SURFACE DEGRADATION ANALYSIS OF ROLLER BEARING

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SURFACE DEGRADATION ANALYSIS OF ROLLER BEARING

by

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LIST OF SYMBOLS

Symbol	Description	Unit
dr	Roller diameter	mm
Di	Inner bore diameter	mm
Do	Outer ring outside diameter	mm
dm	Pitch diameter	mm
B	Width	mm
Ra	Average roughness	µm
Wa	Average waviness	µm
Rq	Root mean square roughness	µm
Z	Numerical value of roundness (roundness error)	-
n	Number of rollers	-
fr	Relative speed between inner and outer races	rev/s
U	Speed parameter	-
G	Material parameter	-
W	Load parameter	-
λ	Film thickness/roughness	-

ANALISIS KEMEROSOTAN PERMUKAAN GALAS PENGGELEK

ABSTRAK

Analisis kemerosotan permukaan dan pengesanan kecacatan pada galas penggelek adalah penting untuk meramal kegagalan mesin. Kajian ini dijalankan dalam dua keadaan iaitu melalui pelinciran gris dan tanpa pelinciran. Analisis difokuskan kepada kemerosotan permukaan gegelang dalam dan penggelek kerana mengalami kadar tegasan yang lebih tinggi. Kemerosotan permukaan galas penggelek secara semulajadi telah dipantau dari permulaan eksperimen sehingga rosak. Ketebalan lapisan pelinciran diperolehi dengan menggunakan persamaan *Hamrock-Dowson* dan menunjukkan galas beroperasi dalam pelinciran separa *elasto-hydrodynamic* (EHL) dengan nisbah ketebalan lapisan pelinciran kepada kekasaran permukaan dari 0.9 kepada 3.65. Galas yang beroperasi tanpa pelincir rosak selepas 20 minit manakala galas dengan pelinciran gris beroperasi sehingga rosak pada 6600 minit. Pemerhatian daripada mikroskop menunjukkan permukaan gegelang dalam mengalami proses pelicinan dimana kekasaran permukaan menurun semasa fasa permulaan ketika λ = 3.65 sehingga 2.3, disusuli dengan proses pengasaran pada fasa stabil ketika λ = 2.3 sehingga 0.9. Ketika galas rosak, nilai λ = 0.9 dan mempunyai paras getaran yang tinggi. Peningkatan paras getaran bertambah dengan cepat pada nilai kritikal λ = 1.6. Ini menunjukkan kadar getaran berkadar langsung dengan kemerosotan permukaan. Tanpa pelinciran, proses in berlaku dengan lebih cepat. Pengukuran kerataan dan kebulatan permukaan menunjukkan perubahan bentuk galas paling banyak berlaku pada gegelang dalam. Analisis bersampul pada penghujung ujikaji menunjukkan kerosakan pada galas dikenalpasti berlaku pada gegelang dalam melalui puncak frekuensi kecacatan gegelang dalam.

SURFACE DEGRADATION ANALYSIS OF ROLLER BEARING

ABSTRACT

Surface degradation analysis and defects detection of roller bearing are crucial for the prediction of the impending failure in machineries. In this work, two conditions of grease lubrication and unlubricated condition are investigated. The analysis is focused on the surface degradation of the inner race and the rolling element since these surfaces suffer from the higher contact stress level. The natural surface degradation of the roller bearing is monitored from the beginning of the experiment until damage. The lubrication film thickness is determined by the Hamrock-Dowson equation which showed that the grease lubricated bearing operated under the elastro-hydrodynamic lubrication, with the ratio of lubrication film thickness to the surface roughness of λ in the range of 0.9 to 3.65. The unlubricated bearing was damaged after 20 minutes whereas the grease lubricated bearing continued to operate for 6600 minutes. The observation under microscope showed that the surface underwent smoothening process where the surface roughness decreases during the running-in state with the λ = 3.65 to 2.3 followed by roughening at the steady state where the surface roughness increases, with λ = 2.3 to 0.9. At damage, the value of λ = 0.9 can be associated with the high level of the bearing vibration. The increase of vibration level becomes rapid at the critical value of λ = 1.6. As such the overall vibration level of the bearing can be related to the surface degradation. For unlubricated condition, these processes took place faster. Both waviness and roundness measurement showed higher form change of the inner race. Envelope analysis at the end of experiment showed that the bearing damaged can be traced at the inner race with the clear peak of the inner race bearing defect frequency.

CHAPTER 1

INTRODUCTION

1.0 Background

Rolling element bearings have widespread applications particularly in industrial machineries. These bearings are considered as critical mechanical components in industrial applications and the failure to detect the presence of defects may lead to catastrophic failure of the machinery. Over the years, researchers have put a lot of effort to investigate the mechanism of bearing failure including detecting the impending failure and the characteristics of the failed bearing. According to FAG Technical Publication (1997), 80 % of bearing failures are caused by lubricant failure. The detail is shown in figure 1.1.

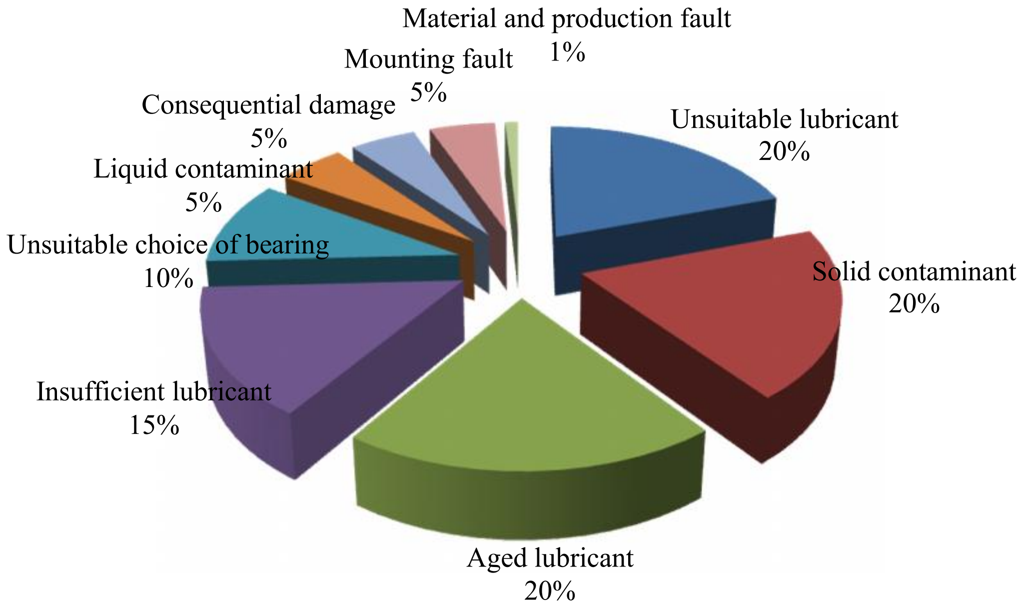


Figure 1.1: Sources of bearing failures (FAG, 1997)

Various techniques have been widely used to analyze the vibration signal of the bearing to acquire the best method in detecting bearing failure. The vibration level of roller bearing reflects the manufacturing inaccuracies, installation and mounting, operating speed, supported load, lubrication and defects (Patil et al., 2008). The operation of rolling bearing is influenced by friction, wear, lubrication, fluid dynamics, material properties, and contact mechanics. Degradation of the rolling surfaces occurs due to plastic deformation, rolling contact wear and fatigue (Halme and Anderson, 2009).

Rolling bearing performance is indicated by the smooth and quiet running of the bearing which is attributed to high efficiency, low friction, low sensitivity and high reliability even when operating under severe condition applications (Ebert, 2010). The ideal bearing operation can be achieved by good design and sufficient lubrication. The failure of a long term operation bearing is usually caused by the fatigue of the surfaces which are subjected to continuous rolling contact. Failure results in the changes of the surfaces texture and geometrical form.

Identification of the bearing surface condition after rolling contact is essential for the prediction of impending failure. Hence the surface roughness is evaluated periodically in this study. At the same time, the lubrication regime produced by grease lubricated film thickness is also determined. For comparison, the roller bearing is also operated in unlubricated condition. The change in the surface roughness and lubrication condition are monitored simultaneously with the vibration level. The main interest in this study is the surface degradation of the inner race and the rolling element and how they are affected by the lubrication regime and how these are indicated by the vibration level.

1.1 The cylindrical roller bearing

The roller bearing is designed to permit relative motion between two machine parts, usually between a rotating shaft and a fixed frame supporting the applied load. The rolling element bearing basically consists of inner raceway, outer raceway, cage and rolling elements. The rolling elements are generally in the form of ball or cylindrical roller. In this study, the analysis is limited to the roller bearing since it has high radial load carrying capacity. The load placed on the bearing is supported by the contact surface between the inner raceway and the rollers. In this type of bearing, the roller is in the form of a cylinder, so the contact between the inner and outer race is a line, not a point. Therefore, the surface characteristic along the line contact is one of the interests in this study. This type of contact distributes the load over a larger area than a ball bearing, hence allowing the bearing to handle much greater loads. The inner raceway is normally fitted on the rotating shaft, whilst the outer raceway is mounted on a stationary housing. The cylindrical roller bearing used in this study is shown in figure 1.2.

Figure 1.2: Cylindrical roller bearing

The rolling bearings have several advantages as compared to other type of bearings (Harris, 1984). This bearing has lower friction torque and lower friction power loss than conventional hydrodynamic bearing. A low friction level is achieved when the starting friction is only slightly larger than kinetic friction. This type of bearing also has lower deflection to load fluctuation and require only a small quantity of lubricant for satisfactory operation as compared to hydrodynamic bearing. The changes in load, speed and operating temperature generate only minimal effect that is within reasonable limit. Most rolling bearings are able to support radial, thrust, or combinations of these loads. The impending failure of this bearing can be predicted by an increase in the operational noise level. However, this bearing also has a few disadvantages including higher cost and higher noise level during operation than hydrodynamic bearings. Its application at high speeds and loads may induce a high friction, and its operation is limited by surface contact fatigue, centrifugal force, and brinelling effects. In most cases, the entire unit has to be replaced when worn or failed.

1.2 Problem statement

Lubrication plays an important role to form a layer of film to separate the sliding and rolling surfaces and to sustain the applied load. Insufficient lubrication normally results in short bearing operation life. The effect of insufficient lubrication can influence the surface characteristic and the vibration level. There is a need to identify the relationship between surface degradation and vibration behaviour of the roller bearing under normal operating condition without defect seeding.

1.3 Research objective

The hypothesis of this study is the degradation of the bearing surface under normal operating condition will influence the vibration level of the bearing and also the lubrication film of the contact interface as the degradation progresses. It is the relationship between the surface degradation, lubrication and vibration level of the rolling bearing that this study seeks to establish. The objectives of this research are:

- To investigate the surface degradation of the roller bearing in operation and its relationship with the vibration level
- To determine the lubrication condition during the surface degradation

1.4 Research scope

The research looked into two conditions of grease lubrication and unlubricated condition. The study of the surface degradation is limited to two components; these are the inner race and the rolling element since they are subjected to moving rolling contact. The degradation is monitored from the beginning of the experiment until damaged. The analysis of bearing form change is performed by surface waviness measurement of the inner race and the roundness measurement of both the inner race and the roller at the beginning and the end of the experiment. The vibration study is divided into vibration level monitoring and defect detection by envelope analysis. The lubrication analysis is performed by applying Hamrock-Dowson equation to acquire minimum film thickness and the relationship between roughness and lubrication film thickness (λ).

CHAPTER 2

LITERATURE REVIEW

2.0 General overview

The surface characterization of the rolling bearing is one of the techniques to identify the bearing condition. Early detection of the surface failure will help to predict the bearing failure which can prevent the unplanned failure of the machinery or the system. Sufficient lubrication is required in order to ensure long bearing operation life. The detection of bearing failure is normally carried out by vibration level monitoring. This chapter reviews the work on rolling bearing surface characterization, vibration monitoring and lubrication method.

2.1 Surface characterization

The rolling bearing defects can be categorized as either distributed or localized defects. According to Sunnersjo, (1985) the distributed defects are produced by improper installation, surface manufacturing error or abrasive wear. The types of defects include surface roughness, waviness, race misalignment and off-size rolling elements (Tallian and Gustafsson, 1965). The localized defects types include crack, pit, spall and flake on the bearing surfaces which is normally produced after certain period of bearing operation. The local defects on the contacting surface will cause an abrupt change in the contact stress; subsequently generating pulse to excite the vibration of the bearing.

Juhasz and Opoczky (1990) claimed that the contacts between surface micro asperities of the rolling elements, raceways and rolling element are one of the sources of vibration. The other contributing sources are the deformation of contaminant particles and the crushing between the rolling elements and the raceways.

Fernandes (1997) discussed the development of surface defects of rolling element bearing due to surface contact fatigue. Initially, the small cracks are formed below the contact surface, subsequently propagate to form micro pits. The micro pits will progress to form spalling and flaking due to continuous rolling contact and stress concentration. Commonly, the flaking is observed on the stationary ring first since every time the rolling elements pass over it, the surface is subjected to maximum stress.

3D surface analysis on the wear of the spherical roller thrust bearing has been studied by Olofsson and Bjorklund (1998). An area analysis of surface texture parameter (Sq) was employed. The parameter basically worked similar to the Ra but incorporating area factor as shown in figure 2.1 where the graph has a bath tub shape. Due to the action during running in, the bearing surface was smoothened by the flattened asperities because of the plastic deformation. After that, the deformation mode becomes more elastic and the wear mechanism can be attributed to the two body abrasive wear (rolling element and raceway). For a long term test, the three body abrasion (including wear particles) is initiated, subsequently increasing the amplitude of the contact surface.

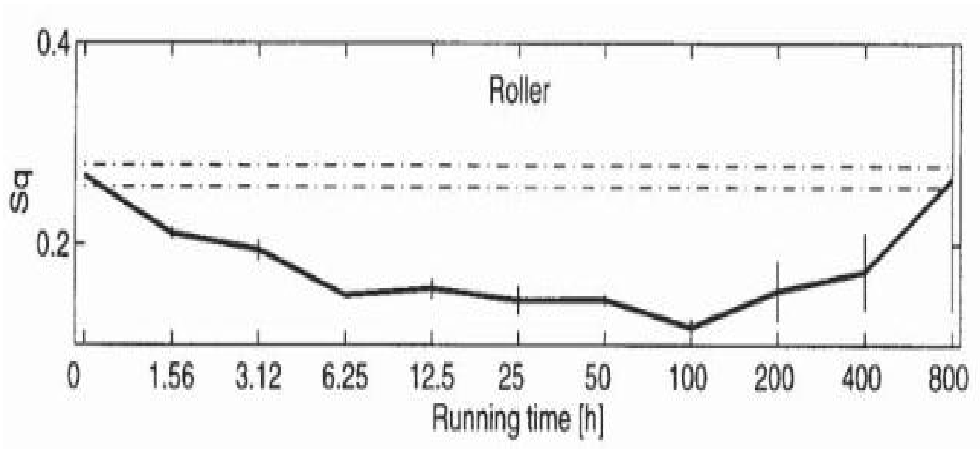


Figure 2.1: RMS deviation of Sq versus time (Olofsson and Bjorklund, 1998)

Kuhnell (2004) set a guideline for the bearing failure signature identification by categorizing them into three main types of defects, of surface distress, pitting and spalling. The differentiation and size of the defects are shown in figure 2.2 where a surface distress is referred to the smooth surface formed by plastic deformation of asperities with a depth less than 10 µm. Pitting formed on the contact surfaces with a depth of the thickness of the work-hardened layer of approximately 10 µm. Spalling is referred to a deeper surface ploughing with a depth of 20 to100 µm.

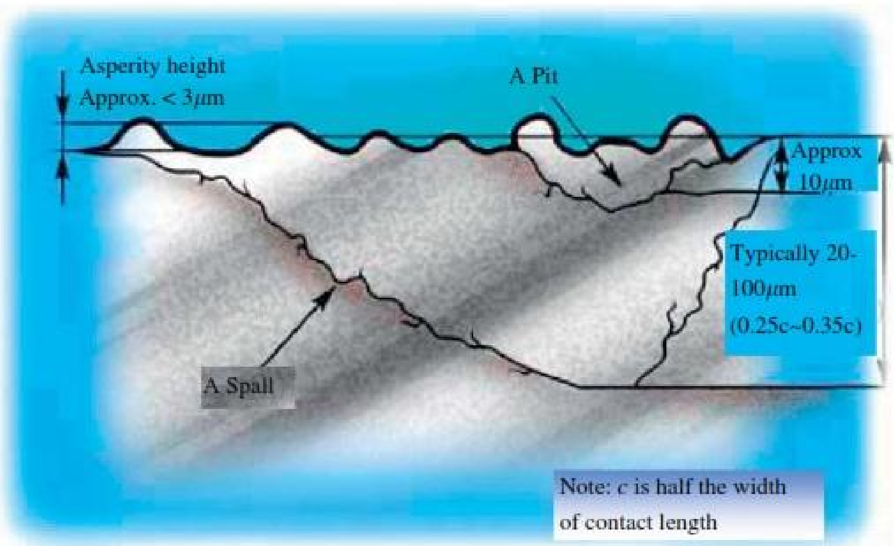


Figure 2.2: Differentiation of surface defects (Kuhnell, 2004)

Olver (2005) reviewed the mechanism of rolling contact fatigue and emphasized the importance of running-in condition. The running-in process consists of wear and plasticity mechanisms and both are strongly influenced by the lubrication film and contamination. The running-in topography will affect the subsequent surface contact behaviour, which differs from the original manufacturing topography. The initiation of rolling fatigue failure is identified by the formation of micro pits, which is attributed to the stress zone associated with the roughness of the surfaces in contact. The stress at the affected asperities zone can become extremely higher than the unaffected zone, hence the damage does not extend to greater depth due to the existence of the different stress level zone.

Nilsson et al. (2006) employed the primary profile depth parameter obtained using stylus profiler apparatus in order to investigate the abrasive wear of the spherical roller thrust. This method contributed to the wider option of surface wear evaluation by using surface parameter of the primary surface height as the wear depth hence allowing for comparison between the new and worn surface profile.

Choi and Liu (2007) investigated the deterioration of rolling contact bearing and divided the last stage of the rolling contact fatigue process into two phases. In the first phase, no significant change in the vibration amplitude can be observed as the crack initiation and propagation occurred below the surface. The second phase shows a significant increase of the vibration amplitude due to the formation and the progress of the spall on the surface.

The study on the development of the localized defects in the ball bearing by Karacay and Akturk (2009) revealed that the first defect was initiated at the inner race defect together with the slight defect of the ball. The outer race defect followed at the end of the experiment. The microscopic photos of damaged bearing showed various surface defects.

In order to study the relation between surface and defects, Ueda and Mitamura (2009) generated a defect on ball bearing inner raceway by dent initiated flaking. The defects seeded on a few bearings with different surface roughness but running in the same test condition. The results indicated that the greater surface roughness of rolling elements have a shorter life.

Halme and Anderson (2009) reviewed the surface polishing effect phenomenon at the running-in stage, which occurred as an effect of the plastics deformation of the surface asperities in rolling bearings due to certain running-in conditions. After this stage, the surface roughening by the formation of micro pits can occur due to stress levels and consequently leading to rolling contact fatigue at asperity contact areas. This small depth of the micro pits gives a low volume loss of the material.

Akbarzadeh and Khonsari (2011) investigated the running in state of gears in the mixed-EHL lubrication condition analytically and experimentally. The experiment used two rollers to represent the contact of a gear pair. The finding showed a substantial drop of surface roughness value initially and followed by slight decrease of surface roughness until the running-in process is completed. The result from the simulation showed an agreement with the experimental result. The authors explained this phenomenon as a result of plastic deformation and polished asperities, which subsequently decreasing the surface roughness. The simulation and experimental results are shown in figure 2.3

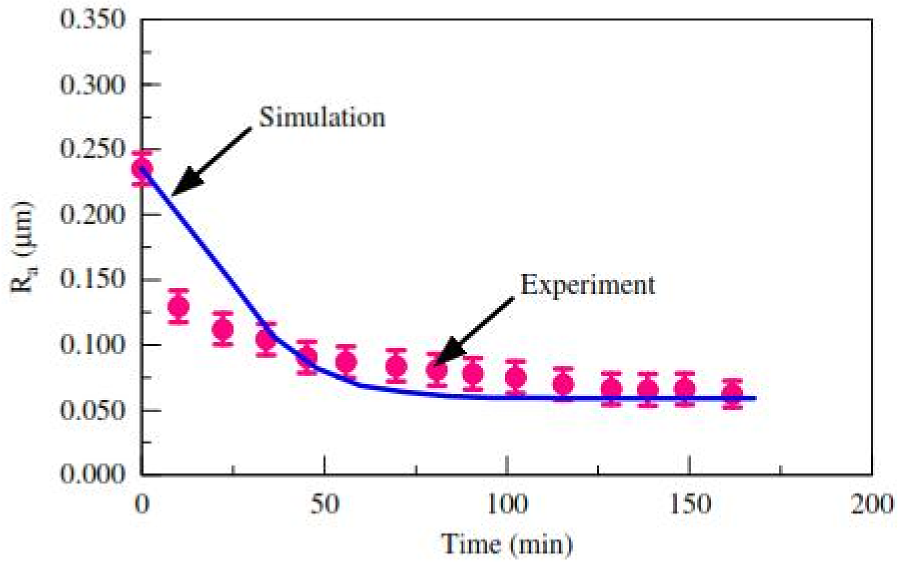


Figure 2.3: Roughness averages of simulation and experimental result as function of time (Akbarzadeh and Khonsari, 2011)

2.2 Form change studies

The rolling bearing form is normally measured by using the surface waviness. Since 1960's several dynamic models were developed in order to understand the vibration behaviour due to the number of undulation in waviness of the bearing components. However, these studies were limited to the surface waviness characterization of new bearing, which was performed to evaluate the quality of the surface finish. The waviness of the inner race, outer race and roller raceway proved to have their own effects toward vibration. According to studies by Wardle (1988) and Yhland (1992), with a radially applied load, the additional numbers of waviness may lead to occurrence of vibration. Momono and Noda (1999) reported that the behaviour of the contact between the rolling elements and raceway is similar to a spring where it fluctuates minutely during bearing operation. The noise and vibration generation regarding manufactured surface waviness were inevitable but the precision and quality of the bearings can be improved to minimize it.

Tallian and Gustafsson (1965) developed a linear dynamic model to analyze the vibration caused by waviness and successfully showed correlation between these two parameters. The surface waviness quality was found to have an effect on the noise and vibration level which was later confirmed by Lura and Walker (1972) where the rising of the vibration level was also affected by the geometrical irregularity in the form of waviness.

A theoretical and experimental work by Sunnersjo (1985) on the effect of bearing component's waviness on vibration showed that for bearings operating with radial load and stationary outer ring, the major vibration amplitude was produced by inner raceway waviness and varying roller diameter. The out-of-roundness of the rolling element produced a smaller effect compared to other components.

An analytical model of a rotor system was developed by Harsha et al. (2004) to investigate the non linear vibration response due to surface waviness in the races. A computer program was developed to simulate surface waviness of the components. The result showed that there are two important governing parameters affecting its dynamic behaviour; the number of balls and undulation in the ball bearing components. The vibration frequency peaks that will occur with a certain number of waviness were determined.

Changqing and Qingyu (2006) developed a dynamic model of ball bearings to investigate the dynamic properties of rotor system supported by ball bearings under the effects of both internal clearance and bearing running surface waviness. The numerical results show a good agreement with already existing models and prior authors' experimental researches. The result shows that in this case, the effect of outer race and inner race waviness on the cage speed variation is more apparent than the ball waviness.

Another dynamic model was developed by Cao and Xiao (2008) to investigate the dynamic characteristic of double row spherical roller bearing. The waviness of bearing components was modelled and the amplitude produced compared. The highest increased of vibration level occurred with the waviness of moving inner raceway and this proved that the surface quality of the moving race is the most critical component for improving vibration performance. The waviness of the stationary race leads to a strong resonance at the shaft frequency, roller passage frequency, and their super harmonics.

2.3 Rolling bearing vibration

The condition of the rolling bearing is commonly indicated by the vibration level. The rolling element bearing produces noise and vibration because of the sliding or rolling contact of the bearing components. According to Norton and Karczub (2003) the primary noise and vibration mechanism for rolling contact bearing is the impact between the rolling elements and the raceways. There are additional factors influencing the noise and vibration generated by the rolling element bearings which include manufacturing inaccuracies, improper installation, operating speed, supported load and lubricant (Patil et al. 2008).

The radially loaded rolling element bearings will generate vibration even if they are produced with a perfect geometry. This is because of the finite number of rolling elements to carry the load and the varying position in the load zone as the effect of bearing rotation resulting in the periodical variation of the bearing total stiffness. This variation of stiffness generates vibrations commonly known as varying compliance (Sunnersjo, 1978). The bearing geometry such as internal radial clearance and manufacturing surface waviness also play some role in generating vibration. However, with a high quality of manufacturing, these parameters have less influence on the bearing vibration levels. A significant increase of vibration level can occur with the presence of surface defects.

2.3.1 Vibration monitoring technique to assess the bearing surface defects

A brief review on vibration and acoustic measurement techniques for detection of rolling element bearing defects indicated the techniques commonly adapted which are vibration measurement in time and frequency domains, sound measurement, the shock pulse method and the acoustic emission technique (Tandon and Choudhury, 1999). The frequency domain or spectrum analysis was found to be the most widely used in bearing defect detection.

Mc Fadden and Smith (1984) developed a model of single point defect on the inner raceway of rolling element bearing under a constant radial load. The model considered the effect of bearing geometry, shaft speed, load distribution, transfer function and the exponential decay of vibration. Measured vibration spectra showed good agreement with the developed model. In order to extend the study to a wider range of defects, the model of single point defect was further improved by applying multiple defects (Mc Fadden and Smith, 1985). The multiple numbers of defects caused the variation of phases angle which subsequently modified the appearance of the spectrum. The phase angle of the components occurs according to the defects position and the frequency of the components.

Berry (1991) monitored the rolling element bearing health with vibration analysis and showed that the frequency spectrum can be divided into for basic categories, these are random ultrasonic frequencies (20–50 kHz), natural frequencies of bearing components (0.1–20 kHz), bearing rotational defect frequencies (1–1000 Hz) and sum and difference frequencies which are created when the different frequencies modulate each other. The vibration levels which are higher than 50 kHz were reported to originate from inside the material in micro geometric scale.

Acoustic emission is widely used technique to monitor the rolling bearing condition. Jamaludin and Mba (2002) employed acoustic emission (AE) technique to monitor the condition of low speed rotating rolling element bearings. The localized defect was seeded by spark erosion. The results showed that the rubbing or sliding action of the contacting bearing element also can be detected from the AE signal.

Kiral and Karagulle (2003) developed a computer program to model the dynamic loading of the bearing structure. The vibration signal was analyzed by using the most popular methods in condition monitoring and the results were compared. The finding showed that both the time and frequency domain techniques are sensitive to the change of the rotational speeds, structure geometry and loading type.

The comparison between acoustic emission and vibration analysis by Al-Ghambd and Mba (2006) identified the presence and the size of the defects on radially loaded bearing. The experimental test rig was designed such that various defects sizes could be seeded onto the outer race of a test bearing. It was concluded that AE provide earlier fault and defect identification than vibration analysis. Furthermore, the AE technique also was able to identify the defect size, which allow for the rate of the bearing degradation to be monitored.

2.3.2 Envelope analysis technique

A technical publication note by Konstantin-Hansen (2003) emphasized that an envelope analysis can be used for diagnostics of rolling element bearing local faults due to the generation of amplitude modulating effect on the bearing characteristic frequencies with a presence of faults. The bearing surfaces with local faults will produce pulse and a series of force impact during rolling contact, which can be detected in the envelope spectrum.

Several different vibration analyses were utilized by Ericsson et al. (2005) to detect local defects on the bearing and found that the best performing methods are wavelet-based and envelope-based techniques. These methods had between 9 to 13 % error rate and have potential to be improved.

Yuh-Tay (2007) studied the envelope analysis technique of the vibration signal to detect the bearing defect which was seeded by electrical discharge machining on the roller, inner race and outer race. The experimental result successfully detects the bearing defects at the seeded location and in good agreement with numerical results.

A vibration level progresses through natural defect development where the defects are not seeded and are closer to actual application (Karacay and Akturk, 2009). The vibration level was acquired and recorded in traditional vibration metrics and showed that the damage at the ball bearing where the location of defect was undetectable. Therefore, the envelope spectrum analyses are conducted at specified test durations in order to predict defect locations. The finding showed that when the defect size increases, the vibration level also increases. However, a general correlation between the defect size and the amplitude of the vibration was not established because the characteristic of the vibrations varies in every system.

2.3.3 Vibration levels in relation with tribological parameters

Wunsch (1992) measured the noise and vibration characteristic of grease lubricated of anti-friction bearings. The main interest in the study is to acquire a relation between grease viscosity and vibration level. The results indicated the higher vibration level was excited with the increase of base viscosity.

Williams et al. (2001) studied the vibration level of rolling element bearing by running a new bearing until failure. The vibration levels were recorded through traditional vibration metrics such as root mean square, peak value, kurtosis and crest factor. The rms trend of vibration acceleration shows sudden increase and decrease of vibration levels during running in. The fluctuation of the signal level was attributed to phenomenon known as healing which is described as smoothing of sharp edge by continuing rolling contact. The results established the defects produced naturally and the spectrum of the defects signal is presented.

The fundamentals for rolling bearing diagnostics of rolling contact fatigue and wear has been reviewed by Halme and Anderson (2009). The authors stated that the rms value of the vibration acceleration level decreases with the smoothened asperities during running-in. However, the background noise and other vibration sources of the bearing influenced the detect ability of this signal. The vibration level of bearing can be induced by micro pitting due to contact dynamic, surface roughening and the increasing contents of wear particles in the contacts.

The detection of starve lubrication is important in order to ensure proper lubrication of the operating bearing. Boskoski et al. (2010) proposed a new technique for detection of starved lubrication bearing in electrical motors. The technique was developed to provide a solution to the problem of detection of improperly lubricated electrical motor bearings through vibration analysis. The vibration signals acquired within a limited time frame of only several seconds proved to have a capability of detecting improperly lubricated bearings. Specific band pass filter parameters were acquired by cyclostationary analysis and spectral kurtosis, and the features were identified in the envelope spectrum. Improper lubrication can be expressed by the increasing amplitude of the cage and ball passage frequency in the spectral component. The finding agrees with the theory that with starved lubrication, the absence of lubricant film will cause a lack of damping and allows more space of the bearing clearance when a ball pass through the load zone. As a result, the vibration amplitude produced during ball passage through the load zone is higher than a properly lubricated bearing. This technique is suitable for the end quality assessment of rolling bearing lubrication.

2.4 Rolling bearing lubrication

Lubrication plays an important role in order to ensure a smooth bearing operating condition. The function of lubrication is to form a layer of a different material between the surfaces that reduce the friction force between them. A vibration in a rotating system might happen because of several factors and one of the possible causes is the lubrication region where the bearing is in operation.

2.4.1 Lubrication performance

Chiu (1974) analyzed and predicted a lubrication film starvation in rolling contact system. The observation indicated at high rolling speed and viscosity, the film thickness shows a tendency to level off. The film thickness was also reduced even with increasing velocity.

The vibration velocity of the light mineral oil and clean grease lubrication is compared by Howard (1975). The finding shows that the grease lubrication produced higher vibration level at high frequency band (1.8 to 10 kHz) while the similar results obtained for grease and oil lubrication at lower frequency band (50-300 Hz).

Sayles and Poon (1981) highlighted that a constant film thickness between the rolling elements and raceway rarely exists even under full film lubrication because of the interaction of the asperities. The refinement by Ebert (2010) stated that the full film lubrication (hydrodynamic lubrication) cannot be reached under certain operating conditions such as low speed, very high load, and at very high temperature as a thin and inadequately separating lubricating film is formed.

Cann et al. (2004) investigated the transition between fully flooded and starved regimes in elasto-hydrodynamic lubrication of oil by controlling four primary factors that affecting lubrication level; they are volume, contact dimensions, oil viscosity and speed. The results showed how the boundary between fully flooded and starved regime depend on these parameters and demonstrated the relationships among load, speed, contact size, oil viscosity and volume, and the lubrication regime. A single dimensionless parameter (SD) has been established indicating the onset of starvation.

According to Maru et al. (2005) and Serrato et al. (2005), the change in lubrication condition of rolling bearing mainly affect its vibration in high frequency (HF) band (600 Hz-10 kHz). This is confirmed by Serrato (2007) by evaluating the vibration behaviour with the change in oil viscosity. The author separated the rms vibration level as function of time in low and high frequency band. The overall rms vibration level similar with the rms values in the HF band. In addition, the vibration level measurement at high speed frequency was found to have better detect ability.

The effect of lubricant viscosity towards vibration behaviour of roller bearings is studied by Serrato et al. (2007). The lubrication regime of operating bearing was presented by lambda λ (the ratio of film thickness to the roughness) calculation. The results reveal the different vibration behaviour in accordance to lubricant viscosity. It has been observed that the vibration level increases with the decrement of the minimum film thickness and lambda value as shown in figure 2.4. A relationship between high frequency vibration and lambda factor was found to be correlated with the Stribeck curve. Similarly, the vibration level and friction coefficient increase with the decrement of lambda value.

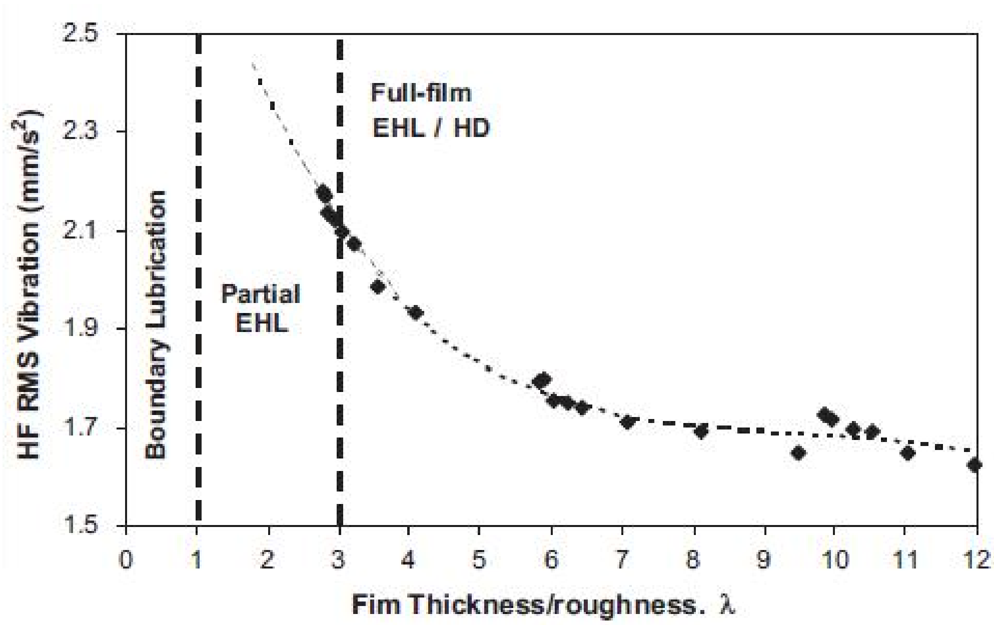


Figure 2.4: RMS vibration level as a function of λ (Serrato et al., 2007)

Ebert (2010) emphasized that with an appropriate application of the tribological system consisting of bearing, lubricant, operating and environmental conditions including the bearing temperature, a long bearing life can be achieved reliably. Elastohydrodynamic lubrication theoretically can form a load carrying lubricating film which produces optimum separation of the rolling contact surfaces. This can be achieved by the proper coordination of bearing load, speed, lubricant viscosity, and surface quality of the contact areas. However, this lubrication (hydrodynamic lubrication) cannot be reached practically under certain operating conditions. After long operation period, the life of a rolling bearing is limited by the formation of first subsurface cracks, which typically progress to macroscopic pitting.

2.4.2 Grease lubrication

According to Moore (1969), the grease re-lubrication of bearing maybe carried out in two ways; 1) by replenishment of the fresh grease to the starved bearing and 2) by repacking which is by dismounting the bearing for cleaning, and the fresh grease applied on the bearings.

The optimum quantity of grease is expected to give the best operating condition was obtained. Shawki and Mokhtar (1979) investigated the optimum grease quantity for roller bearing lubrication. The result from the experiment showed that the insufficient lubrication will give rise in operating temperature and friction. The author suggested that the best quantity is about 20% to 30% of the bearing clearance space of bearing element for optimum condition of lubrication.

The characteristic of grease lubrication with sufficient and starved lubrication condition may differ and influence the performance of rolling contact bearing. the characteristic of grease lubrication is analyzed by Cann (1999). The finding in this study showed that in grease lubrication bearings, a fully flooded condition normally occurred after freshly lubricated. However, the bearing will run under starved condition in a short time.

Miettinen and Anderson (2000) measured the acoustic emission level of rolling bearing running in lubricated condition with contaminated grease. The cleaning and re-greasing of the bearing operated with contaminated grease reduced the acoustic emission level to a level which was about one-half of the AE level before cleaning and re-greasing take places. However, the AE level was still higher than the new bearing lubricated with the clean grease.

A friction torque for the grease lubricated thrust ball bearings was measured by Cousseau (2011) with the surface wear investigated by using scanning electron microscopy. Torque tests were performed using a modified Four-Ball machine, where the Four-Ball arrangement was replaced by a rolling bearing assembly, allowing for the comparison of the performance of different lubricating greases. The bearing friction torque shows a decrement when the bearing speed increases. The results present that for the grease with mineral base, the minimum film thickness (h_{min}) and specific film thickness (λ) tend to decrease at the higher speed. The important aspect to be noted that the film thickness between the contact surfaces not only affected by lubricant parameter but also on lubricant starvation, which depends on the interaction between the grease thickener and the base oil.

2.5 Discussion

The literature review showed the recent research trend in the study of surface degradation of roller bearings in particular defect initiation on the bearing surface and the propagation. Initially, small cracks are formed at the contact surface, which then propagate to form micro pits, subsequently progressing to form spalling and flaking due to continuous rolling contact and stress concentration (Fernandes, 1997).

The running-in state involves smoothening of the surface asperities by plastic deformation which reduces the surface roughness until the running-in process is completed (Olofsson and Bjorklund, 1998; Akbarzadeh and Khonsari, 2011). After this stage, the surface roughening by micro pitting can occur due to stress levels and consequently lead to rolling contact fatigue at areas of asperity interaction (Halme and Anderson, 2009). However, the experimental result regarding roller bearing surface roughness during the running-in has never been reported.

The existing technique for determination of rolling bearing form is by surface waviness measurement. A lot of dynamic models were developed to predict the vibration behaviour due to the undulation of the surface waviness (Tallian and Gustafsson, 1965; Harsha et al., 2004; Changqing and Qingyu, 2006; Cao and Xiao, 2008). The study by Cao and Xiao (2008) showed the highest increased of vibration level occurred with the waviness of moving inner raceway and this proved that the surface quality of the moving race is the most critical component for improving vibration performance. There is a need to measure the surface waviness of bearing after damaged and to monitor the degradation of surface waviness from the beginning of experiment until failure in order to understand how the bearing surface is degraded.

The vibration levels were generally recorded through traditional vibration metrics such as root mean square, peak value, kurtosis and crest factor (William et al., 2001; Karacay and Akturk, 2009). The frequency domain or spectrum analysis was found to be the most widely used approach for bearing defect detection and envelope analysis is one of the best methods to detect the location of the bearing defects (Ericcson et al., 2005).

The lubrication performance can be controlled by sufficient lubrication. The vibration level increases with the decrement of the minimum film thickness and lambda value (Serrato et al., 2007). In grease lubrication bearings, a fully flooded condition normally occurs after fresh lubrication but the bearing will run under starved condition in a short time (Cann, 1999). The optimum quantity of grease required was investigated and expected to give the best operating condition (Shawki and Mokhtar, 1979). A work by Cann et al. (2004) found that the transition between fully flooded and starved regimes in EHL lubrication of oil can be controlled by four primary factors; they are volume, contact dimensions, oil viscosity and speed. Improper lubrication can be expressed by the increasing amplitude of the cage and ball passage frequency in the spectral component of the vibration signals that acquired within a limited time frame (Boskoski, 2010).

2.6 Summary

From the literature review, it is clear that the roller bearing condition particularly the surface degrades throughout the operating life which can be related to the vibration level of the bearing which in turn is strongly influenced by lubrication. Most of the bearing damage analyses were carried out using seeded defects which are difficult to relate to natural wear of the bearing and none of the work reviewed has looked into the progressive damage of the bearing surface throughout its operation. The relationship between surface condition and vibration level has never been reported. It is important to determine how the roller bearing surfaces changes and their effect on the vibration level of the bearing which can be measured using the rms vibration value and the envelope analysis.

CHAPTER 3

METHODOLOGY

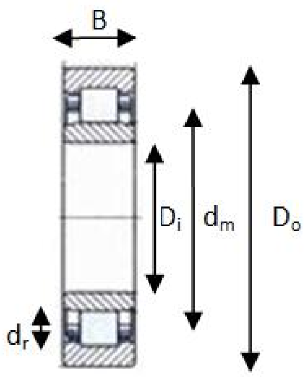
3.0 Overview

In this chapter, the characteristic of the test system, surface characterization, vibration monitoring and lubrication regime identification techniques are discussed. A brief of the research flow is also presented.

3.1 System characteristics

In order to evaluate the surface degradation throughout the bearing operation, a detachable roller bearing is selected for this study which is the NU205E.TVP2.C3 cylindrical roller bearing made from chrome steel (Asia Bearings, 2012). The geometry of the bearing is shown in table 3.1.

Table 3.1: Roller bearing geometry



Number of roller	13
Roller diameter (dr)	7.5 mm
Inner bore diameter (Di)	25 mm
Outer ring outside diameter (Do)	52 mm
Pitch diameter (dm)	38.5 mm
Width (B)	15 mm
Weight	0.14 kg

A rotation system is designed and fabricated for this study. Figure 3.1 shows the experiment test rig which consists of bearings, motor, load, shaft coupling, speed controller and data acquisition system. The shaft is supported by two pillow blocks. The test bearing is fitted to one end of the shaft. During operation, the bearing inner ring rotated with the shaft while the outer ring remained stationary. The steel shaft is