameter

3.1.1 Elliptical Hertzian Contact

This study considers the contact formed by a simple cylinder (referred as the roller) and a cylinder with the circular crown applied along the axial direction (referred as the disk), resulting in an elliptical shape of the Hertzian contact zone. To determine the maximum Hertzian pressure, $p_h$, which is illustrated in Fig. 3.1, the formulation introduced in Ref. [28] is adopted as

$$p_h = 1.5 \frac{W}{\pi ab}$$

(3.1)

where $W$ is the normal contact force, and $a$ and $b$ represent the major and minor axes of the contact ellipse (as shown in Fig. 3.1), respectively, whose values are determined according to

$$a = C_a \sqrt{\frac{1.5W}{DE'}}$$

(3.2a)

$$b = C_b \sqrt{\frac{1.5W}{DE'}}$$

(3.2b)
where $E'$ is the reduced elastic modulus of the contact body 1, whose elastic modulus and Poisson’s ratio are $E_1$ and $\nu_1$, and the contact body 2, whose elastic modulus and Poisson’s ratio are $E_2$ and $\nu_2$, as

$$E' = 2 \left[ \frac{(1-\nu_1^2)}{E_1} + \frac{(1-\nu_2^2)}{E_2} \right]^{-1} \quad (3.3a)$$

In this work, the roller and the disk are made from the same material, such that $E_1 = E_2 = E$ and $\nu_1 = \nu_2 = \nu$. Equation (3.3) is then reduced to

$$E' = \frac{E}{(1-\nu^2)} \quad (3.3b)$$

In Eq. (3.2), $D$ is a geometry related parameter and is defined as

$$D = \frac{1}{d_1} + \frac{1}{d_1'} + \frac{1}{d_2} + \frac{1}{d_2'} \quad (3.4)$$

where $d_1'$ and $d_2'$ are the diameters of the circular crown implemented for the roller and the disk, respectively. According to Chapter 2, the contact pair considered in this study has $d_1 = 31.75$ mm, $d_1' = \infty$ (roller has no crown), $d_2 = 57.15$ mm and $d_2' = 152.4$ mm, yielding $D = 0.056$ from Eq. (3.4). The coefficients $C_a$ and $C_b$ in Eq. (3.2) are determined according to Table 3.1 with $\beta$ defined as [27]

$$\beta = \cos^{-1}\left(\frac{D'}{D}\right) \quad (3.5)$$

where

$$D' = \sqrt{\left(\frac{1}{d_1'} - \frac{1}{d_1}\right)^2 + \left(\frac{1}{d_2} - \frac{1}{d_2'}\right)^2 + 2\left(\frac{1}{d_1'} - \frac{1}{d_1}\right)\left(\frac{1}{d_2} - \frac{1}{d_2'}\right)\cos(2\theta)} \quad (3.6)$$
The angle $\theta$ in Eq. (3.6) is the angle between the plane containing $d_1$ and $d_2$ and the plane containing $d'_1$ and $d'_2$. For the contact setup of this study, $\theta = \frac{\pi}{2}$, and thus, Eq. (3.6) yields $D' = 0.042$. Utilizing Eq. (3.5), it is found $\beta = 40.201^\circ$, and $C_a$ and $C_b$ are then determined from Table 3.1 to be $C_a = 2.128$ and $C_b = 0.569$. From Eq. (3.2), the aspect ratio of the contact is determined as $k = \frac{a}{b} = 3.7$.

### 3.1.2 Film Thickness

As two mechanical elements roll against each other, the supplied lubricant is entrained into the contact zone by the moving surfaces, establishing a hydrodynamic lubrication film between the surfaces to prevent the direct metal-to-metal contact. To estimate the thickness of this lubrication film, Hamrock and Dowson developed the smooth EHL simulation based regression formulae for both the central (denoted as $h_{cen}$) and the minimum (denoted as $h_{min}$) film thickness of point contacts as [28]

$$h_{cen}^N = 2.69U^{0.67}G^{0.53}L^{-0.067} \left(1 - 0.61e^{-0.73k}\right)r' \quad (3.7)$$

$$h_{min}^N = 3.63U^{0.68}G^{0.49}L^{-0.073} \left(1 - e^{-0.68k}\right)r' \quad (3.8)$$

where the dimensionless speed parameter, lubricant parameter, and load parameter are defined as $U = \frac{\eta_0\mu_r}{E'r'}$, $G = \alpha E'$, and $L = \frac{W}{E'r'^2}$, respectively. In these expressions, $\eta_0$ is the lubricant ambient viscosity, $\alpha$ is the lubricant viscosity-pressure coefficient, and $r'$ represents the reduced contact radius that has the form of $r' = \frac{d_1d_2}{2(d_1 + d_2)}$. 

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Fig. 3.1 Illustration of the pressure distribution of an elliptical Hertzian contact.
Table 3.1  Coefficients for Hertzian Contact Width [27]

<table>
<thead>
<tr>
<th>β</th>
<th>$C_a$</th>
<th>$C_b$</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>$\infty$</td>
<td>0</td>
</tr>
<tr>
<td>0.5</td>
<td>61.4</td>
<td>0.1018</td>
</tr>
<tr>
<td>1</td>
<td>36.8</td>
<td>0.1314</td>
</tr>
<tr>
<td>1.5</td>
<td>27.48</td>
<td>0.1522</td>
</tr>
<tr>
<td>2</td>
<td>22.26</td>
<td>0.1691</td>
</tr>
<tr>
<td>3</td>
<td>16.5</td>
<td>0.1964</td>
</tr>
<tr>
<td>4</td>
<td>13.31</td>
<td>0.2188</td>
</tr>
<tr>
<td>6</td>
<td>9.79</td>
<td>0.2552</td>
</tr>
<tr>
<td>8</td>
<td>7.86</td>
<td>0.285</td>
</tr>
<tr>
<td>10</td>
<td>6.604</td>
<td>0.3112</td>
</tr>
<tr>
<td>20</td>
<td>3.778</td>
<td>0.408</td>
</tr>
<tr>
<td>30</td>
<td>2.731</td>
<td>0.493</td>
</tr>
<tr>
<td>35</td>
<td>2.397</td>
<td>0.53</td>
</tr>
<tr>
<td>40</td>
<td>2.136</td>
<td>0.567</td>
</tr>
<tr>
<td>45</td>
<td>1.926</td>
<td>0.604</td>
</tr>
<tr>
<td>50</td>
<td>1.754</td>
<td>0.641</td>
</tr>
<tr>
<td>55</td>
<td>1.611</td>
<td>0.678</td>
</tr>
<tr>
<td>60</td>
<td>1.486</td>
<td>0.717</td>
</tr>
<tr>
<td>65</td>
<td>1.378</td>
<td>0.759</td>
</tr>
<tr>
<td>70</td>
<td>1.284</td>
<td>0.802</td>
</tr>
<tr>
<td>75</td>
<td>1.202</td>
<td>0.846</td>
</tr>
<tr>
<td>80</td>
<td>1.128</td>
<td>0.893</td>
</tr>
<tr>
<td>85</td>
<td>1.061</td>
<td>0.944</td>
</tr>
<tr>
<td>90</td>
<td>1</td>
<td>1</td>
</tr>
</tbody>
</table>
The film thickness formulae of Eqs. (3.7) and (3.8) were arrived under the pure rolling assumption, i.e. \( u_s = 0 \), where the fluid is considered to be Newtonian. In the current study, however, the roller-disk contact pair operates under the combined rolling and sliding condition. The non-Newtonian behavior of the lubricant when the shear rate is high introduces the shear-thinning effect, such that the film thickness reduces. To include the shear-thinning in the lubrication film thickness estimation, the central and minimum film thickness correction factors proposed by Jang and Khonsari [29] are adopted as

\[
\phi_{\text{cen}} = \left(1 + 0.75 \left[ \left(1 + 4|SR| \right) \Gamma \right]^{1.6} \right) \\
\phi_{\text{min}} = \left(1 + 0.83 \left[ \left(1 + 4|SR| \right) \Gamma \right]^{1.6} \right)
\]

(3.9)

(3.10)

where \( n \) is the power-law exponent used in the lubricant non-Newtonian description, and \( \Gamma \) is the Weissenberg number that is defined as

\[
\Gamma = \frac{n\mu_{fr}}{h_{cen}^N G_{cr}}
\]

(3.11)

with \( G_{cr} \) representing the critical stress of the lubricant. The central and minimum film thicknesses under the non-Newtonian condition are then assessed through

\[
h_{cen} = \frac{h_{cen}^N}{\phi_{cen}}
\]

(3.12)

\[
h_{min} = \frac{h_{min}^N}{\phi_{min}}
\]

(3.13)

For practical engineering applications, the surfaces of the contact components are not perfectly smooth. The roughness profiles such as those shown in Figs 2.8 - 2.13...
significantly interrupt the elastohydrodynamic lubrication. To assess the lubrication performance, the specific film thickness (also commonly referred as the lambda ratio) that is defined as the ratio of the minimum film thickness to the composite surface roughness amplitude (denoted as \( R_{q}^{c} \)) has been widely used. Mathematically, the specific film thickness, \( \lambda \), has the expression of

\[
\lambda = \frac{h_{\text{min}}}{R_{q}^{c}}
\]

With the RMS surface roughness amplitudes of surface 1, \( R_{q1} \), and surface 2, \( R_{q2} \), given, the composite surface roughness amplitude is determined as

\[
R_{q}^{c} = \sqrt{R_{q1}^2 + R_{q2}^2}
\]

### 3.2 Test Matrix

This study investigates the friction and scuffing resistance performance of two new materials A and B in comparison to the baseline material of AISI 5120 alloy. The friction measurements are performed according to the test matrix of Table 3.2, where two loading levels of \( p_h = 1.2 \) and 1.7 GPa, and two rolling velocity levels of \( u_r = 3.0 \) and 5.5 m/s are implemented for each material. The continuous sliding sweep is set to be within the range of \( 0 < SR < 1 \) for all the tests. For the friction experiments, the lubricant supply temperature is controlled at 50 °C, where the lubricant viscosity and pressure-viscosity coefficient are \( \eta_0 = 1.502 \times 10^{-2} \) Pas and \( \alpha = 1.576 \times 10^{-8} \) Pa\(^{-1}\). The measured friction distributions within the sliding range of the two new materials are compared with the baseline under the different loading and speed conditions in the next section.
As for the experimental assessment of the scuffing resistance, the test matrix of Table 3.3 is considered, where two speed levels of $u_r = 4.0$ and $8.0$ m/s are implemented. The slide-to-roll ratio is fixed at $SR = 1$, owing to the fact that the scuffing failure usually occurs under the high sliding condition. The normal load applied to the contact pair is set to increase in a stepwise manner from $p_h = 1.2$ to $2.5$ GPa, with the Hertzian pressure increment of $\Delta p_h = 0.1$ GPa. Figure 3.2 illustrates both the normal force and the Hertzian pressure variations with the operating time for one complete scuffing test. Each loading stage lasts for 2 minutes. For the scuffing failure investigation, the lubricant supply temperature is kept constant at $80 \, ^\circ\text{C}$, where the lubricant viscosity and pressure-viscosity coefficient are $\eta_0 = 6.196 \times 10^{-2}$ Pas and $\alpha = 1.335 \times 10^{-8}$ Pa$^{-1}$. The adoption of this elevated temperature is for the acceleration of the scuffing failure. When the scuffing failure occurs, the surface welding due to the extreme surface temperatures introduces the significant jump in the friction. The tribometer automatically stops as such jump is detected by the torque meter.

3.3 Results and Discussions

3.3.1 Friction Measurement

The friction performances of the baseline material, and the new materials $A$ and $B$ are compared within the slide-to-roll ratio range of $0 < SR < 1$ under the four load and speed combinations of F1 to F4 as defined in Table 3.2 in Figs. 3.3 to 3.6. The minimum film thickness calculated according to Eq. (3.13) is also included in these figures. The values are $h_{\text{min}} = 0.22, 0.33, 0.20, \text{and } 0.31 \, \mu\text{m}$ for F1, F2, F3, and F4, respectively. The shear-thinning correction factor of Eq. (3.10) is found to be varying
Table 3.2  Test matrix for friction measurement.

<table>
<thead>
<tr>
<th>Test SN#</th>
<th>$p_h$ [GPa]</th>
<th>$u_r$ [m/s]</th>
</tr>
</thead>
<tbody>
<tr>
<td>F1</td>
<td>1.2</td>
<td>3.0</td>
</tr>
<tr>
<td>F2</td>
<td>1.2</td>
<td>5.5</td>
</tr>
<tr>
<td>F3</td>
<td>1.7</td>
<td>3.0</td>
</tr>
<tr>
<td>F4</td>
<td>1.7</td>
<td>5.5</td>
</tr>
</tbody>
</table>

Table 3.3  Test matrix for scuffing failure measurement.

<table>
<thead>
<tr>
<th>Test SN#</th>
<th>$u_r$ [m/s]</th>
<th>SR</th>
</tr>
</thead>
<tbody>
<tr>
<td>S1</td>
<td>4</td>
<td>1.0</td>
</tr>
<tr>
<td>S2</td>
<td>8</td>
<td>1.0</td>
</tr>
</tbody>
</table>
Fig. 3.2  Normal force and maximum Hertzian pressure variations with time for scuffing tests.
very limitedly, given the lubricant properties of \( G_{cr} = 1 \text{ MPa} \) and \( n = 0.6 \), such that a straight line is observed for \( h_{\text{min}} \). Utilizing the RMS surface roughness amplitudes of the roller and disk surfaces of \( R_{q1} = R_{q2} = 0.16 \mu m \), the composite roughness amplitude is \( R_{q}^{c} = 0.23 \mu m \). The specific film thicknesses for these four cases are then found as \( \lambda = 0.96, 1.44, 0.87, \) and \( 1.35 \), respectively. Within this range of the specific film thickness, the roller-disk contact operates under the mixed EHL condition, i.e. the asperity contacts (metal-to-metal contacts) and the pressurized hydrodynamic film coexist and share the normal load.

Examining Figs. 3.3 to 3.6, it is observed that the friction coefficients of the three materials are generally comparable with each other under all the four operating conditions. In each of these figures, the friction coefficient, \( \mu \), is seen to first increase with \( SR \) sharply within \( 0 < SR < 0.2 \), and then gradually decrease when \( SR \) further increases. The increase of \( \mu \) with \( SR \) is owing to the sliding velocity increase, and the decrease of \( \mu \) that follows is because of the thermal effect, i.e. the lubricant viscosity decrease caused by the lubricant temperature rise under the high sliding condition of \( SR > 0.2 \).

Comparing the tests of F1 and F2, no significant difference in the magnitude of \( \mu \) is found within \( 0 < SR < 0.6 \) when the rolling velocity is increased from \( 3.0 \text{ m/s} \) to \( 5.5 \text{ m/s} \). In view of the viscous shear stress, \( q \), acting on surface 1, which has the form of [30]

\[
q = -\frac{h}{2} \frac{\partial p}{\partial x} - \frac{\eta^* u_s}{h} \tag{3.13}
\]

where \( p \) and \( h \) represent the hydrodynamic pressure and film thickness, and \( \eta^* \) is the effective viscosity, it can be explained that the impact of the sliding velocity increase
on friction (increase $\mu$) is offset by the impact of the film thickness increase on friction (decrease $\mu$). In the sliding range of $0.6 < SR < 1$, however, the friction under the higher rolling velocity is seen to be much lower than that under the lower rolling velocity. Borrowing the formula of the viscous frictional heat flux, $Q$, from Ref. [30] as

$$Q = \frac{h^3}{12\eta^*} \left( \frac{\partial p}{\partial x} \right)^2 + \frac{\eta^* u_x^2}{h}$$

(3.14)

the exponent of 2 of the sliding velocity in Eq. (3.14) introduces larger frictional heat under the higher rolling velocity condition. Therefore, the lubricant temperature increase of F2 becomes larger than F1, and the lubricant viscosity of F2 becomes smaller than that of F1, resulting in the smaller friction coefficient. As for F3 and F4, this thermal behavior within $0.6 < SR < 1$ is not evident. It can be due to the reason that the pressure gradient induced rolling friction [1$^{st}$ term of Eq. (3.13)] and rolling frictional heat [1$^{st}$ term of Eq. (3.14)] becomes relatively more dominant when $p_h$ is increased from 1.2 to 1.7 GPa. Regarding the load effect on friction, the comparison between Fig 3.3 and Fig. 3.5 and the comparison between Fig. 3.4 and 3.6 shows the load increase doesn’t alter the friction evidently within the operating condition range considered.

### 3.3.2 Scuffing Measurement

The scuffing resistance of the baseline material and the new materials $A$ and $B$ under the operating condition of $u_r = 4$ m/s and $SR = 1$ are shown in Figs. 3.7 to 3.9. For the baseline material, the roller-disk contact pair survives the entire loading range without failure. No jump in the frictional torque is observed. However for materials $A$
and $B$, both fail at the loading stage of $p_h = 1.8$ GPa, where the frictional torque is seen to shoots up beyond 10 Nm. When the rolling velocity is increased to $u_r = 8$ m/s while keeping the slide-to-roll ratio unchanged, the baseline material failed at the last loading stage of $p_h = 2.5$ GPa as shown in Fig. 3.10. The materials $A$ and $B$ fail at the loading stages of $p_h = 2.1$ and 2.0 GPa, respectively, as shown in Figs. 3.11 and 3.12. Under both the operating conditions, the new materials are seen to have the lower scuffing resistance (fail at a smaller loading stage) in comparison to the baseline material. The new materials $A$ and $B$ are seen to have comparable scuffing resistance.

Figure 3.13 shows the $100\times$, $200\times$, and $500\times$ magnification micro-images of the roller and disk surfaces after the scuffing test that is associated with Fig. 3.7, where no scuffing failure is observed. For material $A$, the micro-images before and after testing are compared in Fig. 3.14 for the roller surface and in Fig. 3.15 for the disk surface. It is observed the original surface roughness texture (horizontal direction roughness lays) is completely destroyed for both the roller and the disk surfaces. Similar observations are found for material $B$ as shown in Figs. 3.16 and 3.17. When the rolling velocity is increased to 8 m/s, the failed surface images of the baseline material is shown in Figs. 3.18 and 3.19 for the roller and the disk, respectively. It is very interesting to see that the failed surfaces at the higher speed possess the feature of micro-pits, which are missing under the lower rolling velocity of 4 m/s. For the materials $A$ (Figs. 3.20 and 3.21) and $B$ (Figs. 3.22 and 3.23), the micro-pits are also found to be prevalent on the failed surfaces.
Fig. 3.3 Comparison of friction coefficient between the three materials considered under $p_h = 1.2 \text{ GPa}$, $u_r = 3.0 \text{ m/s}$, and $0 < SR < 1$. The red line represents the corresponding film thickness.
Fig. 3.4  Comparison of friction coefficient between the three materials considered under $p_h = 1.2$ GPa, $u_r = 5.5$ m/s, and $0 < SR < 1$. The red line represents the corresponding film thickness.
Fig. 3.5 Comparison of friction coefficient between the three materials considered under $p_h = 1.7$ GPa, $u_r = 3.0$ m/s, and $0 < SR < 1$. The red line represents the corresponding film thickness.
Fig. 3.6 Comparison of friction coefficient between the three materials considered under $p_h = 1.7$ GPa, $u_r = 5.5$ m/s, and $0 < SR < 1$. The red line represents the corresponding film thickness.
Fig. 3.7 The variations of the frictional torque and the corresponding normal force with the number of contact cycles for the scuffing test of baseline AISI 5120 under the operating condition of SN# S1 defined in Table 3.3.
Fig. 3.8  The variations of the frictional torque and the corresponding normal force with the number of contact cycles for the scuffing test of new material A under the operating condition of SN# S1 defined in Table 3.3.
Fig. 3.9 The variations of the frictional torque and the corresponding normal force with the number of contact cycles for the scuffing test of new material B under the operating condition of SN# S1 defined in Table 3.3.
Fig. 3.10  The variations of the frictional torque and the corresponding normal force with the number of contact cycles for the scuffing test of baseline AISI 5120 under the operating condition of SN# S2 defined in Table 3.3.
Fig. 3.11  The variations of the frictional torque and the corresponding normal force with the number of contact cycles for the scuffing test of new material A under the operating condition of SN# S2 defined in Table 3.3.
Fig. 3.12 The variations of the frictional torque and the corresponding normal force with the number of contact cycles for the scuffing test of new material B under the operating condition of SN# S2 defined in Table 3.3.
Fig. 3.13  Microscope images of the roller (left column) and the disk (right column) surfaces at the magnifications of (a) ×100, (b) ×200, and (c) ×500 for the scuffing test of baseline AISI 5120 under the operating condition of SN# S1 defined in Table 3.3.
Fig. 3.14  Microscope images of the roller surfaces at the magnifications of (a)×100, (b)×200, and (c) ×500 for the scuffing test of new material A under the operating condition of SN# S1 defined in Table 3.3. Left column shows the surface before testing. Right column shows the surface after testing.
Fig. 3.15 Microscope images of the disk surfaces at the magnifications of (a) $\times100$, (b) $\times200$, and (c) $\times500$ for the scuffing test of new material A under the operating condition of SN# S1 defined in Table 3.3. Left column shows the surface before testing. Right column shows the surface after testing.
Fig. 3.16  Microscope images of the roller surfaces at the magnifications of (a) $\times 100$, (b) $\times 200$, and (c) $\times 500$ for the scuffing test of new material B under the operating condition of SN# S1 defined in Table 3.3. Left column shows the surface before testing. Right column shows the surface after testing.
Fig. 3.17 Microscope images of the disk surfaces at the magnifications of (a) ×100, (b) ×200, and (c) ×500 for the scuffing test of new material B under the operating condition of SN# S1 defined in Table 3.3. Left column shows the surface before testing. Right column shows the surface after testing.
Fig. 3.18  Microscope images of the roller surfaces at the magnifications of (a)×100, (b)×200, and (c) ×500 for the scuffing test of baseline AISI 5120 under the operating condition of SN# S2 defined in Table 3.3. Left column shows the surface before testing. Right column shows the surface after testing.
Fig. 3.19  Microscope images of the disk surfaces at the magnifications of (a) $\times 100$, (b) $\times 200$, and (c) $\times 500$ for the scuffing test of baseline AISI 5120 under the operating condition of SN# S2 defined in Table 3.3. Left column shows the surface before testing. Right column shows the surface after testing.
Fig. 3.20  Microscope images of the roller surfaces at the magnifications of (a)×100, (b)×200, and (c) ×500 for the scuffing test of new material A under the operating condition of SN# S2 defined in Table 3.3. Left column shows the surface before testing. Right column shows the surface after testing.
Fig. 3.21  Microscope images of the disk surfaces at the magnifications of (a) ×100, (b) ×200, and (c) ×500 for the scuffing test of new material A under the operating condition of SN# S2 defined in Table 3.3. Left column shows the surface before testing. Right column shows the surface after testing.
Fig. 3.22 Microscope images of the roller surfaces at the magnifications of (a) $\times 100$, (b) $\times 200$, and (c) $\times 500$ for the scuffing test of new material $B$ under the operating condition of SN# S2 defined in Table 3.3. Left column shows the surface before testing. Right column shows the surface after testing.
Fig. 3.23 Microscope images of the disk surfaces at the magnifications of (a) ×100, (b) ×200, and (c) ×500 for the scuffing test of new material B under the operating condition of SN# S2 defined in Table 3.3. Left column shows the surface before testing. Right column shows the surface after testing.
CHAPTER 4

SUMMARY AND CONCLUSIONS

4.1 Summary

This experimental study investigated the friction and scuffing resistance performance of two new materials \( A \) and \( B \) in comparison to the baseline material of AISI 5120 alloy, utilizing a two-disk contact set-up. The roller and disk specimen surfaces were finished by grinding that was followed by a polishing process to arrive at the RMS surface roughness amplitude of 0.16 \( \mu \text{m} \), which is common for aerospace applications. The turbine lubrication fluid Mil-PRF-23699 was used as the lubricant whose temperature was controlled at 50 °C and 80 °C for the friction and scuffing experiments, respectively. The higher lubricant temperature for the scuffing test is for the purpose of failure acceleration. The friction tests were carried out under two loading levels and two speed levels to show any impact of the operating condition on the friction coefficient. These tests were run with the continuous variation of the slide-to-roll ratio from 0 to 1, while keeping the other contact parameters constant. For the scuffing tests, the slide-to-roll ratio was selected to be 1, owing to the fact that this extreme surface temperature induced failure commonly occurs under the very high sliding condition.
Two rolling velocity levels were considered for the scuffing failure. During one scuffing test, the rolling velocity and the slide-to-roll ratio was fixed, while the normal load was increased in a stepwise way. For each of the loading stage, the roller-disk contact pair operated for two minutes. If scuffing failure takes place at a certain loading stage, the surface welding caused by the high surface temperatures introduces significant jump in the surface friction, and the two-disk tribo-meter stops when such a signal is detected. That loading stage of failure represents the scuffing resistance of the material under the corresponding rolling and sliding condition. A smaller failure load points to a lower scuffing resistance, and a larger failure load indicates a higher scuffing resistance.

Before each test, a run-in procedure of two hours was implemented to break in the surface roughness. During this run-in process, the roller-disk pair was run under the Hertzian pressure of 0.8 GPa and the same surface velocities as those specified in the friction or scuffing tests. After the run-in stage, the specimens were brought to The Ohio State University for the inspection of the surfaces by using a high power digital micro-scope, the measurements of the surface roughness profiles using a 2D surface roughness profiler. Following these inspections, the friction and scuffing tests were then performed according to the test matrix of Chapter 3. The measured friction coefficients of the two new materials A and B were compared to those of the baseline material under different operating condition combinations to show limited differences. By comparing the scuffing failure loads between the new materials and the baseline material under two rolling velocity levels, it was shown the new materials actually had the lower scuffing resistance performance. Between materials A and B, their scuffing resistance was shown to be comparable.
4.2 Conclusions and Recommendations for Future Work

A total of four friction tests corresponding to four different operating speed and load combinations were carried out for each of the materials. These operating ranges led to the specific film thickness to be within 0.87 and 1.44, indicating a mixed elastohydrodynamic condition, which is prevalent for many rolling contact machine elements. The friction coefficient variations with the slide-to-roll ratio were obtained to show typical trend, i.e. the friction coefficient first increased with the sliding within a relatively low slide-to-roll ratio range due to the increase in the sliding velocity meanwhile keeping the rolling velocity unchanged, and then decreased gradually with the sliding in a relatively high slide-to-roll ratio range due to the thermal effect. The friction coefficients were compared between the different materials, showing generally comparable performance under all the four operating conditions. It is thus concluded the new materials does not have a tangible impact on friction reduction.

The scuffing performance of the new materials were examined under two rolling velocity levels. A total of six tests were carried out for the three materials considered. Under the lower rolling velocity, the baseline material survived the entire loading range from 1.2 GPa to 2.5 GPa without any signs of scuffing on the surface. However, the new materials A and B were observed to fail in the form of scuffing at the same stage load of 1.8 GPa maximum Hertzian pressure. Comparing the before and after testing micro-images of the 100×, 200×, and 500× magnifications, it was shown the surface roughness lays were completely destructed by the thermal welding of scuffing. When the rolling velocity was increased to a higher level, the baseline material failed at the last loading stage of 2.5 GPa maximum Hertzian pressure. The materials A and B failed
at the loading stages of $p_h = 2.1$ and $2.0$ GPa, respectively. It was very interesting to see that the failed surfaces at the higher speed possess the feature of micro-pits, which were missing under the lower rolling velocity. From the experimental observations, it thus can be concluded that the new materials failed to improve the scuffing performance under the operating conditions considered.

In view of the facts that the scuffing performance is dependent on the combination of the lubricant and the solid surface material, and is largely affected by the roughness lay direction, it is recommended for the future work to

- Use several different lubricants in the friction and scuffing experiments to observe the tribological performances.
- Change the roughness lay direction from the direction that is parallel to the surface velocities to the direction that is normal to the surface velocities, since the surface roughness lay direction is normal to the velocity direction for gear contacts.
- Extend the operating speed to a higher range, given the growing power density of modern automotive power transmission systems, which demands the continuous increase of the operating velocities of gear and bearing components.
References


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