İSTANBUL TECHNICAL UNIVERSITY ★ INSTITUTE OF SCIENCE AND TECHNOLOGY

INVESTIGATION OF LUBRICATION EFFECT ON A DIESEL ENGINE TIMING CHAIN WEAR

M.Sc. Thesis by
Özay POLAT, B.Sc.

Department : Mechanical Engineering

Programme: Automotive Engineering

JANUARY 2008

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(503041729)

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Supervisor (Chairman): **Prof. Dr. Metin ERGENEMAN**

Members of the Examining Committee **Prof.Dr. Mustafa ÜRGEN**

Prof.Dr. Cem SORUŞBAY

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<u>İSTANBUL TEKNİK ÜNİVERSİTESİ ★ FEN BİLİMLERİ ENSTİTÜSÜ</u>

BİR DİZEL MOTORUNDA SUPAP TAHRİK ZİNCİRİ AŞINMASI ÜZERİNDE YAĞLAMA ETKİSİNİN İNCELENMESİ

YÜKSEK LİSANS TEZİ

Müh. Özay POLAT

(503041729)

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Diğer Jüri Üyeleri Prof. Dr. Mustafa ÜRGEN

Prof. Dr. Cem SORUŞBAY

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January 2008 Özay POLAT
Mechanical Engineer

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ABBREVIATIONS

NVH	: Noise, Vibration and Harshness
NO_x	: Nitrogen Oxide
EGR	: Exhaust Gas Recirculation
EHD	: Elasto Hydrodynamic
TBN	: Total Base Number
TAN	: Total Acid Number
SAE	: Society of Automotive Engineers
KOH	: Potassium Hydroxide
ZDTP	: Zinc Dialkyl Dithiophosphate
VI	: Viscosity Index
H_2SO_4	: Sulphuric Acid
EP	: Extreme Pressure
OEM	: Original Equipment Manufacturer
ACEA	: Association des Constructeurs de L'Automobile
API	: American Petroleum Institute
CAE	: Computer Aided Engineering
R&R	: Repeatability and Reproducibility
MSA	: Measurement System Analysis
ASTM	: American Society for Testing and Materials
HC	: Hydrocarbon
SEM	: Scanning Electron Microscope

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F	: Force
W	: Weight
N	: Normal Force
θ	: Inclination Angle
μ	: Friction Coefficient
V	: Velocity
h	: Oil Film Thickness
R	: Surface Roughness
η	: Lubricant Viscosity
F_{link}	: Chain Link Force
k_{link}	: Chain Link Stiffness
c_{link}	: Chain Link Damping
Δ	: Chain Pitch Displacement
$(\dot{\ })$	: Derivative with respect to Time
k	: Chain to Sprocket Contact Stiffness
c	: Chain to Sprocket Contact Damping
F_x	: Chain to Sprocket Contact Horizontal Force
F_y	: Chain to Sprocket Contact Vertical Force
π	: Pi number
n	: Revolution per Time
D	: Sprocket Pitch Diameter
p	: Chain Pitch Dimension
P	: Power
T	: Chain Tension

INVESTIGATION OF LUBRICATION EFFECT ON A DIESEL ENGINE TIMING CHAIN WEAR

SUMMARY

Recent growth in automotive industry and enhanced competitive environment have made it an obligation for manufacturers to provide more features in their product development and technology adaptation. Durability and long service life are the important expectations which drive improvements in vehicle and engine technology.

Lubricants take important place in internal combustion engines due to their serious impact on durability and service life requirements. Legislations and increased expectations from engines have made it an obligation to develop specific lubricants to function in different sub-systems of an engine which need to target certain service life. Today's lubrication technology can offer unique features on lubricants by improvements in base oil and additives.

Engine timing chain drive system is to transfer the rotational motion of crankshaft to camshafts which excite the engine valves in valve train system. Chains are under continuous lubrication during operation. Improper lubrication and/or aggressive running environment induce wear on chain internal elements leading to longitudinal elongation.

Due to nature of the combustion process, diesel engines produce soot and residues which form a suitable environment for wear on engine components. Previous tribological investigations demonstrate that particular elements in engine oil apart from soot and certain parameters that designate lubrication performance also have effect on wear characteristics.

In this study, using a special test rig for acceleration of timing chain wear, a mineral, a semi synthetic and a fully synthetic oil have been evaluated in terms of chain wear performance. Wear elongation of chain samples have been measured and elongation trends for each particular test is developed. Further tribological inspections are performed on SEM for visualization of wear characteristics. Same specification lubricants have separately been tested on real life engine environment at dynamometers. Similar elongation and common abrasive wear behaviour have been obtained for mineral and semi synthetic lubricants at either bench or dynamometer tests. Whereas fully synthetic lubricant has been found to significantly reduce elongation of timing chain and increase component service life. Detailed inspections on running surfaces have demonstrated only polishing wear with very minor change in dimensions. The results observed on the bench had an acceptable approximation to the findings obtained from real life engine tests. Overall results of the studies demonstrated that simple chain wear elongation rig tests can be used as part of diesel engine design development process and have potential to reduce the costs of engine tests dedicated to evaluate timing chain system durability.

BİR DİZEL MOTORUNDA SUPAP TAHRİK ZİNCİRİ AŞINMASI ÜZERİNDE YAĞLAMA ETKİSİNİN İNCELENMESİ

ÖZET

Otomotiv endüstrisinde yakın zamanda gerçekleşen büyüme ve rekabet ortamı, müşteri beklentilerini karşılama çabasındaki üreticilerin geliştirdikleri ürünlerin ve kullanılan teknolojinin daha fazla özelliğe sahip olmasını gerektirmektedir. Taşıt ve motor teknolojisinin gelişimini yönlendiren önemli kriterlerden ikisi dayanıklılık ve uzun servis ömrüdür.

Dayanıklılık ve servis ömrü üzerindeki etkisi nedeniyle içten yanmalı motorlarda yağlayıcılar önemli bir yer almaktadır. Beklentiler ve yasal zorunluluklar, belirli bir servis ömrünü hedefleyen motorun farklı alt sistemleri üzerinde çalışacak yağların özel olarak geliştirilmesini bir zorunluluk haline getirmiştir. Bugünün yağ teknolojisi, baz yağda ve katkı maddelerinde yapılan ilerlemelerle, ihtiyaca uygun özellikleri sunabilmektedir.

Motorda supap tahrik zinciri, krank milinden aldığı dönel hareketi külbütör sistemi tarafından tahrik edilen supapları çalıştırmak üzere kam millerine aktarır. Zincirler çalıştırılırken sürekli bir yağlama yapılması gerekmektedir. Uygun olmayan yağlayıcı ve/veya aşındırıcı çalışma ortamı, zincir iç elemanları üzerinde malzeme kaybına neden olarak doğrusal uzamaya sebep olur.

Dizel motorların işletme özellikleri nedeniyle ürettiği kurum ve artık maddeler motor yağına karışarak, motor elemanlarının aşınmasına uygun bir ortam oluşturmaktadır. Triboloji üzerine yapılan bilimsel çalışmalarda, kurum haricinde motor yağının içerisinde bulunan bazı elementlerin ve yağların performansını belirleyen bazı parametrelerin de aşınma üzerinde etkisi olduğu gösterilmiştir.

Bu çalışmada, supap tahrik zinciri aşınmasını hızlandıracak özel bir test rigi kullanılarak, mineral, yarı sentetik ve tam sentetik bazlı üç değişik özelliğe sahip motor yağının zincir aşındırma performansı değerlendirilmiştir. Zincir numunelerinin aşınmaları ölçülmüş, boy uzamalarındaki değişim belirlenmiştir. Daha detaylı tribolojik incelemeler SEM ekipmanı vasıtasıyla yapılmıştır. Aynı özelliklere sahip yağlar, bu kez dinamometrede gerçek motor üzerinde test edilmiştir. Mineral ve yarı sentetik bazlı yağlar hem dinamometre hem de rig testlerinde aynı seviyelerde uzama ve aşınma karakteri göstermiştir. Tam sentetik yağ ise tahrik zincir uzamasını belirgin bir şekilde azaltmış, zincirin ömrünü artırmıştır. Çalışan yüzeylerdeki detaylı incelemelerde ise düşük seviyelerde boyutsal değişim ve yüzeylerde parlama görülmektedir. Genel anlamda rig testlerinde alınan sonuçlar, dinamometrede gerçek motor üzerinde yapılan testlere kabul edilebilir bir yaklaşıklık göstermektedir. Elde edilen tüm sonuçlara bakıldığında ise yapılan bu çalışma ile basit zincir aşındırma test riglerinin dizel motoru tasarım ve geliştirme sürecinin bir parçası olarak kullanabileceği ve supap tahrik sistemi ömrüne yönelik motor testlerinin maliyetini düşürebilecek potansiyele sahip olduğu gösterilmiştir.

1. INTRODUCTION

The outstanding growth of the vehicle technology in recent years has increased the expectations from an automobile and its engine which are mainly framed by regulations. Increase in the number of vehicles available in the world has significantly become apparent after the automobile industry had turned into a form of serial production. The more internal combustion engines running on the roads have made the exhaust pollution and its health effects on human become the most important parameters of legal regulations on automotive industry.

The emission regulations are not the only basis of improvements on vehicle technology. After vehicle has just slipped off being only transportation means, by the effect of the various competitive suppliers in the market, the customers or drivers now have very different aspects from a vehicle like fuel economy, driving performance both from engine and chassis combined with a suitable noise, vibration and harshness (NVH) package. Drivers also are now looking after an acceptable service life and low maintenance costs from overall vehicle especially from their engines.

Considering the different challenges on vehicle technology, the engine developers need to provide complete system which is able to meet both legislation targets and customer expectations. Within this development challenge, diesel engines due to its fuel injection technology improvement in recent years have a more unique group of interest compared to those on gasoline engines. In past, the diesel engines were almost a standard for heavy duty vehicles, but the more easily acquired performance presented in last quarter of 20th century to customer which contains fuel economy, torque and speed performance with optimized NVH characteristics, has made a significant growth of diesel engine variants on all vehicles including passenger cars. Especially one of the most popular future studies on vehicle technology is hybrid technology which consists of an internal combustion engine and an electric motor, are using diesel engine as almost a standard application due to the targeted low energy consumption characteristics of these new vehicle packages.

As summarized above, diesel engines are advanced to present very wide range of performance capabilities like specific power & torque output, fuel economy, combustion efficiency and long lasting durability. Durability is one of the major challenges and diesel engine components are exposed to provide long service intervals without any durability concern that leads to performance loss.

Component durability is more important on diesel compared to gasoline engines due to main differences in general characteristics. Diesel engines have very different operating principles in very basic explanation, spark ignition engines are where the fuel is ignited by a spark while the diesel engine has a compression ignition which provides increase in pressure and temperature in compression process causing fuel to ignite instantly [1]. This expression is able to describe the major variations of the loading and operational severity exposed on to diesel engine components.

Most usual durability issues observed on diesel engines are wear or fatigue induced failures. Wear appears on dynamic components running in certain tolerance of clearance. Lubricants take important place to maintain the durability properties of the running parts which are in contact to each other under different sliding or rolling operation modes. The lubricants used today are not only responsible to provide wear reduction on dynamic systems, but also need to meet some specific requirements. The stretch emission targets are directing significant improvements on exhaust products like NO_x and carbon particles. In order to achieve reduction in NO_x, as applied on different type of vehicles, Exhaust Gas Recirculation (EGR) has been found as an efficient application for light duty diesel engines [2]. However, EGR and other combustion parameters which are optimized to reduce particles in exhaust gases are increasing the combustion deposits passing through the engine oil which are finally leading to a significant increase in wear inducing components inside lubrication [3]. Lubricants now need to have a more complex structure which contains very different type of additives assigned to very different roles.

Wear occurs as a result of unacceptable running condition of the engine components being induced by improper frictional behaviour of contacting surfaces. Engine friction not only acts as a wear operator but also takes an important part of the energy consumption or power loss of a vehicle system. The Figure 1.1 shows the share of engine frictional losses inside the general energy consumption of a vehicle.

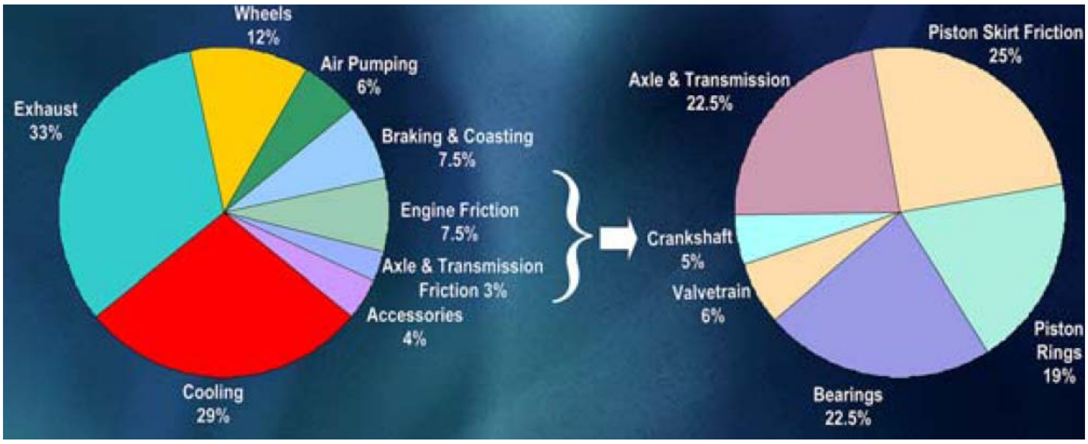


Figure 1.1 Distribution of energy consumption share of a light-duty vehicle [4]

Improvement in lubrication technology has presented a wide range of application on different aspects of vehicles. A heavy duty vehicle equipped with a diesel engine and a sport type passenger car using a high speed gasoline engine naturally have different performance criteria's in terms of service life, vehicle speed, torque level, fuel economy and emission targets. Therefore lubrication selection is critical for the engine overall performance and generally needs specific development for each unique application.

Engine operating conditions and specific system dynamics also contribute to wear. Taking the lubricant effect out of interest, the contact force and relative motion between components determine the regime of lubrication either boundary, mixed or hydrodynamic. These two parameters are affected by the mode of engine and sub system dynamic characteristics, customer usage and environmental conditions. Therefore, drawing the system boundaries is difficult and hard to evaluate for its direct effects on wear performance of specific engine components.

Currently, there is no defined scientific model that can clearly explain the wear in lubricated engine components. As discussed above, engine development is combined by the investigation and optimization in engine dynamics that needs to be supported by specific lubricant developments initiated on component level design verification tests.

This study represents an experimental investigation on one of the critical components of a light duty diesel engine which is timing drive chain. Various type of selected unique specification of lubricants will be tested on a wear elongation rig to identify the differences of chain wear performances. Tests will be carried out in a similar manner on dynamometers with real engines. The chain wear elongation trend will be built for both bench and dynamometer tests. End of test chains will be inspected through metallurgic analysis in order to determine the mode of existing wear. The results will be compared and evaluated targeting to identify the differences between rig and engine tests by an assessment of the rig test results in the process of engine development.

2. TRIBOLOGY FUNDAMENTALS

Relative motion of two contacting surfaces builds up friction, which turns a part of the kinetic energy into heat energy dissipating as temperature increase on the surfaces. Friction happens due to microscopic surface interactions which generally induces wear. In order to reduce or control wear, lubrication is used as an engineering application. The simple definition of tribology is the engineering science concerning the surface interactions under investigations of friction, wear and lubrication. [5]

2.1 Friction

Friction can be expressed as the resistant to motion that builds up when an object slides against another. Friction force is the result of the friction between the surfaces which always opposes to the direction of motion [6]. It's an inevitable situation which exists on engineering applications. Some applications use friction as the basic tool of the transfer function whereas others try to avoid as much as possible. For example, wheel brake or the gearbox clutch systems operate under friction to enable control over desired output from their mechanisms, while on particular dynamic components of engine it is desired to reduce the friction [7].

Specific to each application, there is a dimensionless parameter which is dependant on the material and surface morphology between two interacting bodies called friction coefficient.

There are three types of friction concerning the common engineering applications which are static, sliding and rolling friction.

2.1.1 Static Friction

This type of friction is generally not an interest for tribology science due to accumulation of the force against relative motion happens during the inactive position of the interacting bodies. This is generally called as the friction force against start of the motion.

A typical example for static friction is the object standing on a ramp with no action to downwards as shown on Figure 2.1. Under gravitational force (W) and the external force (F), the object keeps its stationary condition as the compounds of these forces on the axis of motion are not more than the friction coefficient acting on the opposite direction.

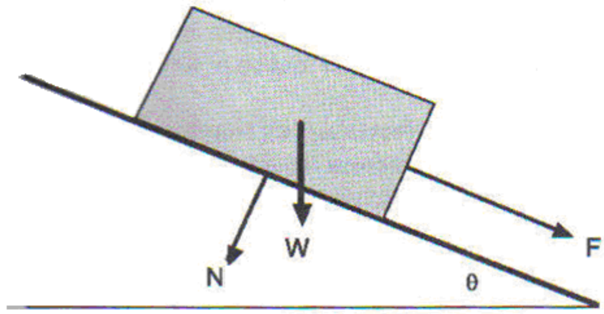


Figure 2.1 Stationary block on an inclined plane [7]

The static friction coefficient is named as the measure of the maximum friction force that should be overcome in order to start a macroscopic motion between two bodies in contact. This coefficient expressed by μ and does not depend on the normal load [8].

2.1.2 Sliding Friction

Sliding friction simply occurs when a body is moved or slides over another in the opposite direction of motion. The sliding friction force occurs after a complex combination of adhesion and deformation [5]. Material transfer from harder to softer asperities due to deformation of surface protuberances and active adhesion forces both induce friction force.

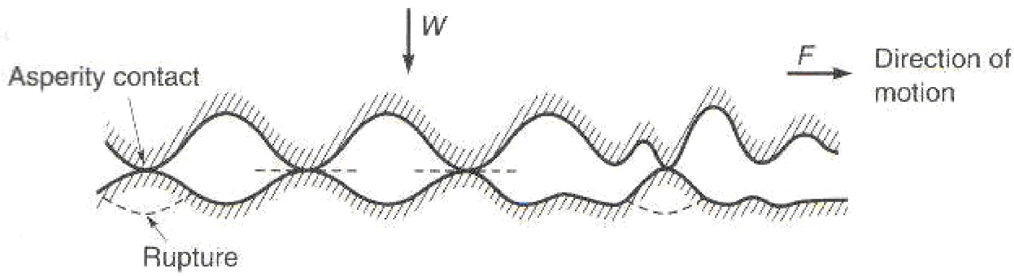


Figure 2.2 Schematic of two rough surfaces in a sliding contact [6]

The friction coefficient concerned in sliding friction is expressed as the kinematic friction coefficient which is different than static friction coefficient.

The general knowledge on this difference reveals that the force required to start up a motion is always greater than the force to maintain that motion. This equally expresses that the dynamic (kinematic) friction coefficient is less than static coefficient. There are different theories and proofs for expression of sliding friction coefficient and friction mechanism in tribology literature which will not be discussed in this section.

2.1.3 Rolling Friction

Rolling friction is a resistance to motion that occurs during a cylindrical or spherical surface is rolled over another. Cylindrical and spherical surfaces are used for this explanation because rolling is almost only applicable to conforming shape of objects which have comparably little surface asperities [6]. Figure 2.3 demonstrates an example for the rolling motion of a cylinder on a horizontal surface. N is the normal load acting on the body and F is the rolling friction force opposite to the direction of rotation.

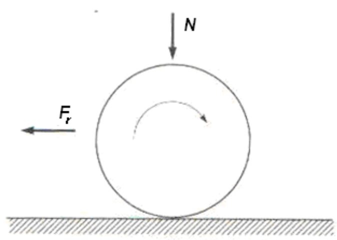


Figure 2.3 Schematic of a cylinder rolling on a horizontal surface

It is not always easy to distinguish rolling from other types of motion like sliding and spinning because on most of the engineering applications, these motions are somewhat occurring as a combination of two or three. As a common knowledge, it's easier to roll a body over another than sliding. Therefore, it can be simply expressed that rolling friction is less than sliding friction for equivalent applications.

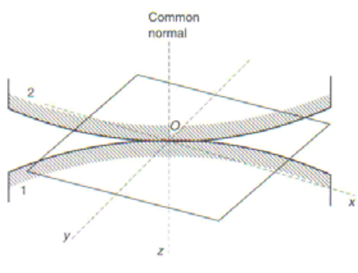


Figure 2.4 Two non-conforming bodies [6]

Rolling friction phenomenon is not completely different than sliding friction but basic distinction arises from the difference in kinematics of these motions. As shown on Figure 2.4, sliding motion takes place on tangential plane to contacting area while rolling motion takes place in normal direction which is main reason for reducing the adhesive forces acting on friction formation. Deformation of the asperities is always valid for rolling friction which takes the bigger share on total resistant force [6].

2.2 Wear

Wear is simply expressed as the loss of material between two contacting surfaces. This can be either from one or both of the surfaces. The type of relative motion on the surface that forms wear may be sliding, rolling or a form of impact action [6].

Wear is an output for particular applications. Wear of a pencil is the result of its usage and is needed to fulfil its basic function. Generic mechanical operations like machining or grinding are also the typical areas that engineers use controlled wear as an outcome.

In general there are two categories of wear in terminology considering the existence of a lubricant in surface which are called dry and lubricated wear. For engineering applications, there are commonly accepted types of wear that mainly can be listed as adhesive, abrasive, fatigue wear [6].

2.2.1 Abrasive Wear

Existence of the protuberances of a surface or particles in the interface of the relative motion leads to surface deformations or cracks called abrasive wear.

The surface roughness, material properties of the particles with contacting bodies and also the operating conditions designates the level of abrasive wear. The harder material or particle deforms or destroys the softer material.

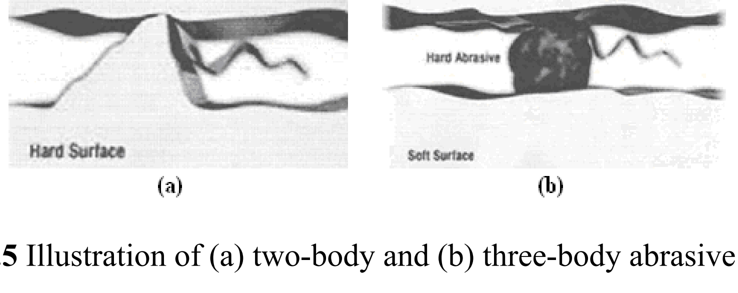


Figure 2.5 Illustration of (a) two-body and (b) three-body abrasive wear [9]

As shown in the first part of Figure 2.5, the asperities on the hard surface contact to the soft surface leading to abrasive wear. The mechanical shaping operations like grinding, turning or cutting are the generic examples for two body abrasive wear that occurs as a form plastic deformation.

Three-body abrasive wear shown on the second part of the above figure explains that the hard surface act as the third body inside the contact region. The hard particles between two surfaces have enough hardness to remove material from softer or both surfaces as plastic deformation or surface fracture [5].

2.2.2 Adhesive Wear

In adhesive wear, the effect of hard particles is not considered as a part of wear mechanism. Relative motion between surfaces exerts a shearing force onto each of the bodies. The severity of this force is affected by the contact stress and speed. As shown on Figure 2.6, asperities existing on each surface are removed or force another asperity on the opposite side to be removed. Under this process, a continuous material transfer occurs from one surface to another.

Transfer of surface particles results with a deterioration in surface conformity and finally loss of material. The new formation of the surface structure induces an improper surface contacts caused by irregular protrusions and continuous wear [5].

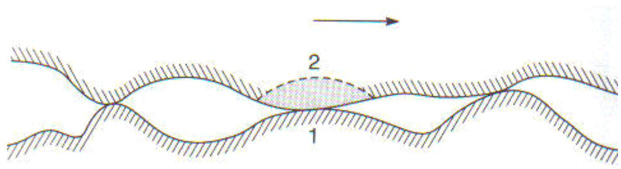


Figure 2.6 Illustration of surface break during adhesion [6]

2.2.3 Fatigue Wear

Fatigue wear is observed when there is an irregular loading acting on the component interface, generally when loading is alternating. The high frequency of the repeating loading cycles combined with high shearing velocities apply a resonance on the surface which induce cracks and finally surface detachments. These detachments leave grooves on the surfaces. In tribology this is called as pitting.

The frequent loading and unloading on surfaces are due to the combination of rolling and sliding motion which both excite the shearing force on tangential plane and the deformation force acting on the normal of the contact surface [10]. Therefore the amount of surface wear which is a measure of abrasive or adhesive wear performance is not an interest for pitting. This type of wear performance is determined by the time or number of cycles before crack initiates.

2.3 Lubrication

Lubrication inserts a region of low shear strength material between contacting surfaces [5]. This layer between two bodies is to be exposed to the shearing effect and hence prevents the surfaces to get into interaction to each other. This may not always be the case as lubricant may not be sufficient to prevent the contact between surface protuberances. There are three different types of lubrication which are called boundary, elasto-hydrodynamic (EHD) and hydrodynamic lubrication. Each lubrication regime is clustered against the film formation characteristics and the type of contact between surfaces.

Lubricants are used to reduce the friction and hence wear between the contacting surfaces of machine elements. Lubrication may be gas, liquid or semi solid like greases. They are mainly used to separate the asperities of the contacting surfaces.

The level of asperity interaction and the conformity of the contacting surfaces are main parameters, which are able to determine the lubrication condition. The lubricant properties, loading, surface roughness and velocity of the relative motion are the important contributing factors acting on formation of contact regimes. Stribeck diagrams are commonly used during tribological investigations concerning friction coefficient variation with respect to dynamic oil viscosity, acting load and operational speed. The combination of these three parameters with the formula shown on the apses of the sample Stribeck curve on Fig 2.11 is called the Sommerfeld number [19].

It is clear from the diagram that a relatively thicker oil film builds up between surfaces with increasing Sommerfeld number. The diagram also shows that there is a strong direct correlation between the lubrication film level and friction force until a certain level, which turns into an inverse effect to increase friction force under active viscous forces building up when the oil gets sufficient thickness.

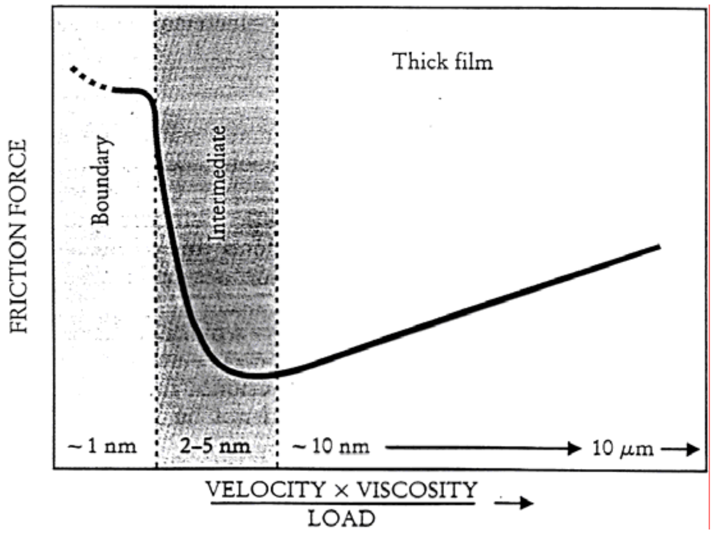


Figure 2.7 Stribeck diagram with representation of oil film thickness [20]

2.3.1 Boundary Lubrication

During the existence of very low relative motion between the surfaces or under comparably higher loading conditions, lubricant is unable to provide a sufficient hydrodynamic oil film that will prevent the direct contact of the surface protrusions. The surface structure is also effective on the level of the asperity contact. The expected responses of this type of tribological system are high heat generation and energy loss due to high friction, which results with high wear rates. The oil film thickness in this condition is too minor that viscous forces are neglected. To prevent excessive wear rates, boundary lubricants are used which form a protective layer on surfaces reducing the direct asperity contact. These kinds of lubricants need also to have a sufficient level of viscosity so that the layer is able to carry the most of the active shearing forces. Under very low viscosity lubricants, the friction and wear control is more difficult to maintain.

2.3.2 Elasto-hydrodynamic (EHD) Lubrication

The metal based surfaces get deformation under heavy loading conditions which causes a relative increase on the contact area. This certainly reduces the amount of load per section and the contact stresses that relatively increase the load carrying capability of the oil film. Another theory in EHD lubrication is that the high contact pressures enhance the viscosity character of the oil thereby prevents the oil to break down under high shearing forces [7].

In general, EHD contacts are valid for non-conforming contact modes like tooth to tooth touch on gear drives or cam lobe to roller touch on valve train systems [5]. Piston and ring to cylinder wall contact is also considered as an EHD interaction.

Commonly accepted assumptions dictate that the oil film thickness is on the level of the height of the surface asperities for EHD lubrication.

2.3.3 Hydrodynamic Lubrication

Hydrodynamic lubrication is valid for the conforming contact geometries that show a uniform surface interaction. With the existence of relatively thick oil film between surface asperities which is above the total height of the two surfaces, the components operate in separated surfaces under only effect of hydrodynamic viscous forces.

As soon as the relative motion speed gets into higher levels, the viscous forces become dominant in surface dynamics and lead to increasing hydrodynamic friction forces.

The lubricants of these types need to have a sufficient level of viscosity so that they do not break down under high speed sliding or rolling conditions.

3. ENGINE LUBRICATION

Lubricants on a vehicle engine need to satisfy several unique requirements, which are maintained by the oil additive technology. Due to the nature of combustion process, the oil contamination and its effects on engine components need to be considered apart from other common applications of lubrication.

3.1 Functions of Lubricants

Main functions of the engine oils are;

3.1.1 Provide Good First Start

Friction is the primary barrier for motion. During first engine start even on very cold conditions, the oil should be thin enough to permit sufficient cranking speed [14]. After a successful start, oil delivered by oil pump should easily flow and quickly reach to critical engine components to minimize the start up wear. This is in general obtained by low viscosity oils.

3.1.2 Prevent Wear

When the engine reaches to steady state lubrication and temperature increases, the oil should not easily break down and should be thick enough to provide sufficient oil film thickness in order to reduce friction and wear.

3.1.3 Cool the Engine

Lubricants not only reduce wear but also aim to minimize friction hence lowering the heat generation between contacting surfaces. This provides a positive effect on the life of the components.

3.1.4 Sealing

Lubricants act as a sealing element especially in piston – cylinder interface. Rings surrounding the piston and cylinder are not capable of providing the sealing requirements needed to satisfy suitable combustion pressures.

Lubricants make an oil film between piston, ring and cylinder wall asperities acting as a sealant in parallel to wear and friction reduction properties [13].

3.1.5 Prevent Corrosion and Dust

Contacting metal surfaces may run into some corrosive attacks during operation or stand still conditions due to combustion based or environmental effects. Also the wear particles or combustion oriented deposits may accumulate on various parts of the engine leading to high size wear elements. Lubricant is to provide a corrosion resistant protection to engine components and also should maintain a homogeneous structure in oil in order to have suitable oil circulation.

3.2 Main Parameters

Main parameters of oils that characterize unique properties are viscosity, viscosity index, total base number (TBN) and total acid number (TAN).

3.2.1 Viscosity

Viscosity is defined as the resistance to flow. In tribology, it's a measure of the resistance of a fluid to shearing flow [5]. It's a measure of the internal friction of the chemical compounds to liquid flow. Lubricant viscosity is assigned to provide a sufficient oil film thickness so low viscosity oil may cause high friction and wear under heavy loaded conditions. Viscosity is also the basis classification of oils accepted by Society of Automotive Engineers (SAE).

3.2.2 Viscosity Index

Viscosity index is a measure of the lubricant to maintain its viscosity against temperature variation. The high index numbers show the oil's ability to resist the viscosity change when the temperature increases or decreases as shown on Figure 2.1. This parameter is important considering the challenging requirements from diesel engines and the severe running temperatures needed.

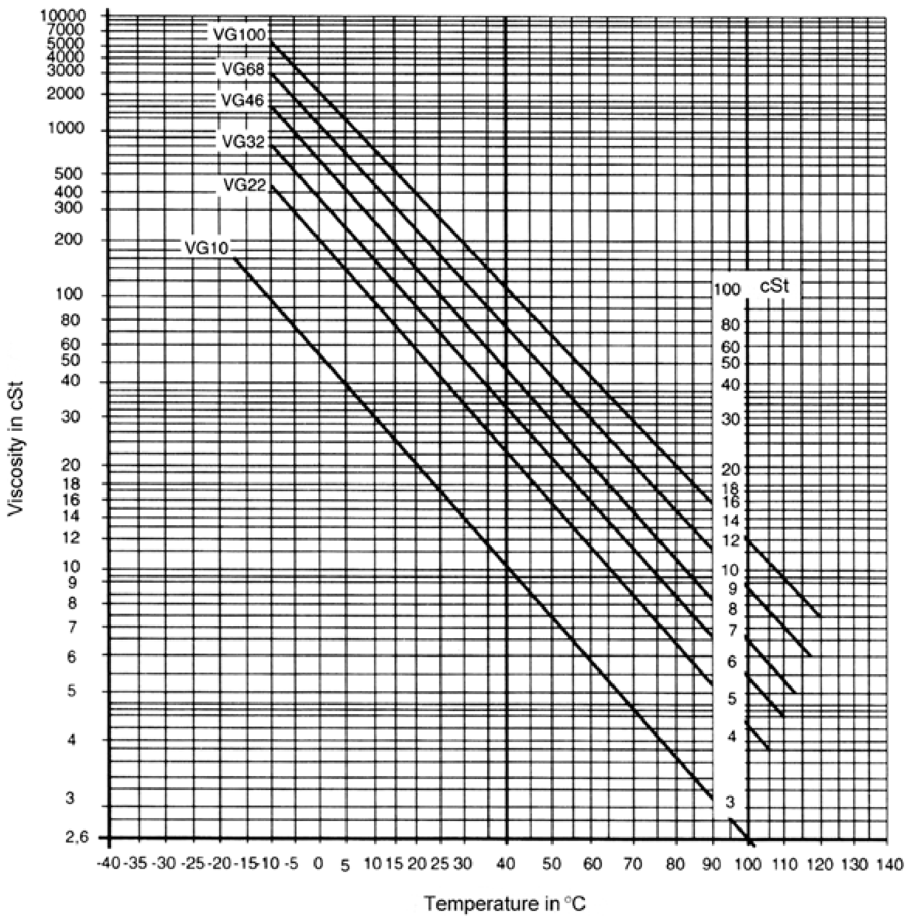


Figure 3.1 Viscosity variation with temperature for different viscosity indexes [15]

3.2.3 Total Base Number (TBN)

TBN is a measure of the perchloric acid level expressed in milligrams with the equivalent potassium hydroxide (KOH) that is able to neutralize the basic elements in one milligram oil [16]. TBN depletion rate is one of the most important parameters that represent the measure of oil service life indication. It's also expressed as the consumption of the alkaline additives in oil [17].

3.2.4 Total Acid Number (TAN)

TAN is a measure of potassium hydroxide in milligrams that is able to neutralize the acidic elements in one milligram oil [16]. Both TAN and TBN are expressed in mg KOH/g.

The excessive sulphur in the fuel after combustion process reaches to engine oil as deposits and undergoes a chemical reaction resulting with sulphuric acid (H_2SO_4) accumulation. H_2SO_4 increases the acidity of the oil hence reducing TBN level.

As per generic requirements of engine oils, once the TAN and TBN values gets crossed over, the oil degradation is complete and lubricant needs replacement. The only reason for the low oil drains in some applications is TAN and TBN crossover other than soot or other wear elements.

3.3 Oil Additives

Additives used in today's oil technology comply with the lubrication requirements of new engines. Here is a summary of the main additives and their roles.

3.3.1 Oxidation Inhibitors

Under high temperature, the oil goes into a chemical reaction with the air available in the environment. This increases the acidity of the oil hence increasing viscosity. Because of these chemical reactions, oxidation leads to varnish and lacquer to form on hot metal surfaces. The most widely available and commercially used anti-oxidant additive is zinc dialkyl dithiophosphate (ZDTP). [18]

3.3.2 Viscosity Index (VI) Improvers

These additives are used to improve resistance of the oil viscosity against change with variation in temperature. There are other advantages of VI improvers which are reducing fuel consumption, and increase of cold start performance. [18]

3.3.3 Dispersant and Detergents

Soot, combustion deposits and other particles existing in crankcase oil may collect and attach to various places inside engine. Dispersant additives act as a separator, break the elements stuck into each other down into small pieces providing low size particles, and hence reduce their wear effectiveness. Detergents provide a clean layer on metal surfaces and prevent residues to build up on engine components.

3.3.4 Antiwear Additives

These additives get in reaction with the asperities and form a surface layer that goes under plastic deformation to allow a new load distribution. Zinc dialkyl dithiophosphate (ZDTP) is also very effectively and widely used as an anti-wear additive [18].

3.3.5 Extreme Pressure (EP) Additives

EP additives are used to control the surface damage when two surfaces are under aggressive running conditions that need severe wear protection [18]. These additives also activate following a series of chemical reaction.

3.4 Crank-case Oil Structure and Classification

Within the complex structure of an engine lubrication system and considering today's expectations, engine designers need to select only one type of oil to drive the whole engine to show required performance and durability. Lubricant development suppliers have increased the technology available to present unique specification oils to support OEM's in engine development stage. These specifications are standardized by the organisations in automotive industry like Society of Automotive Engineers (SAE) or Association des Constructeurs de L'Automobile (ACEA).

In order to match with the different requirements, a various type and content of additives explained in the previous section are used with the base oil. There are four different base oil stocks available in automotive market today which are listed on Table 3.1.

Table 3.1 Base oil groups and their specifications

Group Number	Specification
Group 1	Conventional - Mineral oil derived from crude oil
Group 2	Hydro processed - Highly refined mineral oil
Group 3	Severe hydro processed – Semi synthetics
Group 4	Full synthetics (chemically derived)

Mineral oils are directly produced from crude oil, while semi synthetic oils use a mixture of both conventional and severe hydro processed base stock oils. Fully synthetic oils have a special base stock derived by chemical processes and do not contain any impurities which may exist in conventional or hydro processed oils. As the base stock derivation is application specific, fully synthetic oils have almost all the capability to meet today's engines' requirements.

Most of the internal combustion oil types available currently in the market have the following formulations [12];

- 70 to 95% of basic oil
- 5 to 20% of additives
- 0 to 20% of viscosity improver
- 0 to 15% of pour point depressants

SAE classifies the engine oils according to their viscosity index as shown on Table 3.2. In SAE classification, low temperature viscosity properties of oils are specified against cranking and pumping capabilities of oils under certain temperatures. These properties determine the first index of SAE representation. High temperature viscosity properties of oils are specified against kinematic viscosity and shear rate capabilities, at 100 °C and 150 °C consecutively.

Table 3.2 SAE classification of automotive engine oils [7]

SAE Viscosity Grade	Low Temperature Viscosities		High-Temperature Viscosities		
	Cranking (mPa.s)	Pumping (mPa.s)	Kinematic		High Shear Rate
	Temperature °C	Temperature °C	mm^2/s at 100°C		mm^2/s at 150°C
	Max	Max	Min	Max	Min
0W	6200 at -35	60 000 at -40	3.8	—	—
5W	6600 at -30	60 000 at -35	3.8	—	—
10W	7000 at -25	60 000 at -30	4.1	—	—
15W	7000 at -20	60 000 at -25	5.6	—	—
20W	9500 at -15	60 000 at -20	5.6	—	—
25W	13 000 at -10	60 000 at -15	9.3	—	—
20	—	—	5.6	<9.3	2.6
30	—	—	9.3	<12.5	2.9
40	—	—	12.5	<16.3	2.9
40	—	—	12.5	<16.3	3.7
50	—	—	16.3	<21.9	3.7
60	—	—	21.9	<26.1	3.7

American Petroleum Institute (API) and European institute ACEA characterize the engine oils with respect to their performance specifications. As reference, ACEA classification is shown on Table 3.3.

Table 3.3 ACEA classification of internal combustion engine oils [7]

ACEA categories	A/B – Petrol and diesel engine oils. C - Catalyst compatible oils
A1 / B1	Oils intended for use in petrol and diesel car and light commercial vehicles specifically capable of using low friction, low viscosity oils with high temperature / high shear characteristics.
A3 / B3	For use in high performance petrol and diesel cars and light commercials where extended drain intervals are specified by the vehicle manufacturer and / or for year-round use of low viscosity oils and / or for use in severe operating conditions as defined by the vehicle manufacturer.
A3 / B4	For use in high performance petrol and direct injection diesel engines. Also suitable for applications described under B3.
A5 / B5	For use at extended oil change intervals in high performance car and light commercial petrol and diesel engines designed for low viscosity oils.
C1, C2 and C3	For use in high performance car and light commercial petrol and diesel engines, with diesel particulate filter three-way catalyst and / or requiring low viscosity, low friction, and catalyst compatible oils.

By the recent enhancements in oil technology, the mineral oils have already been placed by the fully synthetic oils on passenger car engines. In early stages, fully synthetic oils had high costs to customer due to specialized production processes. Today’s technology provides latest development items on fully synthetic oils with affordable prices. However, semi-synthetic and mineral based oils are still on the shelf of oil markets due to improved characteristics of these oils coming from recent technology and adaptation from various types of additives.

3.5 Lubricated Engine Components

Internal combustion engines in principal produce kinetic energy from the heat dissipated after the combustion of fuels [22]. Combustion process and the following energy conversion can be performed by operation of a series of dynamic components. These components are subject to different type of contact mechanics and are under varying level of loading effects. In order to improve the wear on the contacting surfaces and increase service life of the engine components, specific lubricants are used. These lubricants operate under different regimes.

The systems and their components which are subject to tribological wear in an internal combustion engine are listed in Table 3.4.

Table 3.4 Engine sub-system components under lubricated wear

Engine Sub-System	Components
Crank train	Crankshaft main bearings, con rod small and big end bushings, piston-ring pack and cylinder walls
Valve train	Valve seat and insert, valve stem and guide, valve tip and rocker pad, cam lobe and rocker roller, camshaft bearings, hydraulic or mechanical lash adjusters
Timing drive	Gears, chain, sprocket, bearings
Turbocharger	Impeller and bearings

Engine oil is first of all sucked from the sump by a rotor or disk type oil pump which is driven by crankshaft. Then oil is pumped to oil filter and goes inside an oil cooler which cools the oil using the circuit of the engine cooling system. Sufficiently cooled down oil is then released to several points on the circuit by pressurized oil galleries available on cylinder block and cylinder head. After oil is driven by engine running components, returned into the sump passing through the drainage galleries and connections on the base engine. These components are shown by an illustration on Figure 3.2.

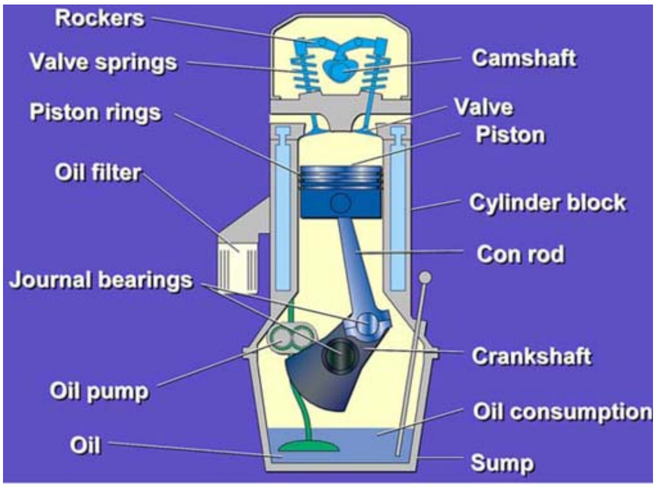


Figure 3.2 Lubricated components of an internal combustion engine [4]

It's difficult to distinguish the components of an engine with respect to the lubrication regimes active on the sub systems. Various components have different contact geometries and unique operational conditions. Some parts of an engine are under sliding contact like pistons and others are under rolling contacts like bearings. All these distinct systems are under separate temperature, loading direction and level, angular and linear velocity conditions. Therefore, each system is considered as a separate unit and unique verification tests are carried out.

Engine bearings are subject to hydrodynamic lubrication in internal combustion engines. Figure 3.3 demonstrates the Stribeck diagram that includes the engine sub-systems and active lubrication regimes.

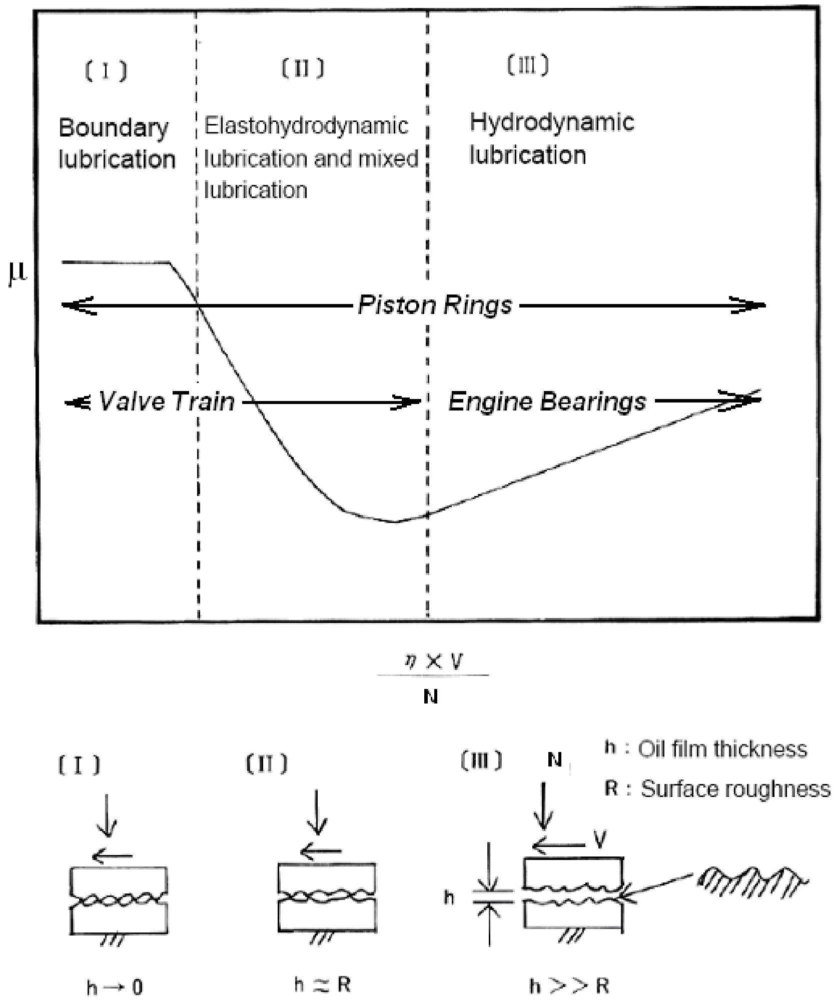


Figure 3.3 Stribeck curve for engine sub-systems [21]

3.6 Engine Timing Chain Wear and Crankcase Oil Contamination

As per previous investigations performed in chain wear and elongation, pin and bush wear contact interface is known as the key factor for wear elongation. Rolling and sliding friction forces between chain pin and bush are acting effectively on the friction and wear properties of timing chains.

However, it’s difficult to make predictions on timing drive chain wear with the active forces on the system. This is due to the complicated tribological condition of wear process. Apart from the forces and dynamic effects, there are many other factors that join into the wear of running parts. This is especially driven by the combustion in diesel engines which produce soot and change other lubrication parameters. A flow chart that describes the effects of diesel engine combustion and lubricant contamination is shown on Figure 3.4.

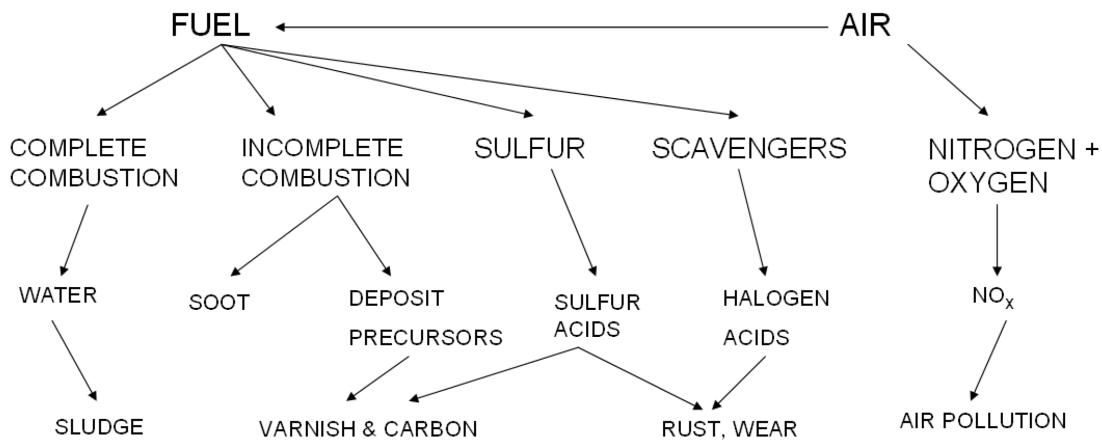


Figure 3.4 Crankcase oil contaminations by fuel and air [7]

The wear of the engine components are also affected by the various elements that exists in lubrication or accumulating during engine operation. The level of metal particles may increase due to wear of particular engine components like Al increase due to piston wear. The increase of these types of elements in lubrication may also have a negative effect on wear of the other engine components. Therefore, the combustion and contaminants influence on lubrication and wear process is still a part of serious investigations in engine technology, and hence engine developers need to provide adequate running conditions to their engine technology in terms of design and lubrication which is currently specific to application. There are numerous type of engine oils available in the market, but the appropriate selection usually needs to be supported with extra testing activities to improve lubrication performance with necessary additives and their feed back to design.

The recent stretch changes in automotive emission legislations have led to important design changes in automotive engines. One of these important emission parameters is NO_X. Diesel engines have been forced to adopt exhaust gas recirculation (EGR) systems to minimize the NO_X. EGR usage in diesel engines resulted with a significant increase in particulate matter emissions. In general means, this has led to negative effect on combustion and soot accumulation performances. Soot formation, rheology and morphology gets into a more aggressive form which increases the adverse effects of soot on wear [2].

To overcome the negative effects of contaminants and soot on wear, lubricant have been improved by various type of additives. The most widely used anti-wear agent ZDTP has been found to provide better wear characteristics by forming an anti-wear surface films on contacting surfaces [18]. The dispersant and detergent additives also found to have positive effect on wear reduction. Various numbers of elements (Phosphorus, Sulphur, and Zinc) available in base oil also tested in high and low levels with different tribometers. In some testing environments, Phosphorus found to have better effect in wear performance, while Sulphur and Zinc increase put adverse effects on wear [2]. However, all these studies are specific to application and testing boundaries, therefore a common approach to wear improvement is still a part of unique development stage.

4. WEAR TESTING METHODS

As the wear properties of each engine component is one of the key parameters on an internal combustion engine design, a proper wear testing during design verification stage is critical for the product reliability. The wear of the dynamic components on an automobile engine are closely affected by the loading, speed, temperature and lubrication conditions. Therefore the new designs that will be a part of tribological investigations for the rest of engine development are first of all subjected to computer aided engineering (CAE) programs. With the recent improvements in accuracy, the computer based investigations provide a wide range of information to the designer. This information may consist of the very useful data like oil film thickness and pressure, lubrication regime and friction losses valid for engineering applications. The simulation program is key for reduction of the number of iterations and hence cost needed during engine tests.

Despite the close correlation of the results achieved on CAE programs, computer simulation may not be always sufficient to simulate all the boundary conditions and operating parameters affecting the real contact environment. It's not always easy to apply surface engineering properties of for example the different type of coating or surface roughness on software. Bench testing is therefore still the most important tool for the design verification phase of engine development.

4.1 Bench Tests

A proper representation of the real life system dynamics is clearly crucial for a convenient bench test performance. There are many types of generic bench tests available for basic tribological investigations, but all have some limitations to meet specific running conditions of distinct applications. Therefore application specific test rigs are built in order to simulate real life conditions as close as possible.

4.1.1 Special Test Rigs

As a part of the successful representation of the tribological system, geometrical conformance is very important as it directly simulates the surface kinematics [4].

Apart from geometrical representation, the operating conditions specific to the application naturally have considerable impact on the robust wear simulation. The bench test environment should be able to provide a form of the real running environment by successfully controlling the speed, lubrication and loading conditions [5].

The designed bench tests need to have a stable operational control which is important for the reliability of the wear performance simulation. Test set up should be easily manageable by different operators and the same results should be achievable within random iterations performed by the same operator. In statistical terminology this is called repeatability & reproducibility (R&R) and the assessment methods concerning this property is called the Measurement System Analysis (MSA).

The general purpose of the bench tests are the accelerated measurement and monitoring of the wear and friction on a single tribological system. The test set up should be able to provide frequent wear measurement ability to the test operator and it should have the necessary instrumentation to monitor the friction force directly or indirectly. Friction force and friction coefficient can be either directly measured by the enabler software & program adaptation to the measurement system or can be driven by the measured loads acting on the tested specimen. [6]

There are many types of standard wear testing devices which are usually used by engineers working on engine and lubrication engineering. These typical testing devices are called tribometers and widely available to usage in different applications.

4.1.2 Tribometers

Tribometers are used for general tribological investigations as almost standard and a number of these test rigs is specified by American Society for Testing and Materials (ASTM). These tests provide good control ability in applied load, speed, humidity and temperature therefore is useful for representation of typical surface investigations [6].

Tribometers enable the test engineer to control the key parameters of the system. Commonly used tribometers are pin-on-disk, pin-on-cylinder, pin-on-flat and rotating four-ball.

Pin-on-Disk Tests

Pin on disk test consists of a rotating disk and a static pin as shown on Figure 4.1. Load is acted to the system directing onto pin by a dead-weight or a more complicated mechanism like hydro-static or magnetic systems [11].

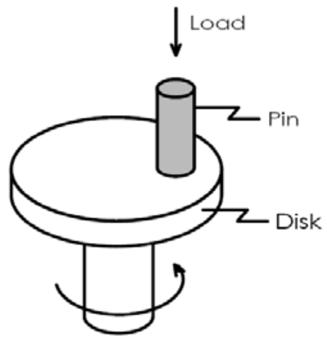


Figure 4.1 Pin-on-Disk tribometer [11]

Pin can be a flat face cylinder or rectangular, and ball or a spherical face cylinder. Different material, hardness and surface specification selections can be tested under similar or different running conditions in order to make optimization on wear performance.

Pin-on-Flat Tests

Pin on flat tests as shown on Figure 4.2 are similar to pin on disk tests. The only difference is on the mode of sliding motion which is linear instead of rotational motion of the flat. One of pin or flat is stationary and the active component makes reciprocating motion in certain frequency to create wear.

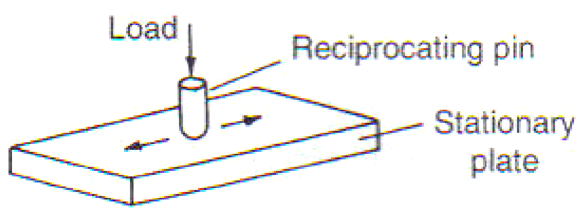


Figure 4.2 Pin-on-Flat tribometer [6]

Pin-on-Cylinder Tests

Pin is stationary and cylinder makes rotational motion during the load acted onto the pin normal to the rotation axis of the cylinder. A schematic illustration of pin on cylinder tests is shown on Figure 4.3.

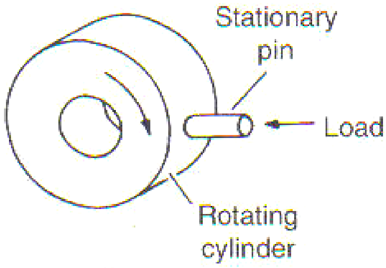


Figure 4.3 Pin-on-Cylinder tribometer [6]

Rotating Four-Ball Tests

Four-ball wear test contains four spheres - three stationary on the bottom and one sphere is rotating on the top of these three balls in contact condition, as shown on Figure 4.4. The rotation speed of the top ball is under control as well as the load acting onto the system from the bottom plate on a certain level.

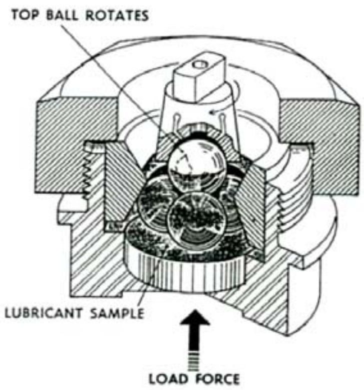


Figure 4.4 Four-Balls tribometer (according to ASTM D2783 standard) [12]

4.2 Dynamometer Tests

Dynamometer tests are used for accelerated durability assessment of internal combustion engine and particular sub-systems.

Piston-ring pack and cylinder walls, crankshaft and camshaft bearings, con rod big and small end bearings, valve to valve seat and guide interface, mechanical or hydraulic lash adjusters on valve train and timing drive sub components (gears, chain elements, sprockets) are the components most suffering from wear attack in diesel engines.

There are also several operational factors affecting the wear characteristics on combustion engines other than the lubrication specific parameters. If any of these factors which determine operating conditions exceeds the specifications suitable for

engine system, this will lead to unacceptable degradation or functional losses on the above vital components. This will certainly result with reduced service life that may even lead to catastrophic failures need replacement and increased maintenance costs. [13]. Some of these engine specific operational parameters can be summarized as follows;

1- Customer usage effects (high or low engine speed selection, urban or autobahn drive)
2- Combustion efficiency hence cylinder temperatures effecting heat transfer
3- Coolant and lubricant temperatures
4- Lubrication filtering system efficiency
5- Dynamic behaviour of running components
6- Air induction system filtering efficiency
7- HC emission in diesel engines

These parameters determine the level of conformity of the engine overall running behaviour hence effect the operation conditions of sub system components whether within or out of specified boundaries. Crank train components mainly which are crankshaft, con rod, pistons and rings, and cylinder walls of the block are generally under high loading and temperature conditions. This system is generally worn when the engine is forced to use steadily the maximum level of the torque curve.

Engine manufacturer design and test engineers cooperate to develop a unique test cycle to assess the crank train system components. Specific tests may be either bench based or a dynamometer/vehicle based tests. Valve train system is another important part of an engine, which contains valves, springs, camshafts, lash adjusters. On valve train elements, high speed contacts are dominant therefore durability performance assessment is focused onto engine high speed operation.

Engine designer companies have their own design verification and testing methods which have been built after years of engine development and testing experience.

Dynamometer tests present the opportunity of accelerated durability assessment of internal combustion engines.

Engine tests are performed with complicated cycles in order to simulate the customer usage profile. Particular tests are dedicated for evaluation of a single sub-system, therefore are designed to simulate most damaging running conditions for quick system checks.

5. TIMING DRIVE SYSTEM IN INTERNAL COMBUSTION ENGINES

The basic function of timing drive systems on internal combustion engines is to maintain the engine timing by providing torque transfer from crankshaft to camshaft(s) for proper operation of the engine valves. Therefore timing drive systems are subject to loading of interactions from driven sources like crank train and valve train system. There are different technologies for transfer of motion determined against durability and functionality requirements.

5.1 Types of Timing Drive Systems

Torque transfer from crankshaft to camshaft(s) is mainly performed by belt, gear or chain drives. Selection for the type of rotational motion transfer is driven by engine package, subjected loads, service life target, NVH expectations and costs of applications.

5.1.1 Belt Drive System

Belt drive is generally used on gasoline engines and relatively low volume diesel engines. For belt timing drives, toothed belt drive usage is a standard as it provides no slip characteristics.

Belt drives have comparably lower ultimate tensile strength and hence reduced fatigue limits compared with chain drives. Due to this fact, they need frequent change in maintenance periods. The limited fatigue levels and frequent service replacement are the main reasons for limited usage in diesel engines. In small size diesel engines, belt drives have been almost common and there are also some light duty vehicle engines using belt drive on diesel machines for extended replacement periods. Belt drives are commonly designed for dry environments, but recent technology presented belt drives running in typical engine lubrication environment.

Main components of belt drives are toothed pulleys and tensioners, as shown on Figure 5.1. According to the design layout and package constraints, idlers also may be added to the belt drive systems.

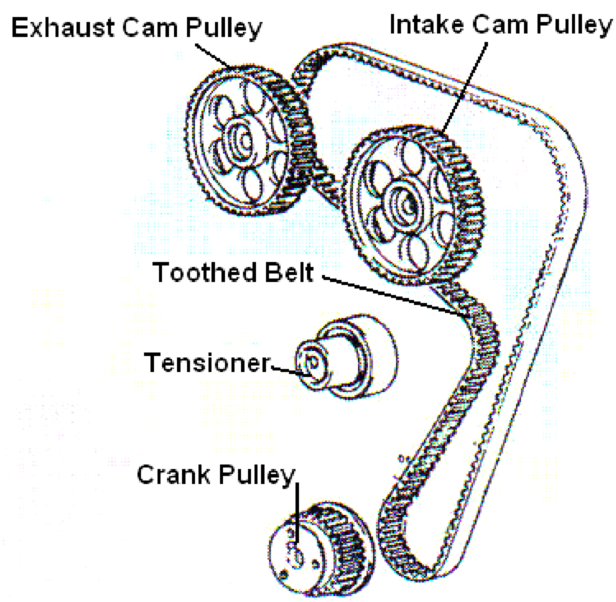


Figure 5.1 Components of belt drive systems [23]

Belt width and thickness determine the fatigue strength of the belt so application specific design optimizations are performed by design engineers. Belts need replacement before engine timing slip that may occur due to belt tooth crack or fatigue failures of belt that may lead to engine catastrophic failures.

5.1.2 Gear Drive System

Gear drives are almost an industry standard for heavy duty diesel engines, due to their very high load carrying capabilities. Under heavy loading conditions, the conventional drive systems like chain or belt circuit is not feasible due to their limited tensile strength.

A gear drive system consists of main gears and an idler gear. Main gears are the gears that observe the rotational motion from crankshaft and transfer this motion to camshaft(s) by the camshaft gears. As naturally specific to application, idler gears exist on the gear drives that are assigned only to torque transfer between main gears. A representation of gear train systems can be seen on Figure 5.2.

The size and thickness determine the load carrying capabilities of the gears. The shape and size of the gear contact area effects the gear wear properties. As gear drives operate under almost boundary lubrication, gear lubricants need to have a sufficient viscosity to withstand under heavy contact loads in order to reduce tooth wear.

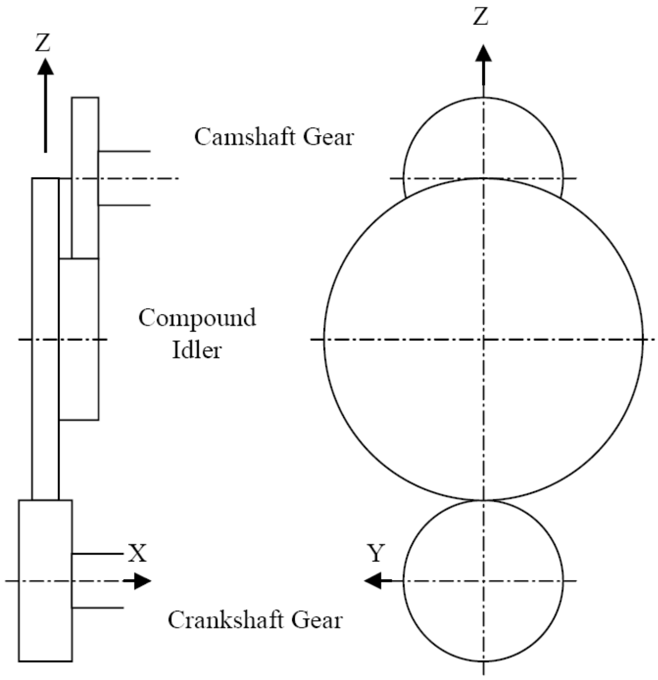


Figure 5.2 Components of gear drive systems [24]

5.1.3 Chain Drive System

Chain drives are commonly used in commercial vehicle engines due to low maintenance costs. Special design chains are always valid for specific applications but there are three different types of chain drives available in today's automotive market which are bush, roller and inverted tooth chains.

Chain drives include sprockets, routing guides and tensioners as part of their generic layouts. Guides and tensioners are used in order to maintain the tightness of the chain under any dynamic condition preventing vibration and dynamic resonances. Sprocket teeth get engaged to the chain pitches for maintaining the transfer torque and rotational motion input from crankshaft to camshafts and other adjacent drive systems.

In general applications, bush and roller chains are a combination of bushings, pins, links and/or rollers, as shown on Figure 5.3. Pins are subject to shearing and bending forces transmitted by the bushes and plates. They need to have sufficient bending and shearing strength, as well as bushes, in order to keep durability under sprocket to chain engagement forces. In contact with bushes, pins make rolling and sliding motions in a certain tolerance, therefore both components also need to have sufficient hardness and wear resistance. Considering the clearance between these components and active impact motions under general dynamic behaviour of chain, these components also need to have sufficient resistance against impact wear and fatigue failures.

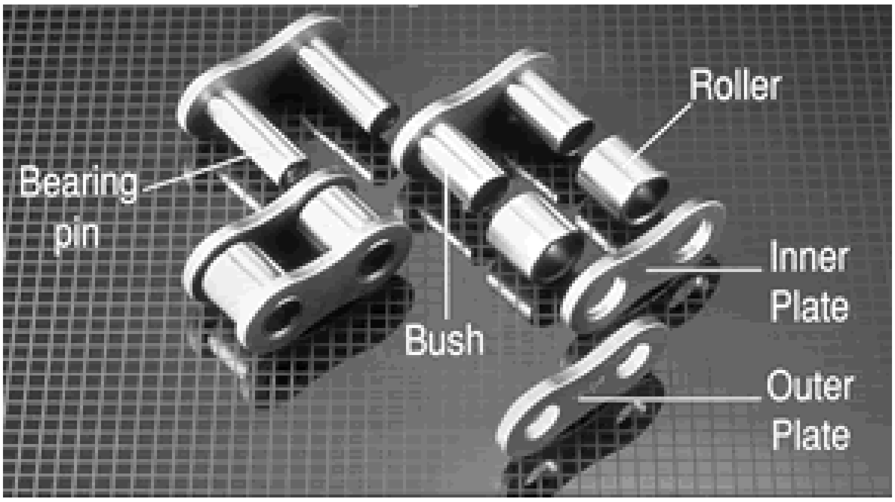


Figure 5.3 Components of chain [25]

Chain inner and outer links provide the linkage between these components and are in contact with sprocket teeth inner and outer surfaces. Therefore they need to show sufficient surface uniformity to minimize teeth and link surface wear which may cause increased axial clearance on the chain assembly. Extreme axial clearances induce excessive misalignment and dynamic irregularities on the chain leading to non-uniform contact patterns on the pins and bushings despite a limited level of misalignment is already given to the system in order to reduce axial loading of the chain. Links also are subject to shearing and bending forces coming from bushings and pins therefore need to have enough strength. They should have good surface roughness on the bottom width which is contact with the plastic guides of the driving system, in order to reduce wear depth on plastic.

Rollers are used in roller chains. They get in impact contact with sprocket therefore needs to have resistance to impact fatigue. The basic function of rollers is to convert the sliding motion between the bushing and sprocket teeth to rolling contact as the rollers are free to rotate over bushings. This leads to reduced friction and wear. However, roller and bushing contact interface is expected to overcome the rolling friction and wear, therefore rollers need to have enough hardness and material properties. Due to roller's existence in the chain package as shown in Figure 5.4, there is much lower space for bushing and pin interface; therefore roller chain's pin and bushings are generally smaller in size compared to bushing chains. This reduces the fatigue strength of the chain, hence roller types are mostly used in the applications subject to lower loads like auxiliary drive systems on engines (e.g. oil pump drives). Small size pin and bush reduces the contact area between pin and bushing increasing contact pressure between components. Increase in contact force per unit area has a negative effect on friction and wear properties of timing chains.

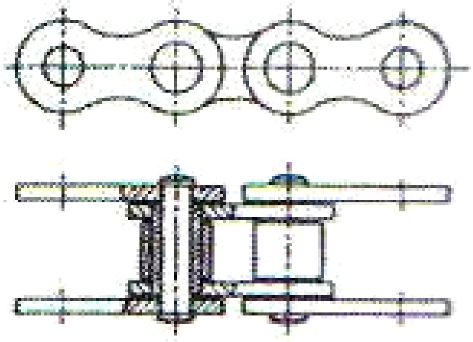


Figure 5.4 Section of roller chain assembly [25]

Bushing chains are the most widely used type of chain especially common on diesel applications. The simple chain package presents many different types of sizes and standards easily adaptable to timing drive layouts. As the size of the pins and bushings may be increased, this fact provides the opportunity to design low grade material and surface coatings on these components. This also gives the flexibility of usage under high loading applications, as the desired fatigue resistance may be adjusted with design optimization on the chain. An illustration of bush chain can be seen on Figure 5.5.

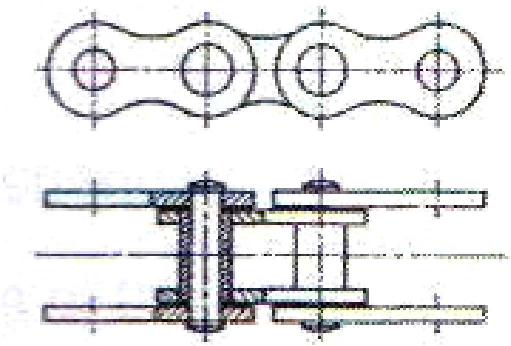


Figure 5.5 Section of bushing chain assembly [25]

During engagement of both bushing and roller chains to sprockets, chordal action is active. This occurs due to the difference of the relative speeds between chain and sprockets, causing chain hit the sprocket and fluctuation in the motion of the chain. The increase in number of teeth on sprockets reduces the impact energy and resonance condition due to the reduced relative engagement motion of the chain. This is also known as polygonal effect shown on Figure 5.6 which is source of increased chain span vibration and noise excitation from bushing and roller chain drives [26].

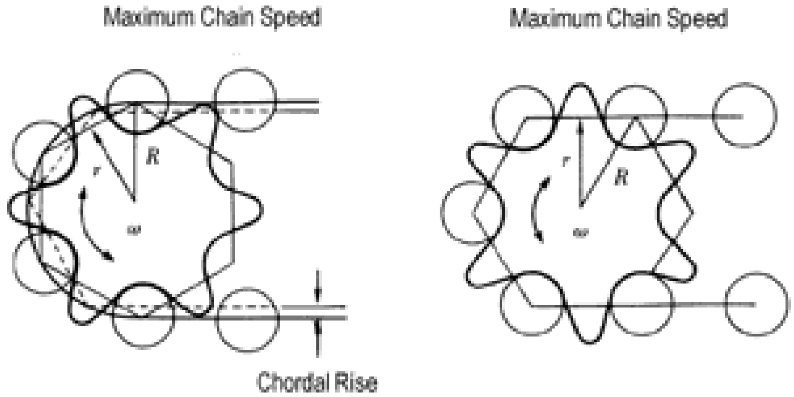


Figure 5.6 Polygonal effect on chain drive system [25]

Silent chain or inverted chain is used on the applications where reduced noise, vibration and harshness characteristics are expected. These chains include link plates, guide plates and pins making it relatively simple design. Guide plates provide guiding on the sprockets while link plates are subject to tension and engage the sprockets. The elements of silent chain are shown on Figure 5.7.

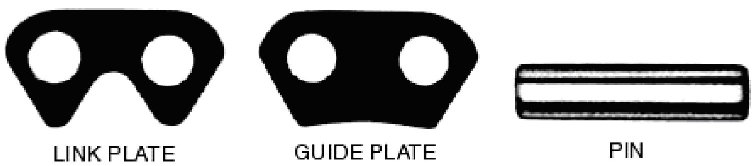


Figure 5.7 Components of an inverted tooth chain [25]

As link plates contact to the sprocket teeth at a defined angle, the impact energy is comparably low on silent chains compared to roller and bush chains. This is the main reason for optimum noise behaviour of the inverted tooth type chains and reduces the polygonal action and its effects on NVH characteristics.

5.2 Chain Timing Drive Dynamics

As similar to either belt or gear drives, chain drive systems are in interaction to other important sub-systems of the internal combustion engines. A generic timing chain drive and adjacent systems are shown on Figure 5.8

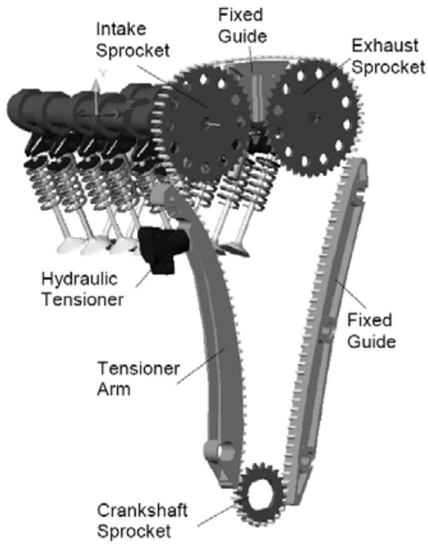


Figure 5.8 Timing chain drive and adjacent systems [27]

As its basic function is to rotate the camshafts in synchronization to crankshaft, the chain drive system is mainly challenging with the loads and torsional vibrations coming from crank train and valve train systems.

Crank train system includes the most famous internal combustion engine components which are crankshaft, con rods and piston assemblies, as shown on Figure 5.9

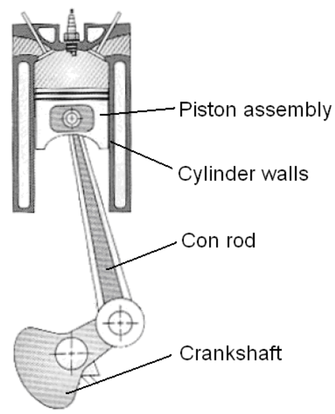


Figure 5.9 Components of crank train system [28]

During engine operation, the piston assembly performs a linear motion. This linear motion is converted to a rotational motion and is transferred to crankshaft by con rod. Transfer of the motion is achieved by the excitation of the combustion process which induces the loading and torsional vibration of the components. The crankshaft counter weights, pistons and con rods exert loads onto the chain drive. While combustion forces affect the level of loading, the combustion order and the number of cylinders available on the engine determine the torsional vibrations combined with the inertial characteristics of crank train system.

On the other side, valve train system consists of camshaft/s, rocker arms, tappets, retainers, valves, valve springs, and other adjacent elements. Camshafts directly actuate the tappets and valves in direct drive systems. In finger follower systems, the camshafts excite the rocker arms and hence the engine valves. Camshaft excitation, valve seating and valve spring forces all contribute the dynamic loads acting on the chain that drives the sprockets ob the camshafts. A direct drive cam-tappet valve train system is shown on Figure 5.10.

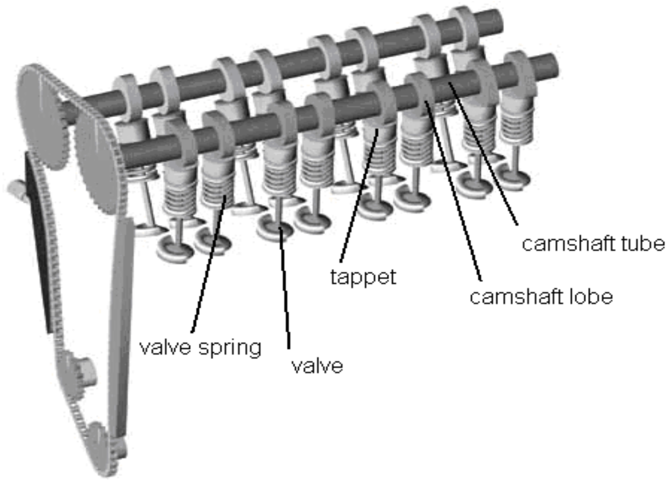


Figure 5.10 Components of valve train system [29]

As it is dependent on the engine and timing drive design architecture and package, the chain drives may also have a secondary function of driving other accessory components. These components may either be a tensioner sprocket that provides the required tension defined on the system layout or may be one of the pumps that need to be driven by an external rotational motion like fuel or oil pumps. These kind of accessory pumps also put extra loading and speed irregularities on to the timing drive chains.

Typical chain drive systems are composed of fixed and tensioning guides, camshaft and crankshaft sprockets, tensioner (mechanical or hydraulic) and chain, as shown on Figure 5.11.

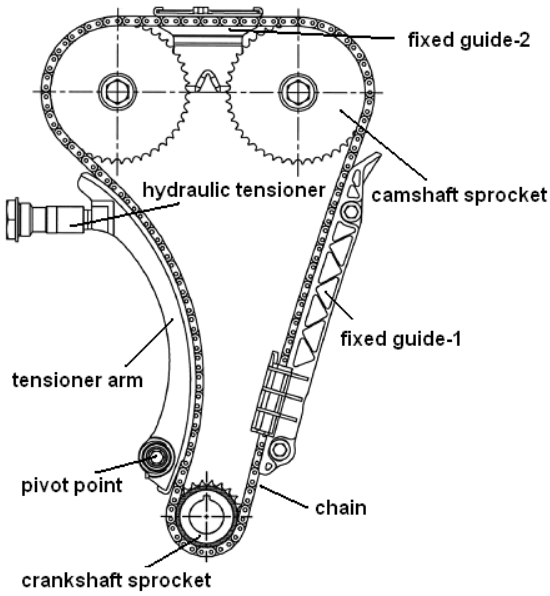


Figure 5.11 Illustration of timing chain drive system and its components [26]

Guides and tensioner arm with their geometries and locations operate as guidance to the chain that needs to follow a predefined driving circuit. There are also various criteria that these components need to be adopted during design phase. They maintain the chain drive dynamic stability on all operating conditions.

Tensioner's main function is to provide the required tension on the chain. It can be either mechanical or hydraulic. Mechanical tensioners generally have a laminar or circular spring that maintains the dynamic stability and also provide sufficient take up on elongation of the chain, under either dynamic or plastic (wear) stretch conditions. However, hydraulic tensioners are widely used on timing drive chain systems. They operate under the oil pressure observed from pressurized oil galleries.

Timing chain layout and the design of the guides and tensioner is critical for chain drive dynamic stability, desired vibration characteristics and durability.

Camshaft and crankshaft sprockets need to be designed so that it exerts the optimum loading and inertial characteristics on the chain drive. In order to achieve this, sprockets teeth profiles need proper optimization for dynamic compatibility, NVH and wear performances. They also need to be weight optimized for inertial effects.

An engine designer considers the above mentioned sources of interactions as a package. The design package and analytical investigations are the first steps in timing drive design. This also is framed with general design considerations and design specifications. With the recent improvements in computational technologies, simulation has been an indispensable part of engine development. There are very capable software's available in the market in use of automotive manufacturers. Today's timing drive systems may completely be simulated in these software's, which significantly reduces the time and cost of engine development. With the help of CAE programmes, many important operational parameters can be calculated and used confidently in the next phases of process which are real life engine tests.

These programmes use various types of inputs concerning the main items of timing drives as shown on Table 5.1

Table 5.1 List of input for chain timing drive simulations

Component	Input Type	Unit
Sprockets	Camshaft sprocket inertia	$kg.mm^2$
	Crankshaft sprocket inertia	$kg.mm^2$
	Tooth profile dimensions	mm
	Sprocket tooth contact stiffness	N/mm
	Sprocket tooth contact damping	N.s/m
	Tooth friction damping	N.s/m
Tensioner	Tensioner contact stiffness	N/mm
	Tensioner contact damping	N.s/m
	Tensioner inertia	$kg.mm^2$
	Tensioner piston mass	kg
	Piston to tensioner stiffness	N/mm
	Piston to tensioner damping	N.s/m
Chain	Chain stiffness	N/mm
	Chain damping	N.s/m
	Link mass	kg
	Link length	mm
	Bush Diameter	mm
Guide	Guide support stiffness	N/mm
	Guide to chain damping	N.s/m

The layout of chain drive systems need to be modelled in every small detail in these programmes and should be supported with the correct input data either calculated separately or observed from single evaluations (tests, previous experience, etc.).

After a proper model built, the CAE programs are able to provide dynamic simulation that can be investigated either visually or numerically. A snapshot from the CAE simulation can be seen in Figure 5.12 of the 4 cylinder diesel engine timing chain drive.

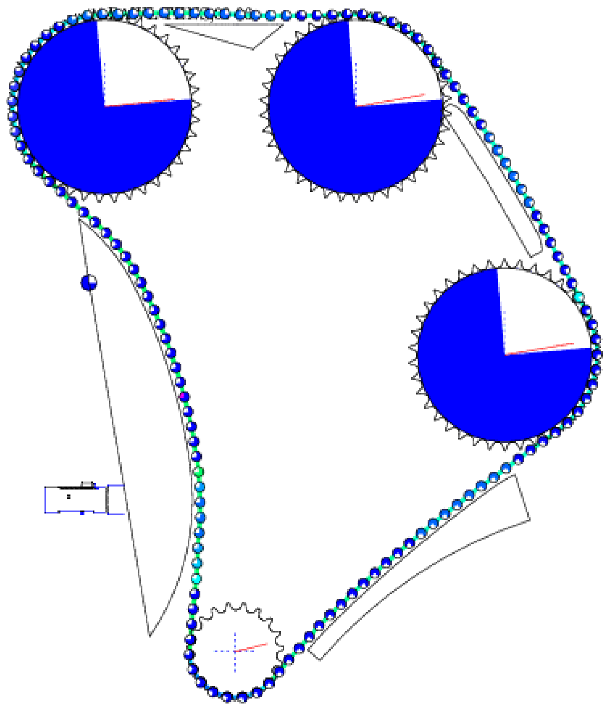


Figure 5.12 Animation of view of timing chain drive simulation programme [31]

The simple theory in chain dynamic modeling is the investigation of each single contact as basic damping and stiffness elements. Each chain link is modeled as rigid body in the plane with 3 degrees of freedom, which is one rotation, and two displacements, where the link center of gravity may vary along the axis of chain link as shown on Figure 5.13 [32].

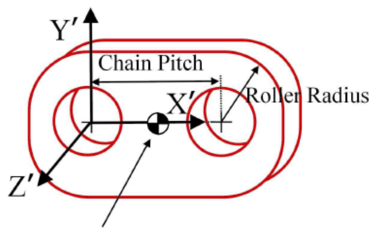


Figure 5.13 Representation of the chain link [32].

Chain link is connected to adjacent pitches by means of linear springs and dampers as shown in Figure 5.14. The points P_1 and P_2 determine the location of the roller centers. Each spring and damper element is connected to the N_{th} link at point P_2^N and to the following one, $N_{th}+1$, at point P_1^{N+1}. The chain internal stiffness and damping characteristics are specified by these spring and damper elements.

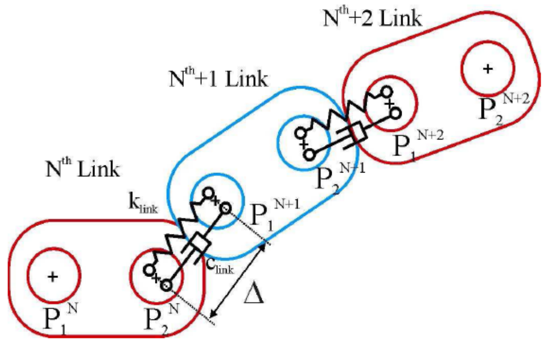


Figure 5.14 Stiffness and damping elements on chain modeling [32].

The basic chain link force that occurs between chain elements are described with following formula;

$$F_{link} = k_{link}.\Delta + c_{link}.\dot{\Delta} \tag{5.1}$$

where k_{link} and c_{link} are the chain link stiffness and damping coefficients, respectively; Δ is the relative displacement between the geometric centers of each chain bush and $(\dot{\ })$ denotes the derivative with respect to time.

Each tooth profile is formed by means of circular arcs in different radii. The pitch dimension of tooth profile determines the main standard of chain drive application. Respectively, other key dimensions of the tooth profile are selected as base inputs from the range determined by ISO tolerances, and optimized through simulation response. Sprockets are also modeled as a single degree of freedom (rotational) or as a rigid body in the plane with two displacements and a rotation.

In chain drive dynamic simulation, contact forces between chain and sprocket are also one of the important design parameters that affect the chain dynamic behavior and durability. The contact forces that build up during chain elements are passing through the sprocket elements are evaluated by following formulas,

$$F_x = k.x + c.\dot{x} \tag{5.2}$$

$$F_y = k.y + c.\dot{y} \tag{5.3}$$

where F_x and F_y are the forces in X and Y directions, as shown in Figure 5.15. Here, k and c represent the bushing to sprocket contact stiffness and damping coefficients, x and y are the relative displacements between the sprocket geometric center and the ground in the X and Y directions. Similarly, ($\dot{}$) denotes the derivative with respect to time.

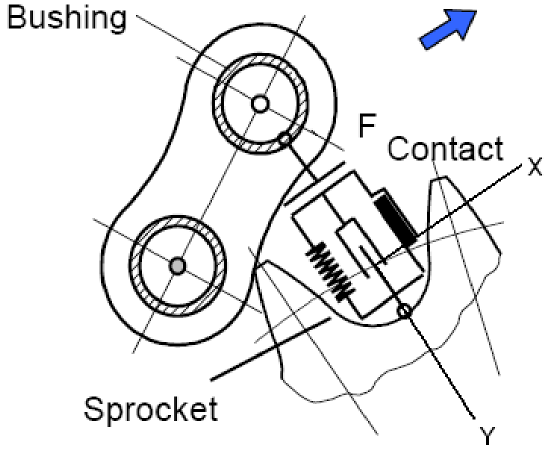


Figure 5.15 Representation of the chain bushing to sprocket contact [27]

Guides are also modeled as a rigid body in association to a pre-defined curve. Rigid bodies may either be fixed or in connection to a tensioner [32]. The contact force to guide is also defined with a similar approach, with spring and damper mass element between chain link and guide contour, as shown in Figure 5.16.

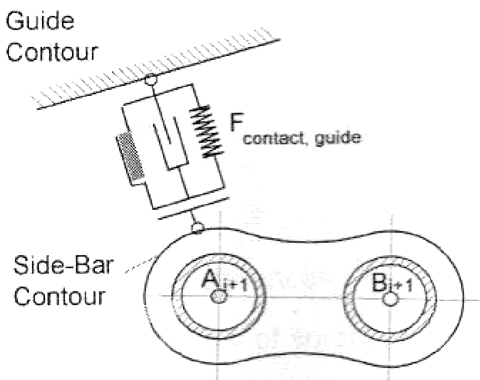


Figure 5.16 Representation of the chain bushing to guide contact [23]

The following important outputs can be obtained from CAE simulation that is key to understanding the system dynamics and the forces acting components. ;

- Chain link load

- Chain to sprocket contact forces
- Chain to guide contact forces
- Chain pin to bushing contact forces
- Tensioner motion
- Sprocket displacements

The above system responses provide the necessary information to designer about the selected design parameters and existing safety factors with respect to components durability limits.

There are many other type of outputs may be observed from simulation but in terms of wear interaction, the most significant feedback is achieved by chain link load, chain to sprocket contact forces and chain pin to bushing contact forces. The timing drive chain wear and hence elongation is mostly dependant on the pin to bushing interaction, where the forces built between pin and bush is one of the main parameters which effect the level of contact stress. This is clearly having an effect on the lubrication regime and the level of friction coefficient between pin and bush. The contact dynamics between pin, bush and also adjacent components is important in wear process. The type of motion and forces active on a drive chain is clearly shown on Figure 5.17.

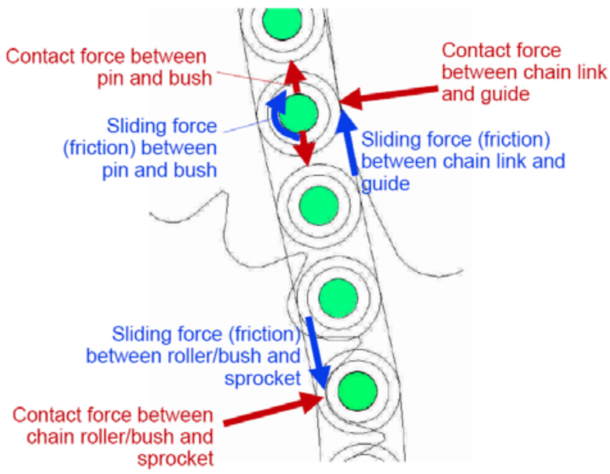


Figure 5.17 Active forces on chain drive system [33]

As it's defined by the driving circuit, the chain pin and bush makes both rolling and sliding motion relative each other. This case generates a complicated dynamic motion between these elements and hence produces a contact force on the elastic region between chain pin and bush. As a result of relative motion, there is also an active sliding force and friction. Similar to this case, sliding and contact forces build in connection of chain side links to guide and chain bush/roller to sprocket teeth. These forces directly affect the chain dynamic behaviour and its NVH characteristics. The durability against fatigue and wear is also closely dependant on the level of the forces acting on each single component.

6. PROJECT SCOPE

6.1 Background

During the development stage of an engine, three parallel validation stages are followed after analytical work completed. These can be listed as component level, sub-system level and engine level validation tests. Each sub-system and its components are determined in a separate way of testing, which is finalized by the whole engine and vehicle confirmation tests. According to this process, timing drive system components need to be applied to specific component development tests. In this validation plan, the chain is subjected to loaded durability rig tests to assess its fatigue limit and failure performance. For its material, hardness, coating and hence wear performance evaluation, simple accelerated wear rig tests are performed during design stage of chain.

6.2 Rig Test Details

As for specific testing applications in tribological investigations, wear elongation rig tests are nearly an industrial standard for chain performance testing. However, there is no single way of chain wear elongation tests; all applications are specific to methods followed by chain suppliers and engine development companies. The general concept of chain wear elongation rig tests are shown in Figure 5.1.

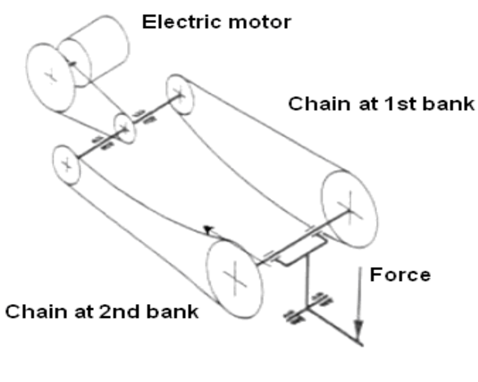


Figure 6.1 Layout of chain wear elongation test rig

One electric motor is used to drive one of the shafts of the chain test rig. Two parallel chain banks are generally used to maintain the required load balance between chains and also provide the opportunity to have two results of chain wear performance in one test completion. The shaft that is directly driven by electric motor is called the driving shaft. The shaft that holds the corresponding sprockets on the other side of the rig is subject to a normal force which keeps the chain under a certain level of tension, and is called driven shaft. The loading and its application to shafts and hence to the chain is sustained by a series of sliding bearings, which allows the shaft to make displacement under chain dynamic stretch and wear elongation. It's also important that the tension on the chain can be controlled accurately so that unacceptable changes or fluctuations do not exist in tested loads.

The number of teeth on sprockets may either be equivalent or different, specific to the selection of testing application. The size of the sprockets is important because the number of teeth is determining the linear speed level of chain and hence the level of relative motion.

The driven sprocket should be rotated against a resisting torque for a chain drive rig to transmit power. One span of the chain layout should be slack so that the torque transfer is performed through the tight span only.

A close alignment should be maintained between sprockets in order to provide a smooth running circuit for chain on the rig. On all chain drive systems and the rig test, a small amount of misalignment is given to the sprockets allowing the chain itself to find its running route. This positively effects chain dynamics and also reduces the axial load acting on the chain, due to the relatively low axial load carrying capability of bushing and roller type of chains.

Chain linear speed on a wear elongation test rig is expressed with following formula,

$$V = \frac{\pi..n.D}{60} = \frac{p.N.n}{60} \tag{6.1}$$

where V is in m/s, n is the sprocket rotational speed in rpm, D is the sprocket pitch diameter in m, p is the chain pitch dimension in m and N is the number of teeth on sprocket. The minimum power of the electric motor needed to drive the chain test rig is calculated by

$$P = \frac{T.V}{1000} \tag{6.2}$$

where P is in kilowatt, T is the chain tension in N and V is the chain linear speed in m/s. The motor power needs to drive the rig against chain tension and other additional factors like friction, resisting torque of the shafts and bearings.

A hydraulic or mechanical tensioner and a tensioner guide are generally used in order to provide a stable dynamic running condition on the chain wear test applied on the slack side.

As the chain needs sufficient oil on its testing environment, a special lubrication system needs to be set up. The oil should be delivered on to the chain such that it can provide the minimum lubrication conditions. Therefore, the chain should be located in a leak free case. The lubrication of the chain may either be manually done or provided with location in an oil bath. The principles of these two types of lubrication can be seen in Figure 6.2.

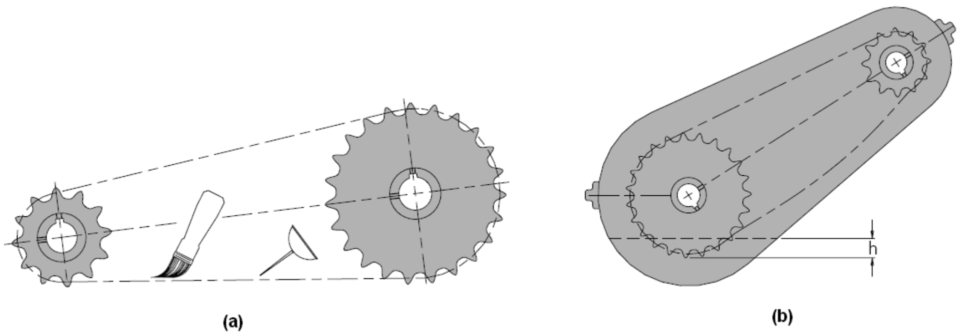


Figure 6.2 Chain drive rig lubrication with (a) manual operations or (b) oil bath [34]

As it's more applicable to represent the engine lubrication conditions, nozzle type of oil delivery components are used in wear elongation test rigs with the help of an oil pump and a simple oil circuit. The lubrication method is shown on the Figure 6.3. When the oil runs and circulates inside the rig, its temperature also needs to be maintained with cooling system acting on the coolant tank.

The rotational speed of the rig and the tension on the chain are kept constant, in order to remove the noise factors that may derive due to the fluctuations in speed or loading, as shown on Table 6.1.

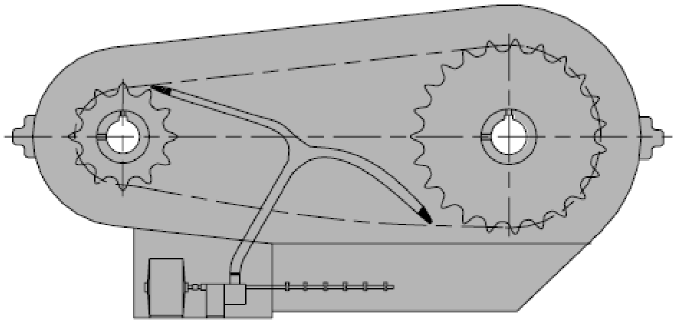


Figure 6.3 Chain drive rig direct lubrication with oil jet [34]

Also, as a standard for all wear elongation rig tests performed within this study, %1 level of artificial soot is used. Artificial soot is different than that is observed in diesel engine combustion process. It has an artificially adjusted size, morphology and chemical characteristics produced in laboratory conditions. This provides the opportunity to testing people to avoid the perturbations and variations among equivalent rig tests that may build due to different characteristics of naturally contaminated oils, retrieved from engine tests.

Table 6.1 Rig test parameters and definitions

Item	**Definiton**
Driving speed	4.500 rpm
Chain tension	500 N - 3.000 N
Sprocket size (T: teeth)	18T-18T
Chain type	9,525 mm pitch - Bush chain
Chain length	132 pitches
Speed alteration	Not applicable
Load alteration	Not applicable
Lubrication	Oil jet
Oil contaminant	%1 artificial soot

Figure 6.4 shows the general layout of the wear elongation test rig used within this project. An electric motor is used to drive one side of the chain circuit. As the aim of the rig test is to provide a reliable input for timing drive chain design in terms of lubrication environment and operation conditions, the above testing principles have been selected as they provide representative wear performances when compared with engine tests.

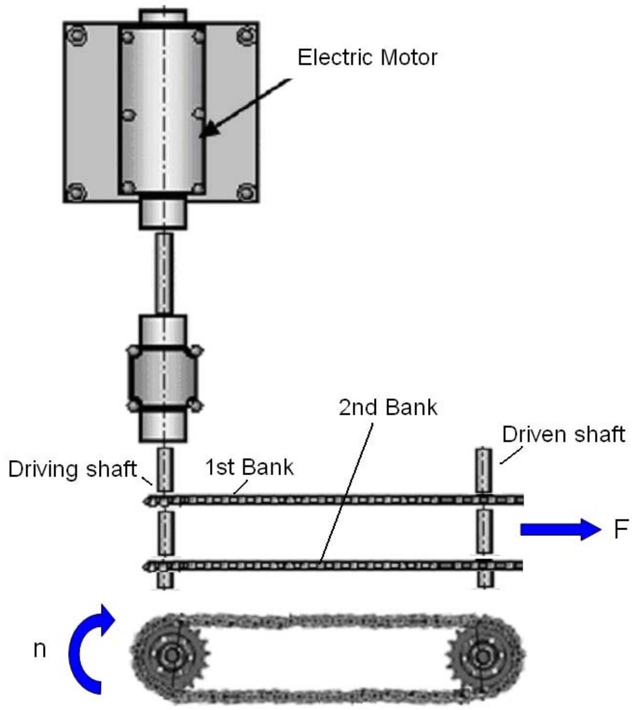


Figure 6.4 Chain wear elongation rig layout

Up to this point, chain wear and chain elongation terminology have been expressed as equivalent statements. Indeed, there is a real close correlation between chain wear and elongation, because the wear in chain pins and bushes increases the clearances of the elements. This increase in total clearance of all chain pitches lead to an increase in the total length of the chain when measured under a certain level tension. As shown in following expression, elongation of chain is calculated and expressed in percentage with respect to initial length;

$$\text{Chain elongation (\%)} = \frac{L_f - L_i}{L_i}.100 \qquad \textbf{(6.3)}$$

where L_f and L_i are the lengths of the chain after and before test, respectively. There is a certain limit for elongation of chain in percentage, which is more critical for timing chains. As timing chain delivers the rotational motion of crankshaft to camshaft in a pre-defined synchronization, elongation will lead to reduction in accuracy of this synchronization. For timing chains, %0.5 is the generic elongation limit for structural constraints (piston to valve clearances) and engine general performance.

6.3 Dynamometer Test Details

Dynamometer tests are key tools for engine and sub-systems at verification stage of product development. For assessment of timing drive chain wear performance, the general durability test has been used on all variants of tests in order to keep consistency. The specification of the engines used for development is shown in Table 6.2.

Table 6.2 Specification of tested engine

Item	Unit	Specification
Engine Type	-	4cylinder Inline Diesel
Capacity	Litres	2.4
Rated Speed	rpm	4000
Maximum Torque	Nm	370
Power	kW	105
Emission Level	-	Euro4
Fuel Injection System	-	Common Rail

6.4 Component Design Specifications

The chain tested under this configuration is a single strand, bush type chain which has 9,525 mm standard pitch dimension. Material and hardness specifications of the chain elements are shown on Table 6.3.

Table 6.3 Basic specifications of chain internal elements

Chain Element	Material	Heat Treatment	Coating	Surface Hardness
Internal Plate	SAE 1050	Quenching & Tempering	-	700Hv
External Plate	SAE 1050	Quenching & Tempering	-	700Hv
Bush	17NiCrMo6	Quenching & Tempering	Carbonitriding	900Hv
Pin	SAE 1050	Quenching & Tempering	Carbonitriding	900Hv

As it is known from chain wear test methods, the critical wear that directly leads to chain elongation happens between pin and bush elements. Therefore, the material and hardness compatibility between these elements are crucial for an acceptable tribological performance. These elements of the selected test chain have similar coating specification and hardness levels.

6.5 Objective

Wear elongation rig test is targeting to simulate the timing chain wear trend on a simple bench in a way that it fits to the real life engine test wear conditions. It aims to provide the same wear trend in a short running time.

The objective of this study is to evaluate the chain wear performances of a mineral, semi-synthetic and fully synthetic oil and determination of the rig test capability to meet engine test results.

7. WEAR TESTS

7.1 Rig Test Conditions

The selection of accurate parameters for rig testing is crucial for test to be sufficiently representative. The level of chain tension applied on the rig test is mainly determined according to the loading levels observed in engine conditions. This also needs to satisfy the maximum allowable running limit of the chain. Any tension equal to or above the chain fatigue limit will lead to unrepresentative conditions for chain wear process. Considering the engine running conditions, two levels of chain tension is determined as basis to the rig testing which are 2250N and 1600N. The test sequence and configurations are listed in Table 7.1.

It's not always possible to simulate the real running layout on the rig so suitable size and tooth profiled sprockets are selected for rig tests. 18 teeth sprockets have been selected in a way to simulate the existing tooth profile. Chain running speed or the electric motor speed is selected such that it maintains the linear velocity of the chain equal to engine condition.

Table 7.1 Rig test sequence and description of selected parameters

Test #	Chain Tension	Speed	Oil Type	Soot level
1	2250N	4500rpm	SAE 10W30	%1 artificial
2	1600N	4500rpm	SAE 10W30	%1 artificial
3	600N	4500rpm	SAE 10W30	%1 artificial
4	2250N	4500rpm	SAE 5W30	%1 artificial
5	1600N	4500rpm	SAE 5W30	%1 artificial
6*	1600N	4500rpm	SAE 5W30	%1 artificial
7	2250N	4500rpm	SAE 5W30-LowSAPS	%1 artificial
8*	2250N	4500rpm	SAE 5W30-LowSAPS	%1 artificial
9	1600N	4500rpm	SAE 5W30-LowSAPS	%1 artificial
10*	1600N	4500rpm	SAE 5W30-LowSAPS	%1 artificial

All the tested chains have been measurements have been done before test and during test. All tested chains have been measured at 4^{th}, 8^{th}, 12^{th}, 16^{th}, 20^{th} and 30^{th} hours. The initial measurements have been performed at more frequent points in order to derive the wear curve more precisely during chain bedding in period. After 30th hour measurements, every 10 hours has been used as measurement interval in order to reduce the measurement work when the wear curve has already built up.

7.2 Rig Test Results

7.2.1 Rig Tests with Mineral Oil

First test has been run with SAE 10W30 mineral oil under 2250N chain tension. %1 artificial soot in oil and 4500rpm of shaft speed are common parameters for all rig tests. The results shown in Figure 7.1 have been observed after the completion of the test at 30hrs.

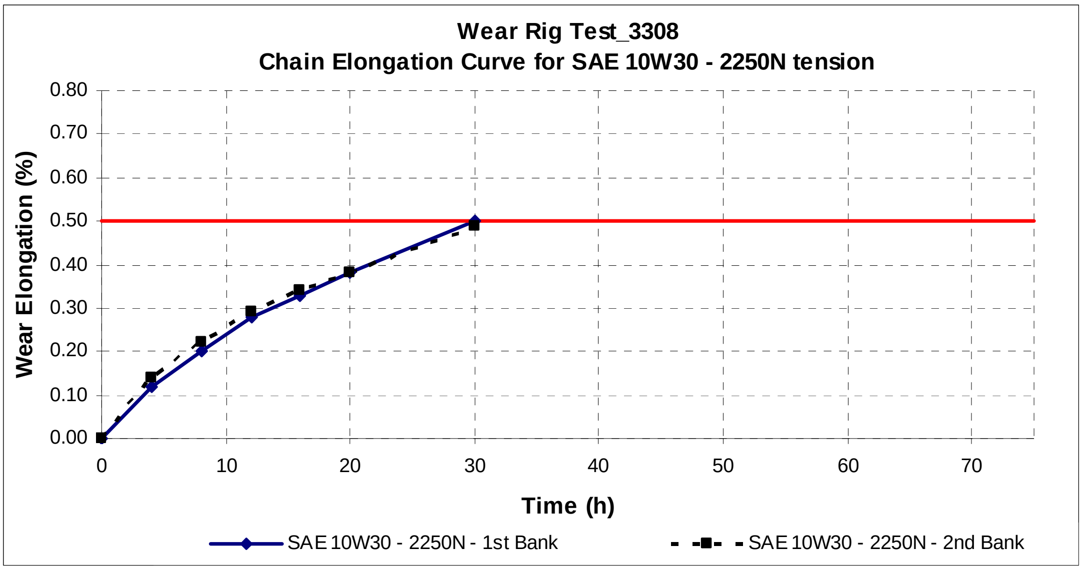


Figure 7.1 Chain wear elongation for mineral oil with 2250N chain tension

The horizontal bold line on the diagram shows the chain wear limit accepted as 0.5 in percentage for this application. The straight and dotted lines show the first and second banks of chains wear elongation curves, respectively. As it is clearly seen on the figure, these chains have similar chain wear trend which reaches to the limit after 30hrs test rig running time. Due to these are the first results obtained on the rig test, it's difficult to make a confirmed statement on the observations.

The second test has been run with the same conditions except for the chain tension which is reduced to 1600N.

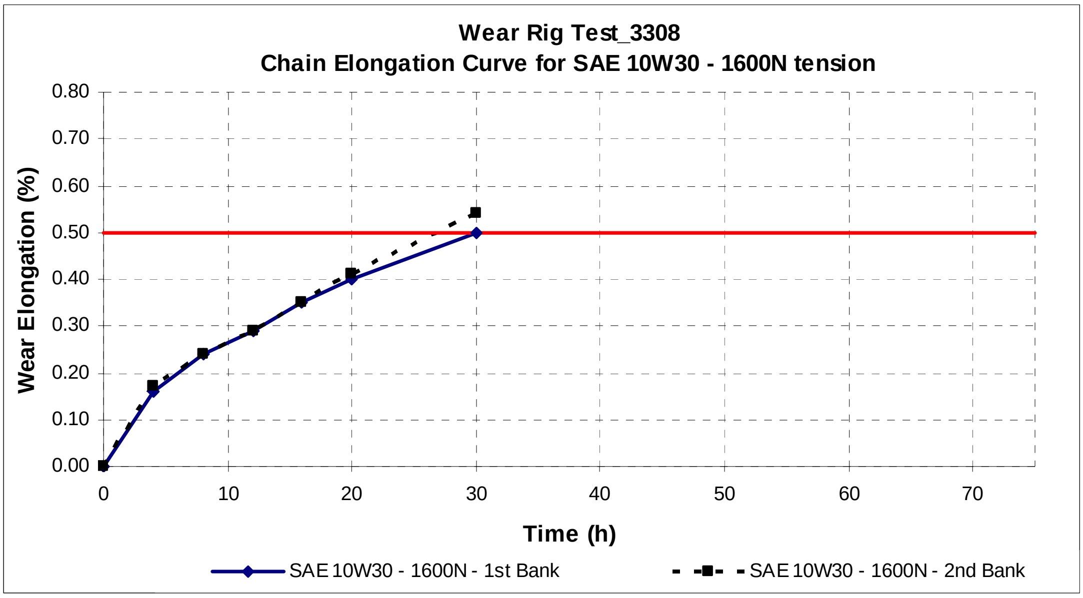


Figure 7.2 Chain wear elongation for mineral oil with 1600N chain tension

It can easily be expressed from Figure 7.2 that reducing the chain tension to 1600N has not improved the chain wear. The dotted line that shows the second bank chain wear even shows a slight increase in final wear point at 30th hour. This fact has been advised to happen due to variations of the test rig or the chain itself manufacturing tolerances. It can be clearly stated that the reduction in chain tension from 2250N to 1600N has provided no improvement in chain wear performance with other selected test parameters.

In order to make a prediction for chain wear response with respect to chain tension, loading has been reduced to 600N. Chain tension reduction to 600N shown Figure 7.3 has increased the chain elongation time to limit from 30 hours to 50 hours. On this level of chain tension, the two test bank chains wear curves have been more significantly separated especially after 8th hour, which is assumed to be a combination of measurement errors, test set up variation and chain manufacturing tolerances. It can be concluded that during rig tests with mineral oil, the tested chain tension should be considerably reduced to see its effect on wear.

Results of the mineral oil tests show that lubrication and surface parameters are more dominant on chain wear in 1600N to 2250N tension interval, other than contact forces.

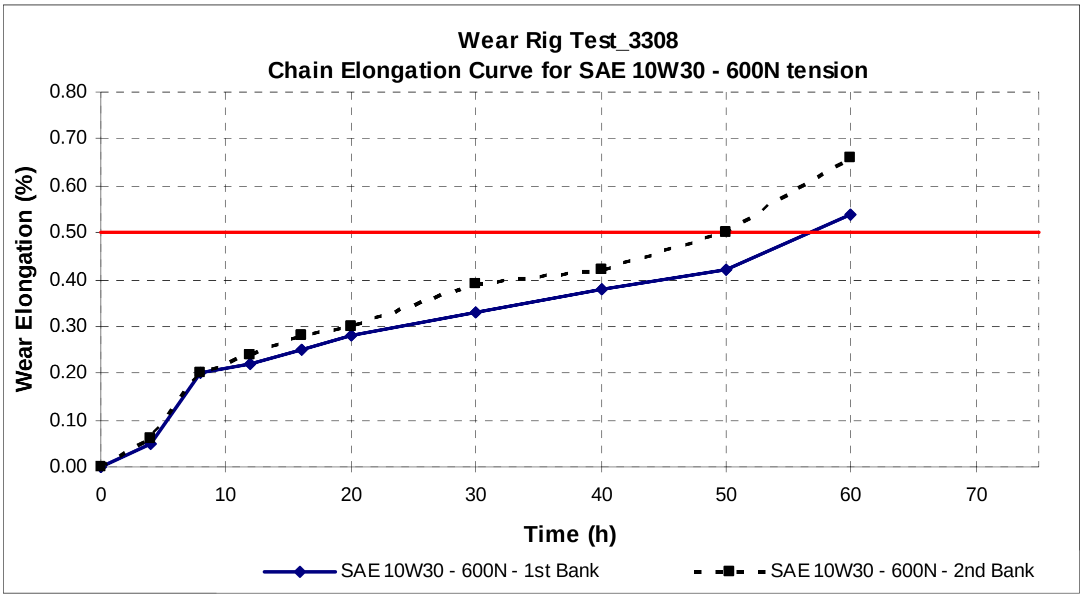


Figure 7.3 Chain wear elongation for mineral oil with 600N chain tension

7.2.2 Semi Synthetic Oil Rig Tests

The next series of rig tests have been carried out with SAE 5W30, semi-synthetic oil. As shown on Figure 7.4, semi-synthetic oil tests with 2250N tension showed similar wear elongation curve for the parallel banks.

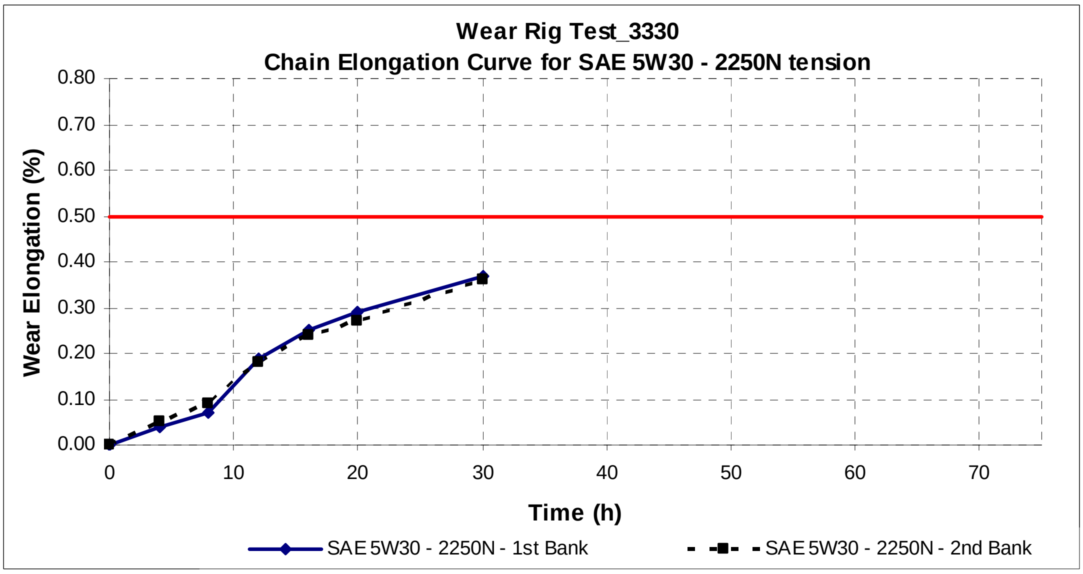


Figure 7.4 Chain wear elongation for semi-synthetic oil with 2250N chain tension

This test also finalized after just 30hours in order to keep the testing consistency with other tests. According to 30 hours performance results it can be expressed that the semi synthetic oil wear performance under 2250N tension is %25 better than mineral oil.

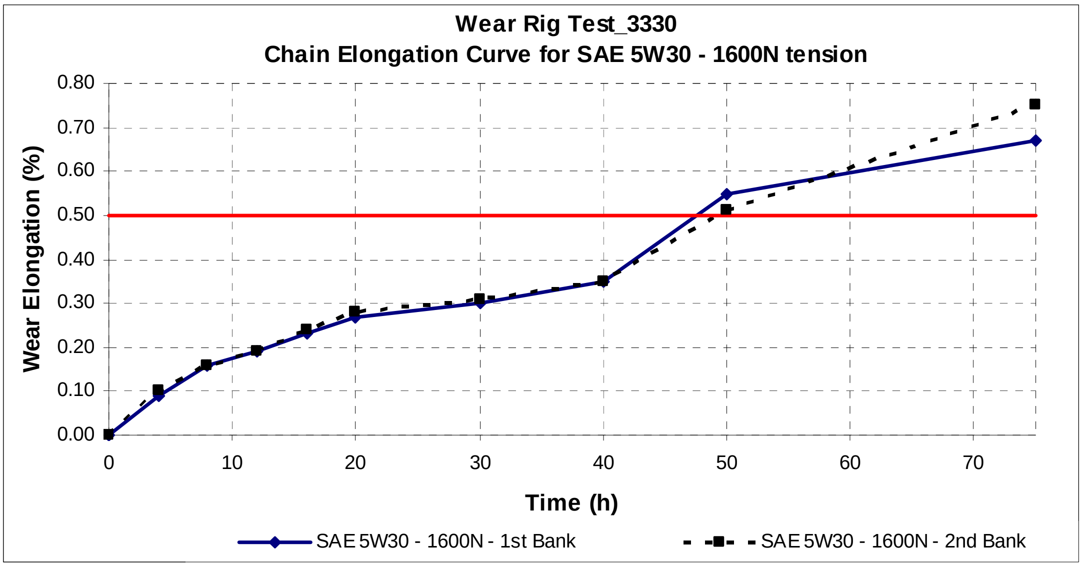


Figure 7.5 Chain wear elongation for semi-synthetic oil with 1600N chain tension

Semi synthetic oil this time tested with 1600N tension for 75 hours. Similar wear elongation behaviour is also observed for two banks of test rig, with minor deviation after 40 hours as shown on Figure 7.5. After 40th hour, the wear curve seems to make a step which is supposed to be due to the removal of the hardened coating on the pin which leaded to a softer contact condition and hence uncontrolled wear.

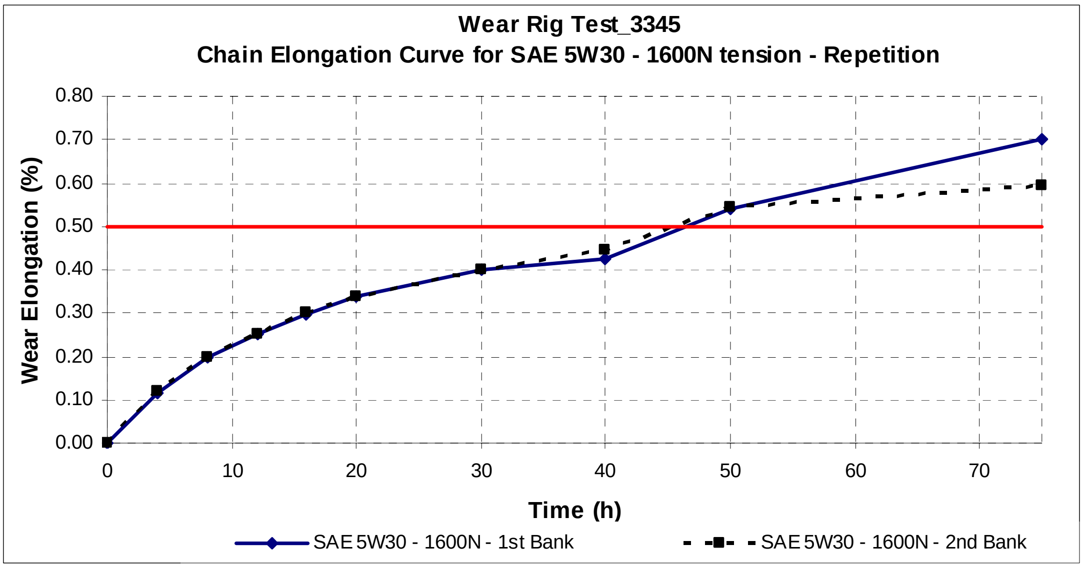


Figure 7.6 Repetition of semi-synthetic oil rig test with 1600N chain tension

Comparing the 30th hour results for 1600N chain tension, it can be seen that SAE 5W30 oil chain wear performance is %40 better than SAE 10W30 oil. For assessment of consistency of the test results and evaluation of the repeatability of the test rig, it's decided to perform a repetition test on this configuration.

The repetition test shown on Figure 7.6 generally shows the same behaviour in chain wear trend. Without a statistical analysis, this result satisfies that the bench test's running environment's repeatability is acceptable for the existing level of accuracy.

7.2.3 Fully Synthetic Oil Rig Tests

To keep consistency and commonality among the tests performed with mineral and semi-synthetic oils, fully-synthetic oil tests have been run in the same sequence. The first test completed with 2250N chain tension is shown on Figure 7.7.

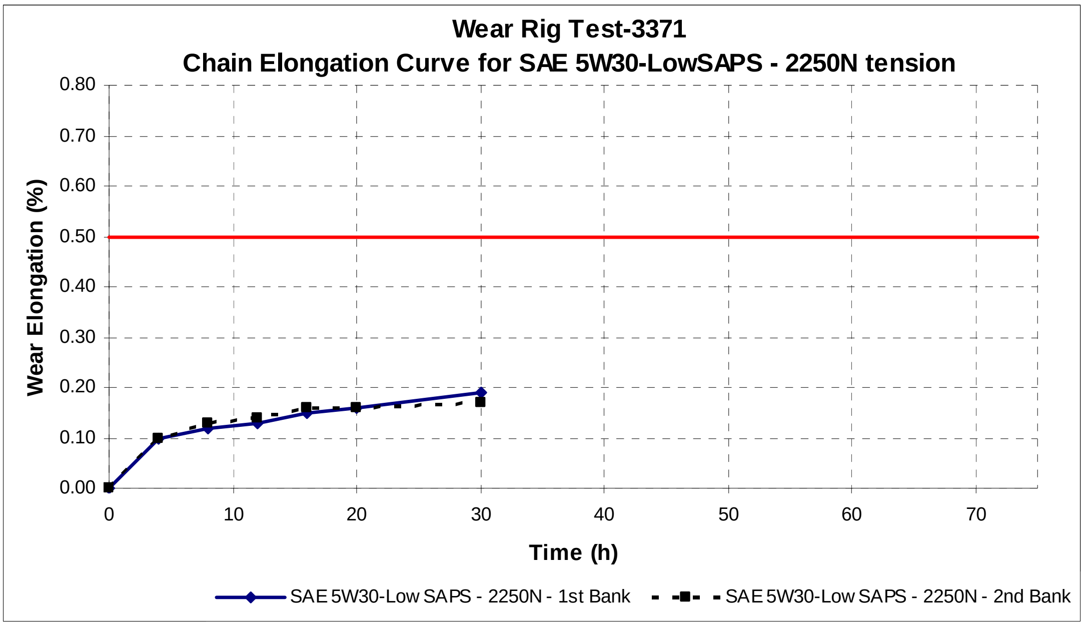


Figure 7.7 Chain wear elongation for fully-synthetic oil with 2250N chain tension

As common with the previous tests, two banks of chains show similar wear elongation trend until the tested hours. The diagram clearly shows that the bedding-in period of the chain with fully synthetic oil is earlier finalized when compared with the rest of oils tested. 30 hours wear elongation level shows that fully synthetic oil in the ranges of %250 and %200 improves the wear elongation performance when compared with mineral and semi synthetic oils, respectively. A repetition test is performed for this configuration, as shown on Figure 7.8.

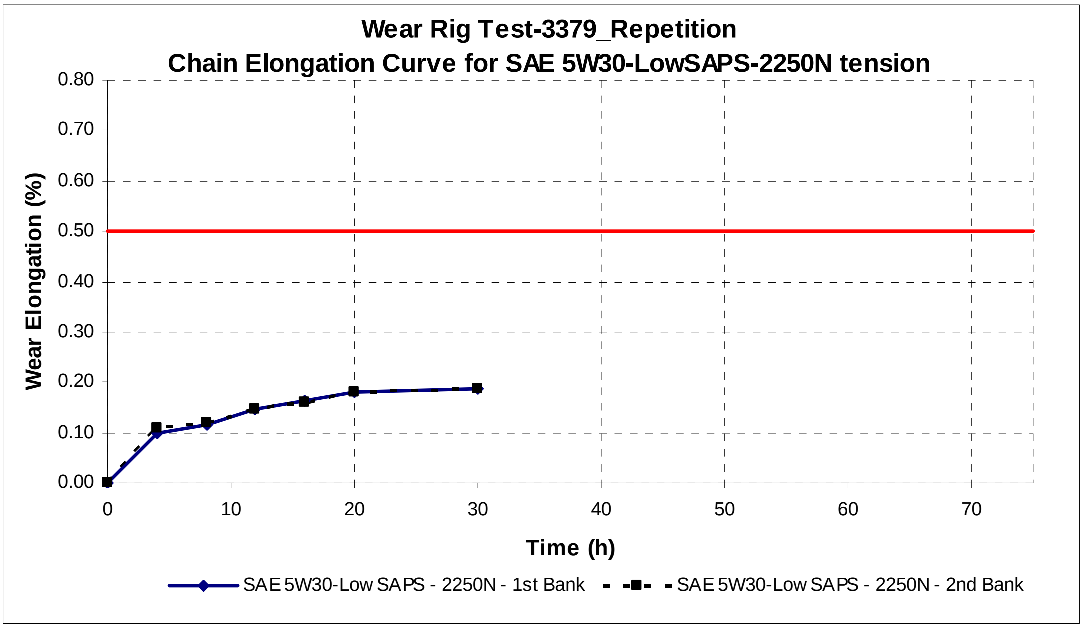


Figure 7.8 Repetition of fully-synthetic oil rig test with 2250N chain tension

Repetition test with 2250N chain tension on fully synthetic oil has almost given the same results. The straight line observed between 20th and 30th hours has been assumed as the measurement system variation of chain elongation measurement equipment.

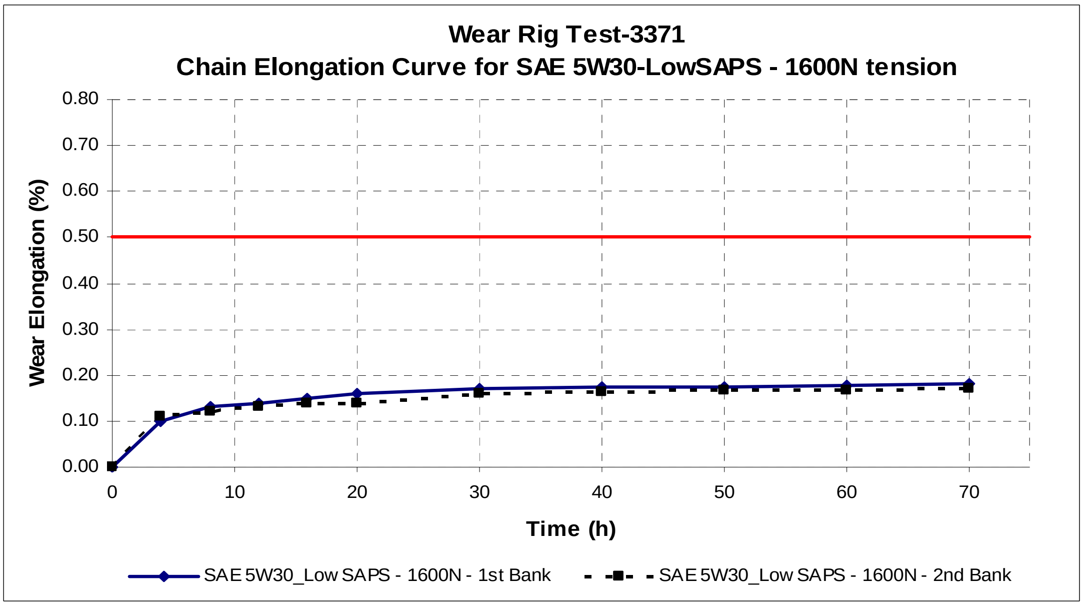


Figure 7.9 Chain wear elongation for fully synthetic oil with 1600N chain tension

Fully synthetic oil tests have been preceded with 1600N chain tension and its repetition as shown on Figure 7.9 and Figure 7.10, respectively. Two tests show similar chain wear trend and the elongation of the chains of consecutive banks are close to each other. The elongation curves between 30 and 70 hours with 1600N tension do not demonstrate any significant change, the curve is nearly horizontal and the wear is assumed to be negligible.

The horizontal elongation trend indicates that the chain wear is on such a level that is within the tolerance margin of measurement system. End of test wear elongation which is %0.19 dictates that the fully synthetic oil wear performance is 3 times better than semi synthetic oil under the tested conditions after 70 hours.

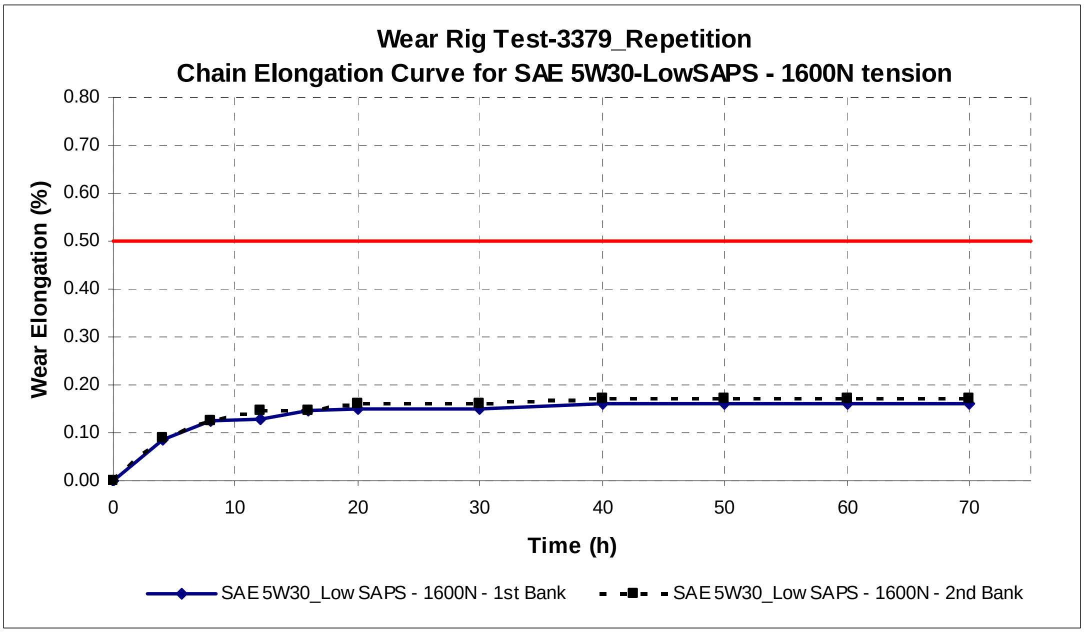


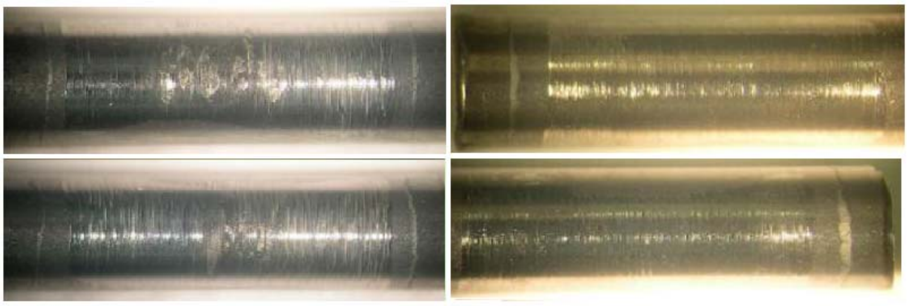
Figure 7.10 Repetition of fully-synthetic oil rig test with 1600N chain tension

7.2.4 Teardown Inspection

After completion of rig tests and elongation measurements, end of test chains are broken down into pins, bushes and links by grinding the pin projections on both ends of the chain. Pins and bushes are examined under SEM followed by visual inspections.

Visual Inspections

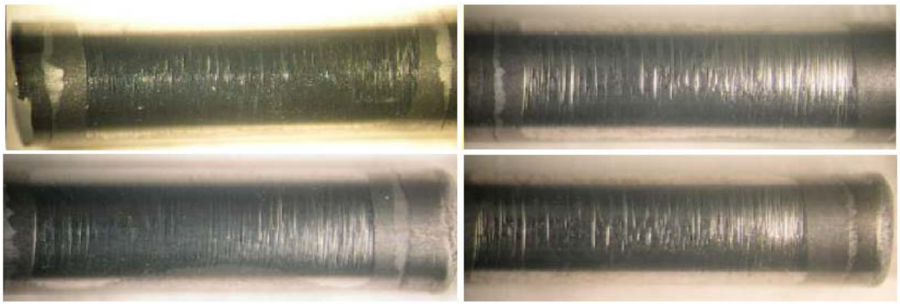
End of test chain pins were inspected visually from SAE 10W30 mineral oil used rig tests. As shown on Figure 7.11, the surface conditions of the pins from 2250N loaded chains point out that these pins had abrasive wear with scratch marks unevenly distributed. The abrasive wear lines are more uniform on pins of 1600N loaded chain, although these chains displayed similar end of test elongation performance at the same running time.



(a) (b)

Figure 7.11 Pins tested with mineral oil under (a) 2250N (b) 1600N chain tension

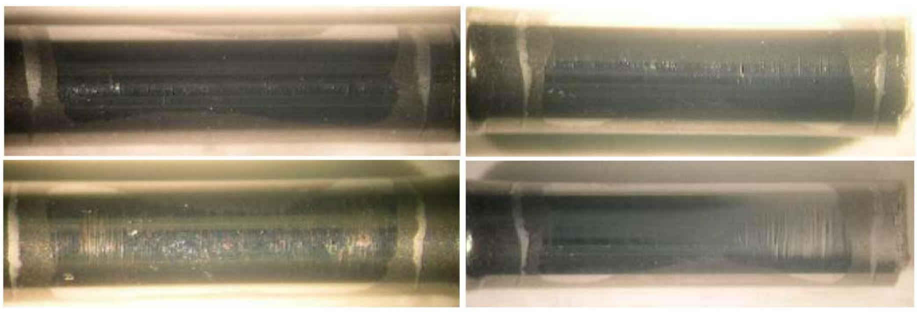
End of test condition of the pins of the chains tested with SAE 5W30 are shown on Figure 7.12. The pins of 1600N loaded chains demonstrate more severe abrasive wear marks on surface and the scratch lines have no uniform distribution.



(a) (b)

Figure 7.12 Pins tested with semi synthetic oil under (a) 2250N (b) 1600N tension

On the contrary, pins of 2250N loaded chains show lower depth abrasive wear lines which are more evenly distributed. The better condition of 2250N loaded chain pins is due to comparably higher level of running time of 1600N loaded chains.



(a) (b)

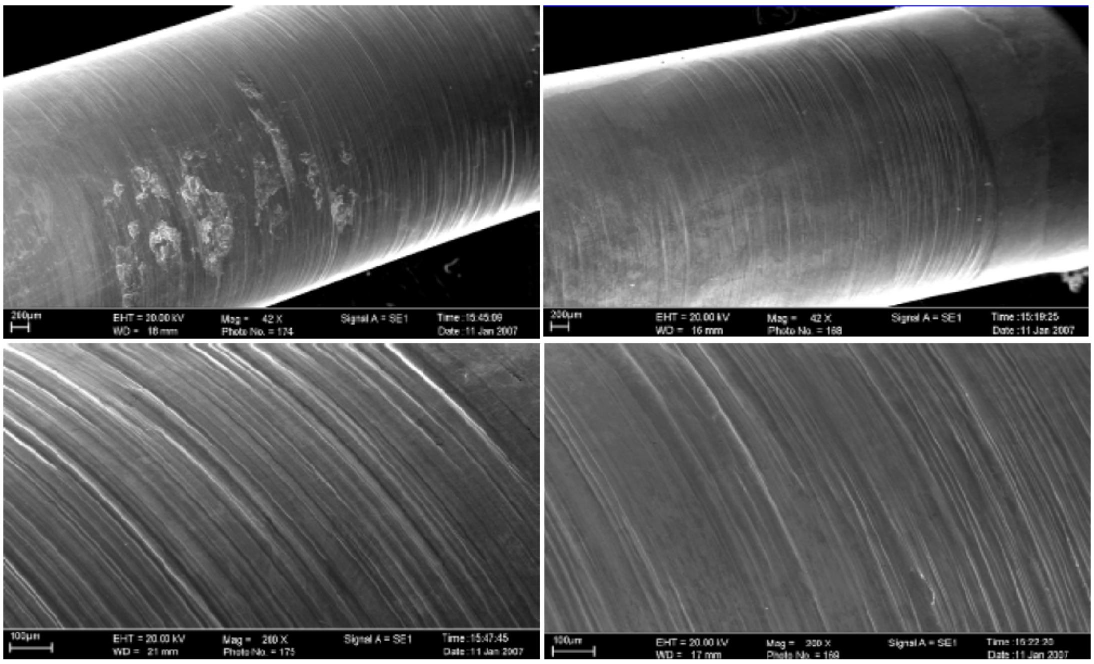
Figure 7.13 Pins tested with fully synthetic oil under (a) 2250N (b) 1600N tension

For fully synthetic oil, the end of test chain pins show similar visual appearances. Both show very slight scratch lines on each ends, which are more visible on 1600N loaded chains. This is because of the comparably higher running time of the pins under 1600N chain load. As seen on Figure 7.13, the pins after rig tests with SAE 5W30-LowSAPS oil show very light polishing appearance.

Scanning Electron Microscope (SEM) Examinations

SEM is one of the widely used techniques on material science for surface analysis. Visual examinations by naked eye generally provide rough information which may not be sufficient for a complete assessment of surface structure at the end of test.

It's clear from the Figure 7.14 that the surface conditions of the pins are more visible under SEM. Higher magnification confirms that both pins from 2250N and 1600N loaded chains show apparent abrasive wear marks. The depth of scratch marks is deeper for 2250N loaded pins. The hard soot particles existing inside the lubricant and their wear effects are more severe on 2250N tension, due to the relatively greater contact forces on this running condition. There are even signs of surface damages on some of the pins that agree with the three-body abrasive wear condition, induced by hard particles between surfaces.



(a) (b)

Figure 7.14 Pins tested with mineral oil under a) 2250N b) 1600N tension

Bushes of the chains that have been run with mineral oil can be seen on Figure 7.15. Similar to pins, end of test bushes also show deep wear scratches calling abrasive wear.

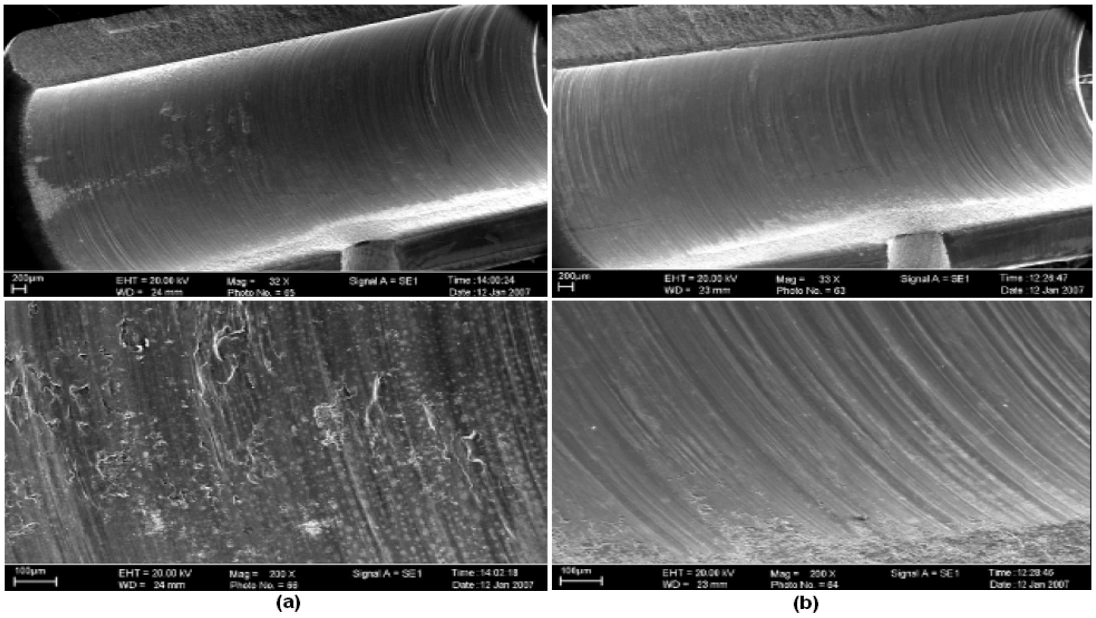


Figure 7.15 Bushes tested with mineral oil under a) 2250N b) 1600N tension

On both bushes show visible deep running traces, however, the bushes of 2250N loaded chain demonstrate local adhesion areas with unevenly distributed surface impurities. 1600N loaded chain bushes do not indicate adhesion marks, possibly due to lower contact forces exerted on surface compared to 2250N loaded chains.

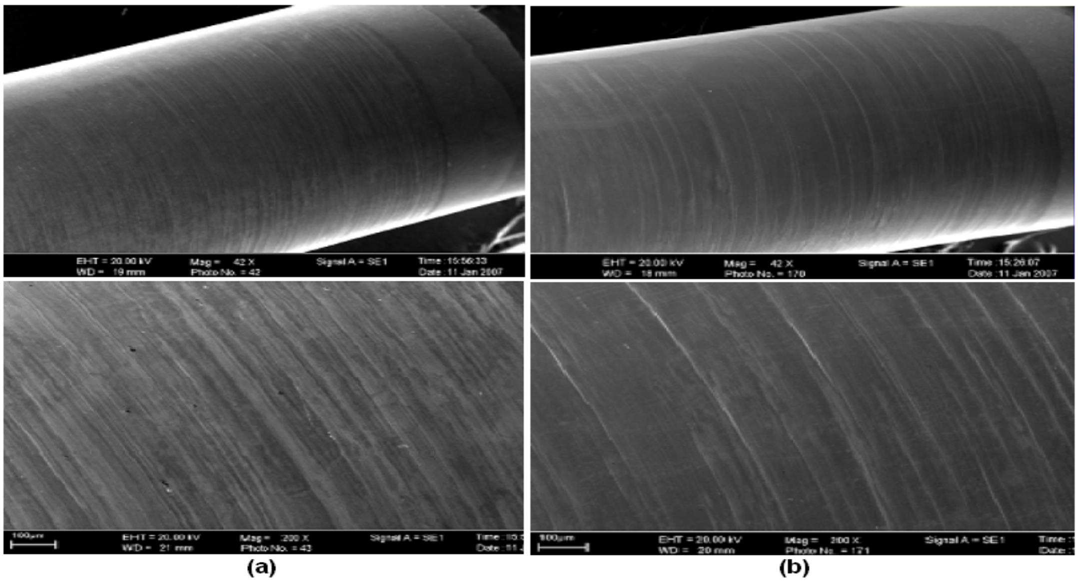


Figure 7.16 Pins tested with semi synthetic oil under a) 2250N b) 1600N tension

Pins of the chains that were run with semi synthetic oil are shown on Figure 7.16. Both 2250N and 1600N loaded chain pins demonstrate similar abrasive wear marks which are not as severe as the pins of mineral oil tests. Although 2250N loaded chains have run 45 hours less than their counterparts on the rig, distribution of the wear scratches are more uniform but in higher depth compared to 1600N loaded pins.

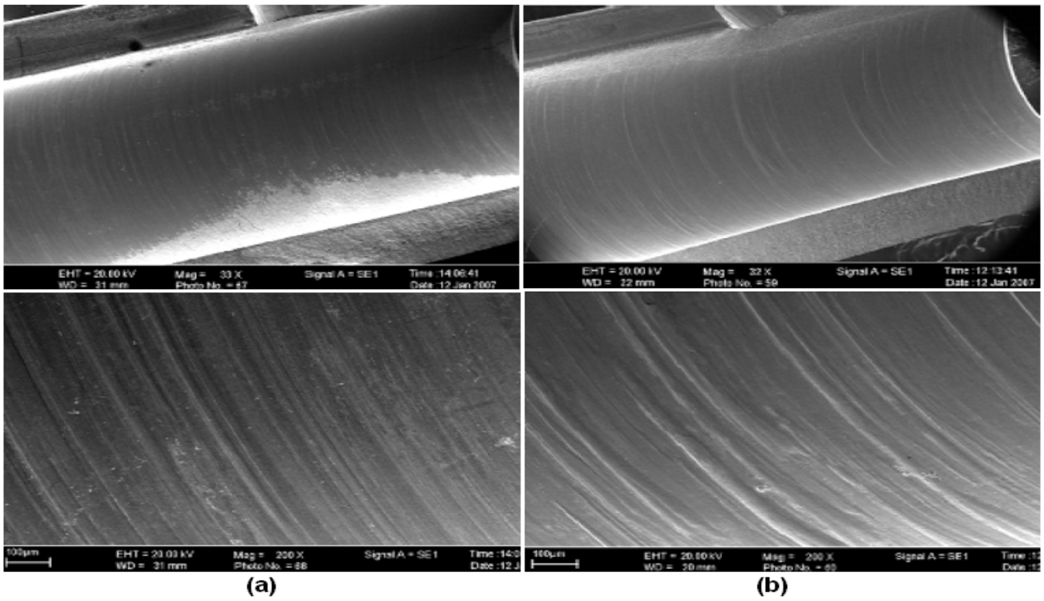


Figure 7.17 Bushes tested with semi synthetic oil under a) 2250N b) 1600N tension

On the contrary, the bushes of 1600N loaded semi synthetic lubricated chains have rare but deeper abrasive wear marks on the surface. This also can be referred to the high testing time with lower tension chains.

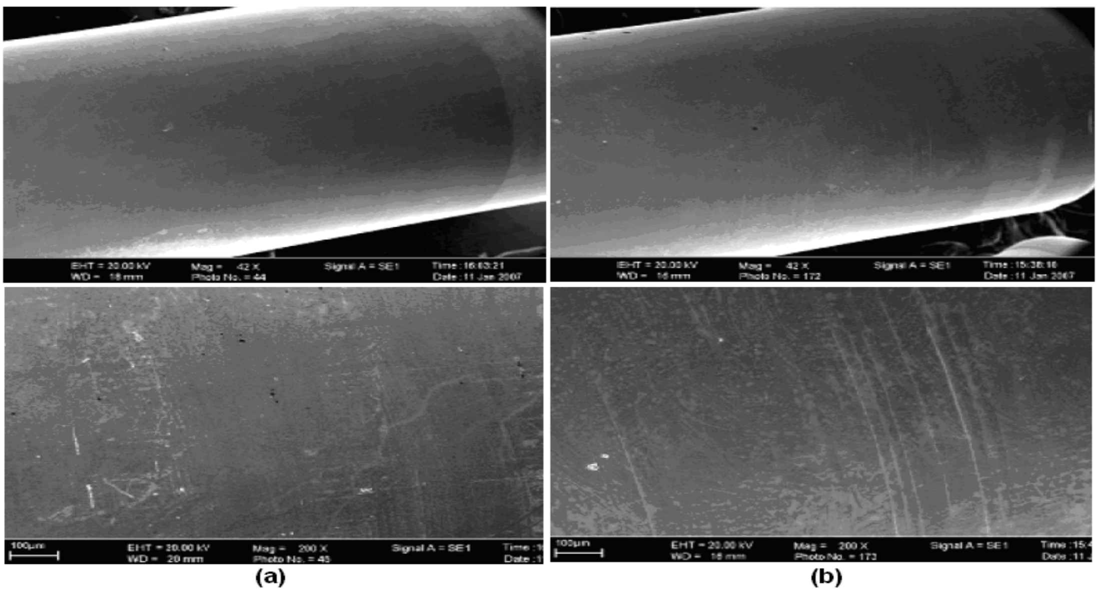


Figure 7.18 Pins tested with fully synthetic oil under (a) 2250N (b) 1600N tension

The pins of the fully synthetic oil tests demonstrate no abrasive wear marks but slight and local wear scratches on one end of the pin for 1600N loaded chains. End of test pins can be seen on Figure 7.18. This also may be referred to the high running time of low tension chains. There are no surface damages on both pins; only bright surfaces are apparent which can be called as polishing.

Also both bushes of fully synthetic oil tests show comparably similar end of test surface structure, regardless of the applied chain load. As displayed on Figure 7.19, there are slight signs of local adhesions on the axis of bush orientation which may be referred to the oil starvation during test operation.

Looking at the results observed on the rig tests, it’s obvious that the best chain wear elongation performance has been achieved with fully synthetic oil. The end of test components of chains tested with fully synthetic oil are also acceptable and meet elongation targets while satisfying the surface conditions expected from the application.

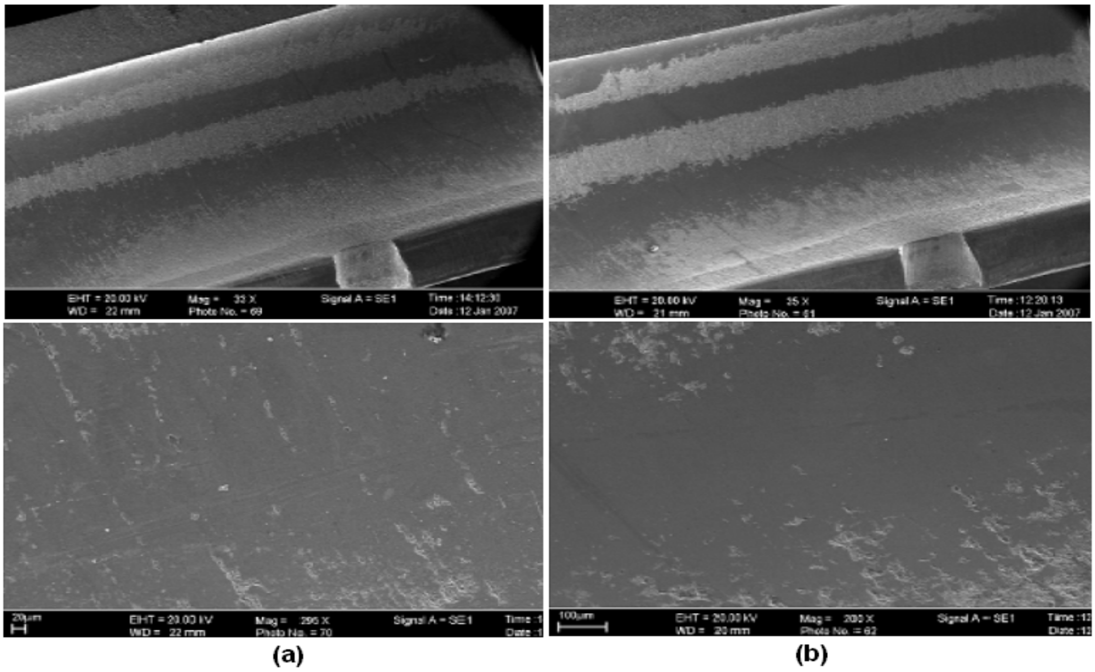


Figure 7.19 Bushes tested with fully synthetic oil under a) 2250N b) 1600N tension

Semi synthetic oil has shown slightly better chain wear elongation performance on the rig for short running hours. The extension of test duration has put adverse effects on chain leading the surface coating to remove and hence resulted with unacceptable chain wear. Although the mineral oil wear performance was considered to be more aggressive than semi synthetic oil, detailed inspections on chain components have shown no different wear observations, all pin and bushes did show abrasive wear. On the contrary, the pins and bushes of the chains that were run with fully synthetic oil were completely polished with only a few slight scratches due to long period of running time.

7.3 Engine Tests

As a general testing procedure, frequent oil samples are taken from the sump in order to visualize the change in oil parameters that are critical for engine test. The oil analysis provide information regarding soot accumulation rate, TBN depletion rate and TAN-TBN cross over estimation, fuel dilution rate and the change in critical elements inside oil that can indicate or induce wear on engine components. The soot content is maintained maximum at %3 level as a generic industrial standard for tribological applications, due to high effect of soot in wear after passing the limit. TAN – TBN crossover is another limiting factor which indicates limit in increased acidity of the oil. There are also various limits for fuel and other elements in oil are monitored during engine tests. As soon as the oil reaches one of these limitations, the oil is drained and replaced. On the other side, in order to keep close control on the wear and elongation trend of the timing drive chain, the engine has been stopped and partially torn down. The chain taken outside the engine is measured in the same way as performed for the rig tests, and change in chain length is recorded frequently. The expectation for tests is to keep the chain wear elongation within limits after 1200 hours of total running time.

7.3.1 Dynamometer Tests with Mineral Oil

Three different engines on the same specification have been run with SAE 10W30 mineral oil on dynamometer. Chains have been tested along a certain running time on the engines, as shown on Figure 7.20. Due to the difficulties of intermediate inspections on timing chain drive of engines, low frequency of chain measurements have been performed on these tests. Among three engines have been tested, triangular marked chain wear trend clearly shows the best performance. This chain reaches the elongation limit at 600 hours where as the other two tests have a trend of wear to pass the limit before 400 hours of running rime.

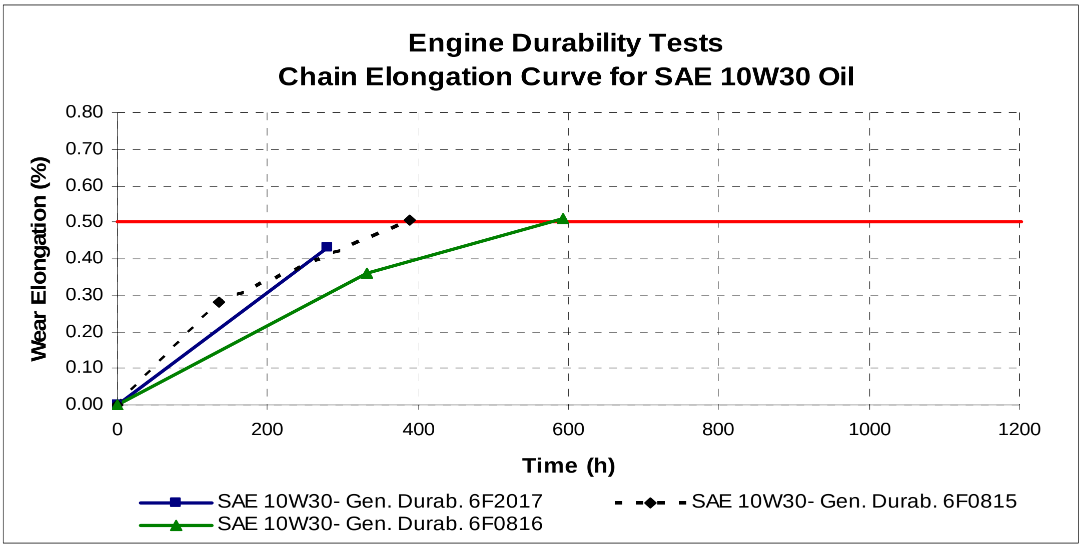


Figure 7.20 Chain wear elongation on dynamometer tests with mineral oil

There is a significant variation between the results achieved for three different engines which can be caused by the engine to engine operational and manufacturing differences that may affect chain running environment to demonstrate different behaviours.

7.3.2 Dynamometer Tests with Semi Synthetic Oil

Similar to tests run with mineral oil, as displayed on Figure 7.21, semi synthetic oil used engine tests show significant variation in chain wear trend. This is obviously due to the difficulty of keeping the same level of consistency on real life engine tests as achieved on rig tests.

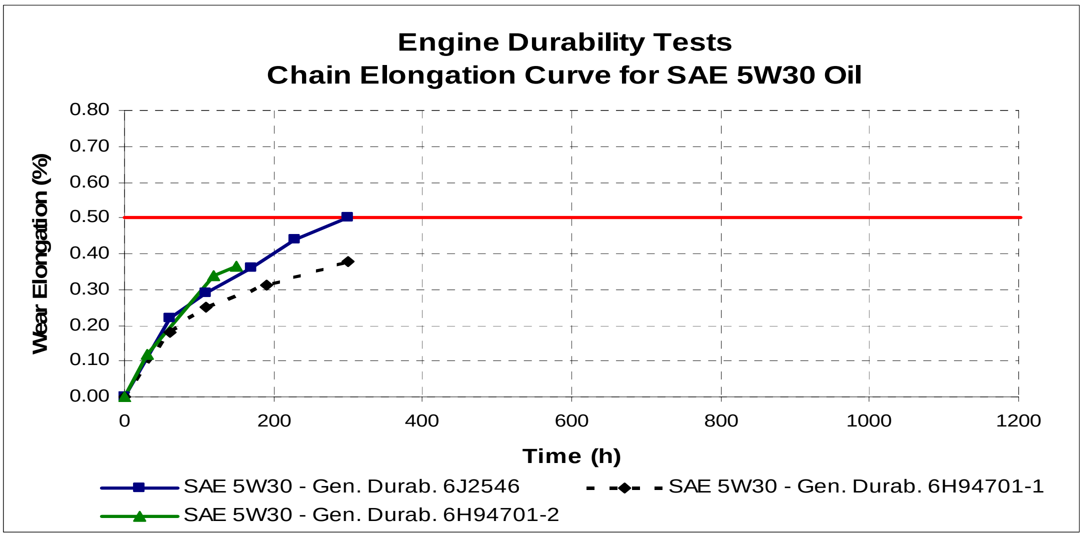


Figure 7.21 Chain wear elongation on dynamometer tests with semi synthetic oil

The square marked line of chain wear trend reaches the elongation limit after 300 hours of running time, whereas the dotted line demonstrates a better wear performance such that it goes to the limit at a certain time between 400 and 600 hours.

7.3.3 Dynamometer Tests with Fully Synthetic Oil

Two dynamometer engines have been run with fully synthetic oil as shown Figure 7.22. Both engines display a close correlation in performance comparison. However, the dotted line has a slightly higher wear accumulation rate. Two chain wear trends observed with fully synthetic oil indicate that the end of test chain elongation levels will be within expected tolerances.

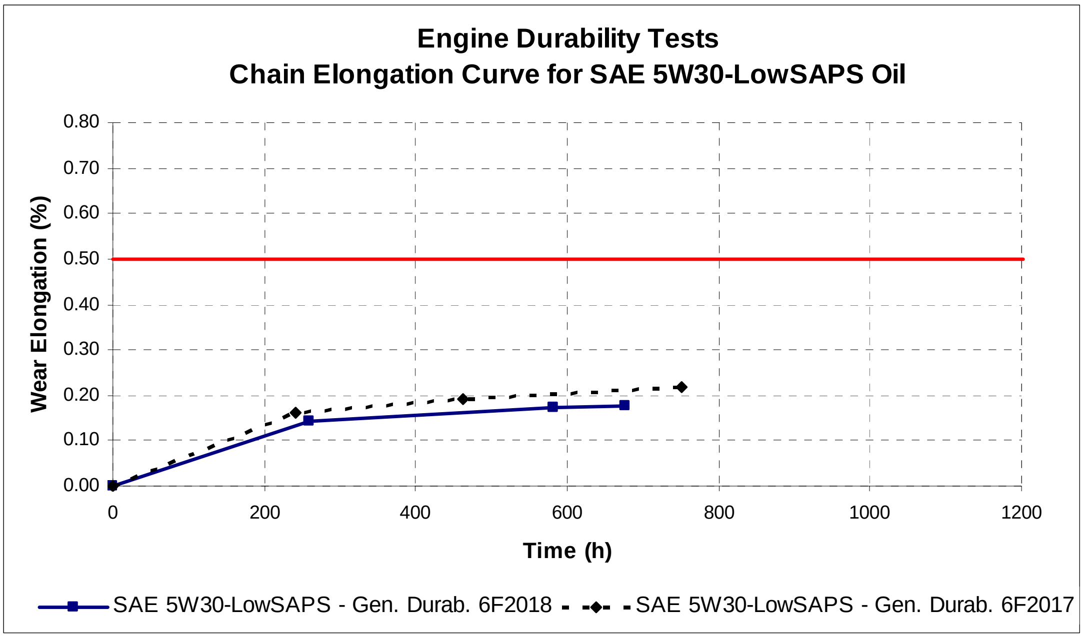


Figure 7.22 Chain wear elongation on dynamometer tests with fully synthetic oil

7.3.4 Teardown Inspection on Dynamometer Tests

After completion of wear measurements, end of test chain samples have further been inspected by SEM technique. Particular pins and bushes are selected from the chain batch which reflect the most recurring surface structure among the number of pitches. As shown on Figure 7.23, both pins and bushes show deep abrasive wear with continuous scratch marks on surface. The abrasive lines are finer on bushes whereas deeper on pins. Also the distribution of the lines indicating wear is not uniform on pins.

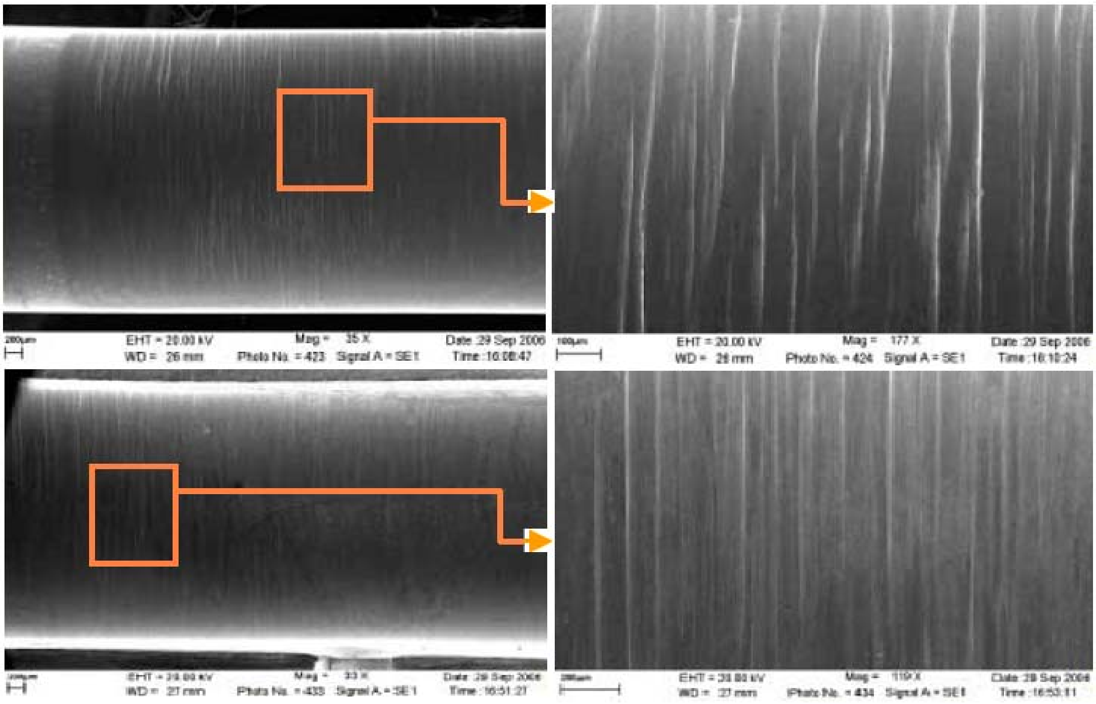


Figure 7.23 Pins (above) and bushes tested with mineral oil on dynamometer

The pins and bushes of the chain that has run on the engine with semi synthetic oil are demonstrated on Figure 7.24. Both elements show similar surface condition and uniform abrasive wear is common. The scratch lines are more frequent on semi synthetic oil used components, compared to parts tested with mineral oil.

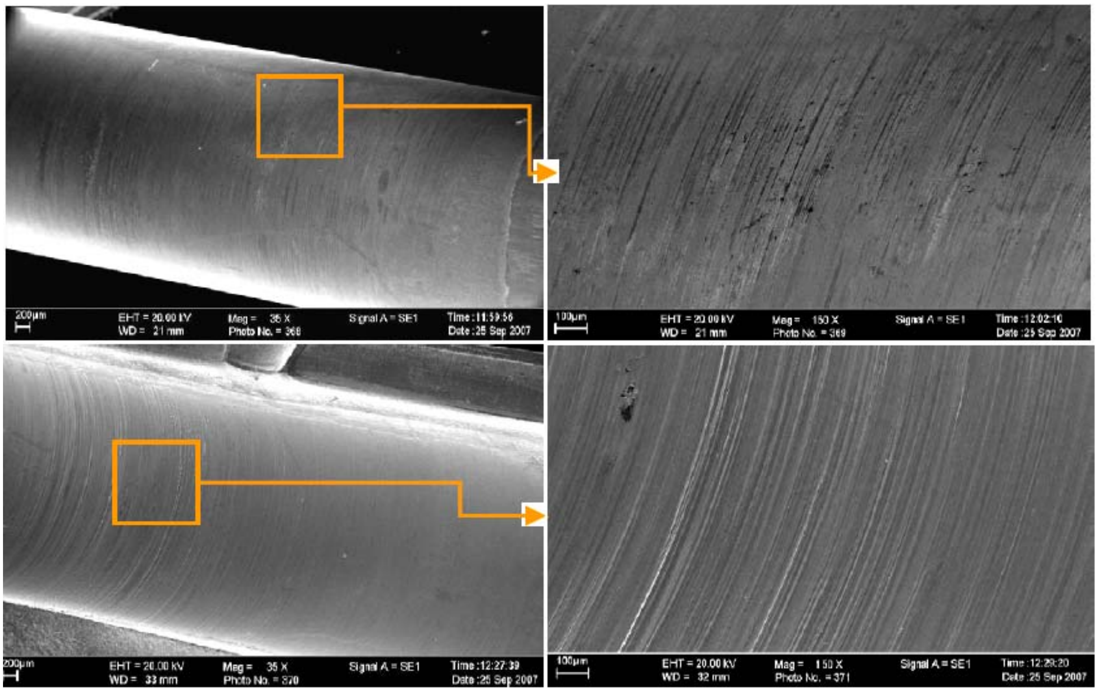


Figure 7.24 Pins (above) and bushes tested with semi synthetic oil on dynamometer

The only sets of components that show different wear mode exist on the chains tested with fully synthetic oil. As obviously seen on Figure 7.25, the pin shows comparably good surface condition, with almost no wear marks. There are very low local scratches.

On the contrary, the bushes show uniform and frequent abrasive wear lines, which is significantly lower depth compared with the chains tested mineral and semi synthetic oils. This can be referred to the higher running time of the fully synthetic oil using engine tests.

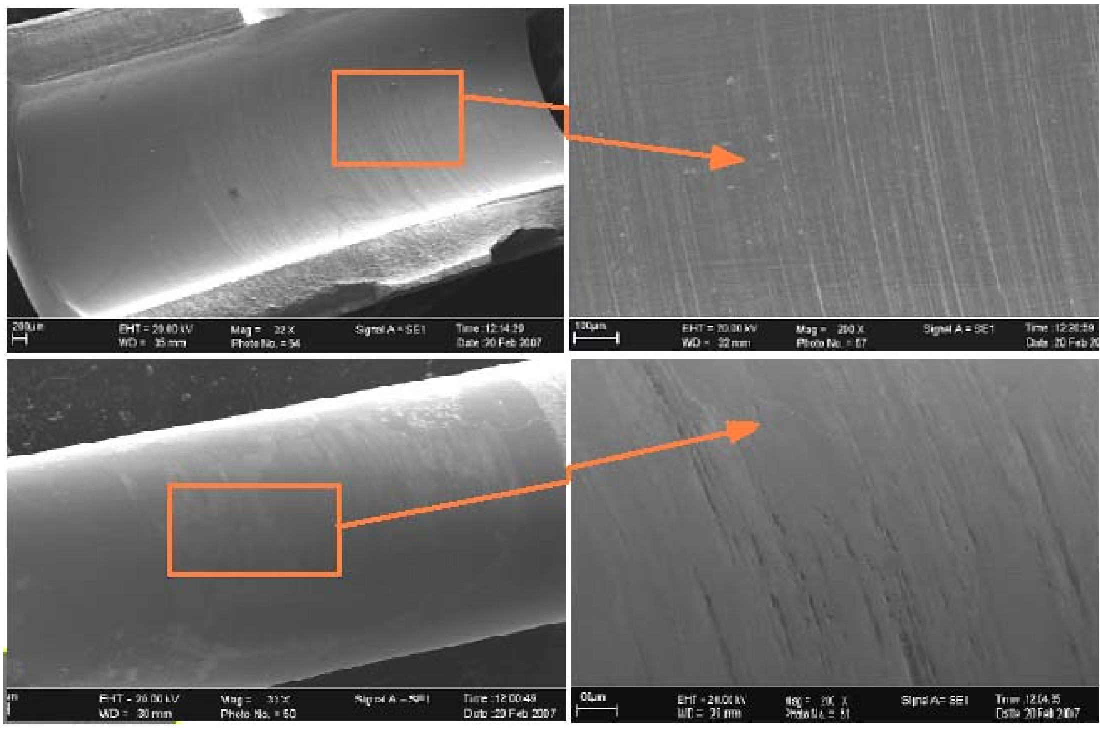


Figure 7.25 Pins (below) and bushes tested with fully synthetic oil on dynamometer

In general, the impressions observed after rig tests have been repeated during inspection of the elongation curves and SEM analysis of the tested components. For both of the mineral based and semi synthetic oils, chain wear elongation curves have reached the limit between 300 and 600 hours of engine running time. Conversely, the engine tested with fully synthetic oil has been run until 775 hours and only %0.22 chain wear elongation observed. As per engine validation requirements, fully synthetic oil gives the best performance in chain wear characteristics.

With a rough overview after completion of the rig and dynamometer tests, it’s easily observed that there is an acceptable correlation between elongation rig and engine dynamometer in terms of chain wear elongation performance. SEM analysis performed on the surface structure of the chains also supports this statement.

At this point of study, it's important to examine the specifications of the selected oils in more detail. As shown on Table 7.2, there are various parameters that characterize the engine oils defining the performance specifications. Most of the tribological studies in the past were concentrated on sulphated ash content, sulphur, phosphorus and zinc level, dispersant, detergent and ZDTP level, and also dispersant and detergent TBN values.

Table 7.2 Important parameters of the selected oils

Specification		**Mineral**	**Semi Synthetic**	**Fully Synthetic**
Certification		**ACEA E5-99**	**ACEA E4/E5/E7**	**ACEA E6**
SAE		10W-30	5W-30	5W-30
Density @ 15 °C	kg/m3	880	851	853
Viscosity @ 40 °C	mm2/s	73	55.5	72.3
Viscosity @ 100 °C	mm2/s	11.7	9.72	11.6
Viscosity Index	-	155	160	155
Flow point	°C	-45	-39	-54
Total Base Number	mg KOH/g	11.9	8.2	10
Sulphated ash content	% m/m	1.60	1.0	0.85
Sulphur level	mg/kg	9171	4000	2000
Phosphorous level	mg/kg	1507	1000	550
Zinc level	mg/kg	1262	1000	0
Dispersant TBN	mg/kg	-	-	-
Dispersant level	mg/kg	-	-	-
Detergent TBN	mg/kg	-	-	-
ZDTP type	mg/kg	-	-	-
ZDTP level	mg/kg	-	-	-

Effectively, some of the parameters have been found to reduce wear solely or combined with another one. Inversely, some parameters have been found to have no effect on wear properties which used to improve wear on particular applications. Hence, it's difficult to make a confirmed statement for any specific property to have common influences on every practice. Considering that this study has been performed on completely different specification of oils, it is not possible to judge the specifications of the oils for contribution on wear performance. Therefore this study is recommended to be extended by support of the lubricant suppliers for a detailed investigation on the significance of oil parameters that effect wear phenomena.

7.4 Rig & Dynamometer Test Correlation

A good representation of the real life engine environment is important for obtaining realistic results from test rigs. Dynamometer tests are developed for assessment of durability of the components under customer usage profile. For shortening test timing, the critical damaging operating conditions and durations are selected specific to these systems.

The best simulation of an engine's timing chain wear performance on a bench can be achieved by full representation of driving layout, interactions with neighbour systems like torsional vibrations and resisting torques, operation modes, lubrication environment that forms under effects of combustion and other engine components, chain loads and vibration characteristics. A bench test to simulate all the above conditions can only be satisfied with again a real life engine run. This type of approach will also be out of the scope of bench tests which needs to be simple, low cost, easily manageable and fast responsive.

8. CONCLUSIONS

After the recent rapid development in automotive industry and increased competitiveness, automotives are subject to far more challenging expectations and various requirements at the moment than ever before. Especially internal combustion engines are under great pressure to meet the demands of customer in terms of performance and durability together with emission targets. In respect to meet these requirements, engine components are expected to function under several operational conditions with maintaining targeted durability. Lubrication is of great importance on engine components for keeping durability against wear. Although oil specifications are under standardized classification for automotive applications, the recent improvements in lubricant technology has made it possible to produce engine oils to meet application specific durability and performance requirements.

Within this study, a fundamental approach to tribology has been built with detailed explanations of wear, friction and lubrication. Engine lubrication system and affected components have been described by providing specific information on automotive engine oils and lubricant contamination process which effect wear mechanism. Special and standard bench tests have been discussed as a part wear assessment techniques. Different timing drive systems and specific properties have been evaluated where chain drive systems have been investigated in details in terms of chain types and basic system dynamics.

As a part of the study a test rig has been used for assessment of timing chain wear performance of a 4 cylinder inline diesel engine on a light duty vehicle. A mineral, a semi synthetic and a fully synthetic oil have been used as a lubricant for chain wear evaluation. Each chain sample has been measured before test as initial status and during test for development of wear elongation trend at each particular test. The fully synthetic oil of SAE 5W30 specification which has Low SAPS technology performed the best chain wear elongation trend to meet targets, whereas the mineral oil of SAE 10W30 and semi synthetic SAE 5W30 oil have responded with similar but unacceptable wear elongation trends. Further tribological inspections are performed on SEM for visualization of wear characteristics. In agreement with wear measurements, SEM inspections demonstrated an abrasive wear on pins and bushes of chains tested with mineral and semi synthetic based oil while fully synthetic oil has just shown polishing wear. Same specification lubricants have separately been tested at real life engine environment on dynamometers. Similar elongation and common abrasive wear behaviour have been obtained for mineral and semi synthetic lubricants at dynamometer tests as well. Whereas fully synthetic lubricant has been found to significantly reduce elongation of timing chain and increase chain service life. Detailed inspections on running surfaces have demonstrated only polishing wear with very minor change in dimensions.

As a result of the investigations, following judgements can be truly considered;

- Fully synthetic oil which has SAE 5W30 Low SAPS technology presents the lowest chain wear and highest service life among tested oils.

- Standard SAE 5W30 specification semi synthetic oil and SAE 10W30 specification mineral oil has similar chain pin and bush surface wear structure, despite the slight increase of wear elongation with mineral oil.

- For assessment of effect of oil parameters on wear performance, a separate study should be performed in cooperation with oil suppliers as the tested oils have unique specifications and most of the lubricant specific information is confidential to manufacturers.

- Electric motor driven rig test is able to distinguish the performances of different oils under constant speed and loading conditions.

- Results observed from rig test have good capability in matching the performance seen on real life engine test.

- Rig tests can be used as a part of engine sub-system and component level design verification stage and offer a potential to reduce the costs of engine tests dedicated to wear performance of timing drive chains.

- A detailed investigation on running conditions of dynamometer durability test and determination of effective timing chain operating parameters are recommended for closer approximation of rig test results to real life engine test results.

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Ozay POLAT was born in Kocaeli in 1982. After graduation from Kocaeli Anatolian High School, he entered in Mechanical Engineering Program of Istanbul Technical University where he received his Bachelor in Science degree in 2004. He involved in the new 5 cylinder diesel engine development project at Ford Otomotiv Sanayii A.S. where he has been working in since 2004.