Chapter

Tribology in Marine Diesel Engines

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Abstract

This chapter deals with the tribology of marine diesel engines. Several types of diesel engines have been installed and used in the engine room of marine ships. Some of them, used for propulsion, operate at low-speed in a two-stroke combustion process in conjunction with propellers. Four-stroke engines are used for power generation and operates at medium-speed. In general, two or more four-stroke engines, including spares, are installed in the large ships. Tribological problems are important issue in the respect of reliability in the marine diesel engines, and there are many tribological engine components including bearings, pistons, fuel injection pumps and rollers. Moreover, the marine engines have lubricant problems such as lacquering. Improvements to the tribological performance of marine engine components, and lubricants can provide reduced oil and fuel consumption, improved durability, increased engines power outputs and maintenance. Therefore, this chapter shows better designs and methods in order to improve the tribological problem in the marine diesel engines.

Keywords: marine diesel engine, medium-speed diesel engine, low-speed diesel engine; fuel injection pump, lacquering; bearing, condition monitoring

1. Introduction

The marine diesel engine was first installed in the Selandia, which is oceangoing vessel, in 1912. According to statics, diesel engines used as a power source more than 95% in the ships of 2,000 tons or more [1]. Various types of engine used in the ship according to fuels such as diesel engine, gas engine and duel fuel engine. Among those marine engines, diesel engines have a large portion in marine engine market from the past to the present. Marine diesel engines burn refines but mostly residual fuels with a legislated maximum sulfur content between 0.0005% and 4.5% w/w sulfur. The intention is to reduce SO_x exhaust gas emissions for environmental reasons [2]. There has been a steady decline in NO_x emissions at an acceptable level due to the increasing number of rigorous exhaust legislations for marine diesel engines to minimize NOx emissions. Reduced NO_x emission can also be achieved by selective catalyst reduction (SCR), which is the most widely used and established technology [3]. Although ship pollution rules of IMO (international maritime organization) strengthened like IMO Tier III, the demand for engine performance improvement is also increasing.

Marine diesel engine types are two-stroke cycle and four-stroke cycle. The two-stroke engines are engine that complete a power cycle with two strokes of the piston during only one crankshaft revolution, and operate with about 100 rpm.

On the other hand, the four-stroke engines are internal combustion engines in which the piston complete four separate strokes during two crankshaft revolutions, and work in the speeds of 250 to 850 rpm. In case of large ships such as container ship, the two-stroke engines commonly installed as propulsion and the four-stroke engines operated as generators. This is because the two-stroke engines have higher torque and superior power-to-weight ratio than the four-stroke engines. However, the four-stroke engines used as propulsion in relatively small ship such as passenger boats and ferries.

Distillate oil and residual oil are the two most common types of marine fuels. "Intermediate oil" is the third type of marine fuel that combines the first two main types. In a refinery, petroleum fractions of crude oil from distillate fuel are separated by a boiling process. Residual fuel, also called 'tar' or 'petroleum pitch', is a fraction of the fuel that does not boil. Unwanted substances, such as chemical waste, are found in this type of fuel. Fuel system components, fuel injection equipment, pistons, piston rings, and cylinder liners can be damaged by these undesirable materials [4].

Various problems arise in the process of increasing the size of an engines or improving its performance. The solution and improvement of tribological problems related to friction wear and lubrication is very important in terms of reliability in marine diesel engines. This chapter present researches on machine components of marine diesel engines, which have tribological problems such as wear, sticking and oil consumption.

2. Tribological improvement of machine components in marine diesel engines

Many ship owners demanded from marine engine manufacturers to execute various technological modifications to increase the engine efficiency and to extend the life time of machine components. One of methods to accomplish these demands is to improve the tribological characteristics of machine components in the marine engines. In the Section 2, tribological improvement of machine components such as cylinder liners, fuel injection pumps, and bearings are explained. In addition, it includes the contents of lacquering that causes the tribological problems and the contents of machine condition diagnosis using the lubricant analysis.

2.1 Cylinder liner

The cylinder liners are a hollow cylinder shell which acts as the enclosure in which the combustion takes place. They have enough strength under the fluid pressure due to combustion and high-level stress induced in them. The cylinder liners are a cylindrical part to be fitted into an engine block to form a cylinder, and they are called "cylinder sleeve" in some countries.

They have main functions such as formation of sliding surface, heat transfer and compression gas sealing. The cylinder liners are served as the inner wall of a cylinder, and forms a sliding surface for the piston rings in order to reduce wear of piston, piston ring and oil consumption while supplying the lubricant in the clearance. The cylinder liners receive combustion heat through the piston and piston rings, and transmit the heat to the coolant. The cylinder liners are responsible to tolerate the combustion pressure, the contact with piston and piston rings while guiding the piston [5]. When the lubrication characteristics between the cylinder liner and the piston are poor, wear occurs on the lubricating surfaces, and scuffing phenomena also appear in severe cases.

Since there are few failure dates, it is difficult to estimate the reliability of the cylinder liners and ad hoc reliability tests are prohibitively expensive. Therefore, the liners are subjected to rigorous maintenance in order to reduce the frequency of failures while in operation. Both wear degradation and thermal cracking are the leading causes of failure of cylinder liners [6]. The significant number of abrasive particles on the piston surface generated by intensive combustion of fuel and lubricant deterioration is the primary wear process in this location [6]. A fatigue process caused by repetitive thermal shocks results in thermal cracking of the liner. As a result of insufficient chemical treatment, a thermal shock is generated by a fast temperature shift of the cooling fluid that laps the liner's exterior surface. Furthermore, this shock is typically exacerbated by scale and corrosion in the cooling water spaces [6].

The cylinder liner-piston ring system is one of the major contributors for the mechanical losses in marine diesel engines [7]. Lubrication of cylinder liner is very important for any marine diesel engines because it controls the wear and enhance the life of the engine. In general, lubrication performs three purposes. First, it prevents metal-to-metal contact between the cylinder liner and piston rings by creating an oil layer. Second, it neutralizes sulfuric acid and controls corrosion caused by detergent. Third, it cleans the cylinder liner, especially the piston ring pack, and prevents failures and corrosion due to neutralization residues and fuel combustion. So, several researches related to lubricant of cylinder liners are performed. Measurement of iron ppm in the cylinder drain oil was used to assess the degree of cylinder wear. Moreover, the role of alkaline additive in protecting surface of the metal from the acid was estimated in cylinder liner. That is, linear wear rates were estimated as a function of sulfur contents in fuel, feed rate and alkalinity of lubricant [8].

Surface texture of the cylinder liner identified as one of the significant factors that improve the tribological properties of marine diesel engine with regard to wear, oil consumption, fuel efficiency, lubrication oil (wear particle concentration). Surface texture is one of technical methods which alter a material's surface by modifying its texture and roughness. Various surface texturing methods employed such as laser surface texturing (LST), pulsed air arc treatment, electropolishing, reactive ion etching (RIE), photochemical machining, maskless electro chemical texturing, micro-computer numerical control (CNC) texturing. Among the different surface texturing techniques, LST is one of the most widely used non-contact type method and **Figure 1** show the cylinder liner with surface texturing. The effect of partially LST is evaluated through testing of marine diesel engines. The ellipse patterns oriented at the sliding direction contributed the most to reduce the friction coefficient [7].

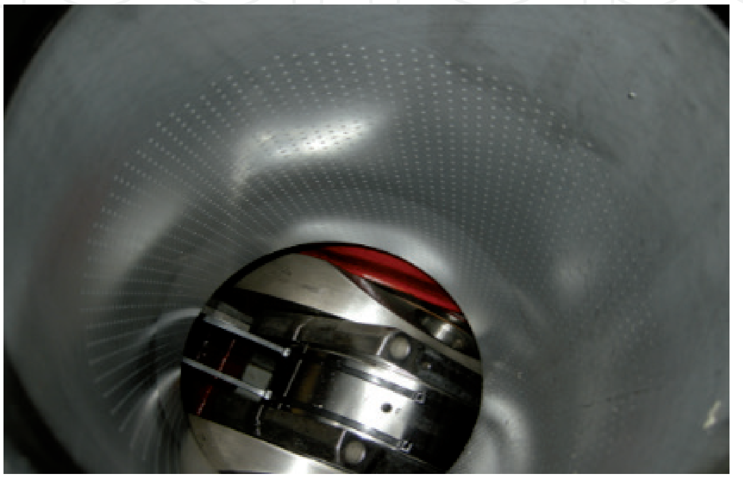

Figure 1.
Surface textured cylinder liner.

Generally, the honing applied to inner wall of the cylinder liners as shown in **Figure 2**. The function of the honing to hold a satisfactory amount of lubricant oil and reduce friction. The honing process is applied to get surfaces with good functions for the ring or liner contact. The honing tool is comprised of a number of honing stones, which are similar to grinding wheel [9]. The surface of honed cylinder liner is characterized by a negative skewness, R_{sk} (S_{sk}) and kurtosis, R_{ku} (S_{ku}) values higher than 3. The honing is comprised of smooth wear resistant plateaus with deep valley, which have the effect of oil reservoirs and traps for wear particles [10]. Moreover, the selection of honing machine depends on the honing process (vertical or horizontal honing), honing angle, depth of honing, bore diameter, stroke length. The processing condition of honing should be selected according to engine's output, engine type, and engine manufacturer recommendation.

2.2 Fuel injection pump

Fuel injection pump is one of the main components of the marine diesel engines. The fuel pump is device that supplies compressed fuel into the cylinders of the diesel engines, and controls the amount of fuel oil needed to gain the desired power. Moreover, it operates with a timing that keeps the engine running smoothly. The mechanical reciprocating fuel pump which used in marine engines is referred to as a "Bosch type pump," and it consists of a barrel and a helical plunger as shown in **Figure 3** [11, 12].

Various factors, such as lacquer, foreign debris, and insufficient clearance, can block the plunger inside the barrel and cause a 'stick' problem in most fuel pumps. **Figure 4** shows how the development of the lacquer in the pump limits the space between the plunger and the barrel [11]. Several researches tried to solve and improve the stick problem as follows:

- Optimal design of clearance between the plunger and the barrel
- Application of taper shape and grooves
- Design of cross-sectional shape in groove.

Figure 2.
Honing in cylinder liner.

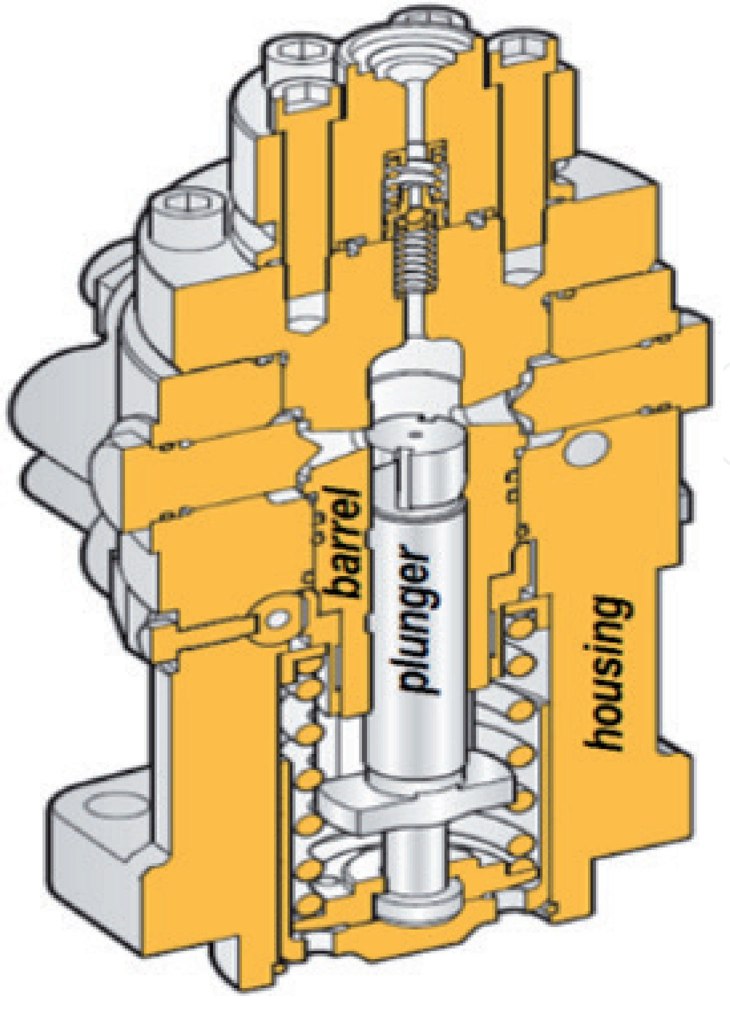

Figure 3.
Fuel injection pump [11].

- Prevention of lacquering

In this section, researches on optimal design of clearance, and application of taper shape and grooves focused among several methods. Grooves applied various machine components have several functions such as reduction of friction loss, oil reservoir and trapping of wear particles [13]. Trap effect of groove investigated with

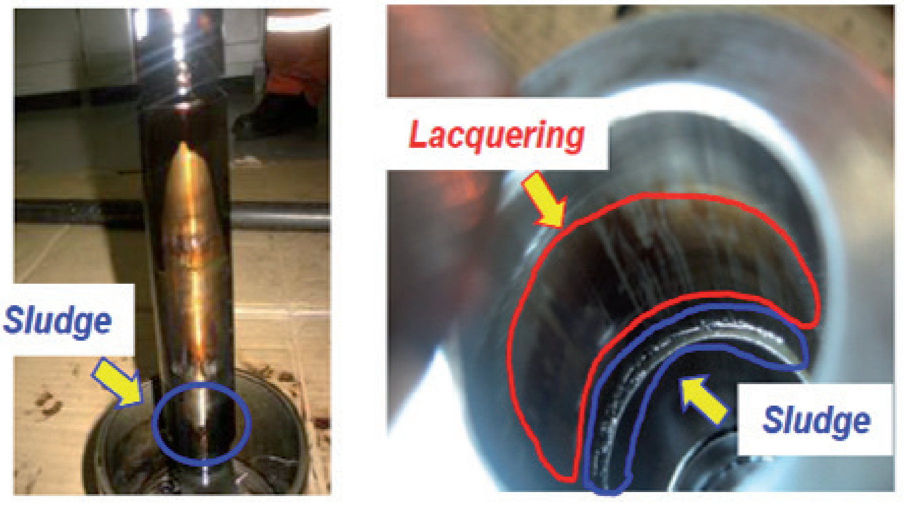

Figure 4.
Lacquer formation in fuel pump [15].

variation in cross-sectional shape and Reynolds number. The effect of groove is evaluated using computational fluid dynamics (CFD) analysis. The simulation based on the standard k-ε turbulence model and the discrete phase model (DPM). The simulation results are also capable of showing the particle trajectories in flow field. Various cross-sectional shapes of groove such as rectangular, triangle, U shaped, trapezoid, elliptical shape evaluated. Through the CFD analysis, it found that the cross-sectional shapes favorable to the creation of vortex and small eddy current are effective in terms of particle trapping effect [13]. In particular, the residual fuel used in marine diesel engines contain relatively many foreign materials or impurities, so the design of groove which have a good trap effect needed to prevent the three body abrasive wear and sticking. Prevention of lacquering explained in Section 2.3.

2.2.1 Optimal design of clearance

Regarding the lubrication characteristics of lubricating system, the influence of the clearance is significant, and many studies have been performed on the effect and design of clearance in terms of varieties of bearing and joints. The clearance between the plunger and barrel in fuel injection pump is also important to keep stable operation for marine diesel engines. In marine diesel engines, high viscosity fuel oils such as heavy fuel oil (HFO) and low viscosity fuel oils such as light diesel oil (LDO) have been used, and several lubrication techniques have been used depending on the fuel oil viscosity. The supplied fuel and lubricant oil lubricate the system when low-viscosity fuel oil is used, while only the fuel oil lubricates the system when high-viscosity fuel oil is used. Therefore, it is necessary to test the pump's lubrication characteristics at different viscosity levels. Furthermore, since the pump may work at high pressures of up to 1,500 bar, deformation should be taken into account. Due to the substantial effect of restriction circumstances on pump deformation, structural analysis is adequate for clearance design when maximum fuel oil supply pressure is applied. In addition, the highest reduction in clearance is used solely as a design limit.

The clearance in the fuel injection pump is estimated by structural and hydrodynamic lubrication studies. A structural study looks at structural changes in the plunger and barrel when they are subjected to maximum supply pressure due to fuel oil compression. Furthermore, the structural study also evaluates the maximum reduction in clearance caused by deformation. As viscosities and clearances vary, the hydrodynamic study of lubrication analyzes the lubrication properties. The clearance between the plunger and the barrel is divided into two sections, head and stem as shown in **Figure 5**. The lubrication characteristics are then compared by calculating the film parameters from the minimum film thickness and surface roughness, as shown in Eq. (1). The surface roughness is determined from the surface roughness of the plunger and barrel, as shown in Eq. (2).

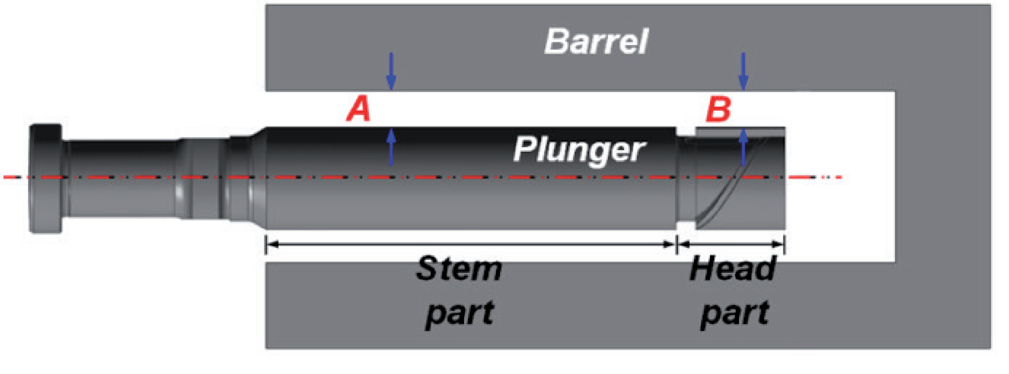

Figure 5.
Dimensionless clearance of stem part (A) and head part (B) [11].

$$Film\ parameter(\lambda) = \frac{Minimum\ film\ thickness\,(h_m)}{Surface\ roughness\,(R_q)} \tag{1}$$

$$R_q = \sqrt{R_{q1}^2 + R_{q2}^2} \tag{2}$$

Figure 6(a) shows the positions of three sections where the deformations of the barrel and plunger have been evaluated. The deformation (enlarged 300 times) in the top, middle, and bottom sections is illustrated in **Figure 6(b)–(d)**. The ratio of displacement to clearance of the stem in the plunger is represented by a dimensionless displacement, which is a dimensionless value. Since the study is conducted in the absence of a clearance condition, overlapping regions indicate a decline in clearance. This is because the primary orientations for the barrel and the plunger are virtually perpendicular, and the spill port of a barrel has a significant impact on deformation. The clearance reduction is assessed in two parts. In the head, the barrel's and the plunger's deformations are studied. However, deformation of the plunger is investigated only in the stem as a low film pressure does not distort the plunger barrel. As shown in **Figure 7**, the deformation of the pump is studied quantitatively to determine the highest reduction in clearance. This figure also shows that the plunger's centerline and the barrel's line have a dimensionless displacement. The dashed line represents the displacement of the barrel, and the solid line represents that of the plunger. When deformation of the plunger is larger than that of the barrel, a reduction in clearance at the head occurs. **Figure 8** shows the dimensionless displacement at the red line of the plunger. In **Figure 8**, the region to the left of the pink dashed line is the area in which the plunger is inside the barrel during reciprocating motion. The maximum reduction in the clearance is the maximum displacement in the stem part. The structural analysis found that the clearance should be higher than the maximum clearance decrease to avoid metal-to-metal contact between the barrel and the plunger due to deformation.

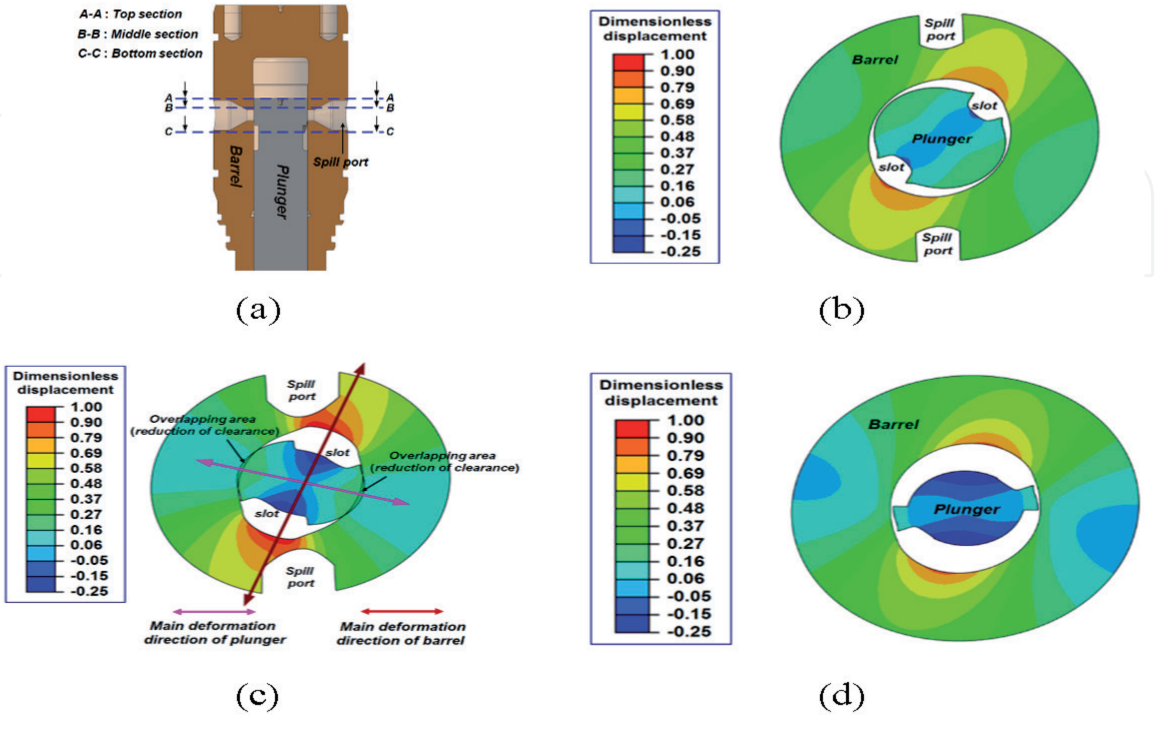

Figure 6.
Dimensionless displacement on the three sections [11]. (a) Position of three sections in the axial direction (b) displacement of top section (c) displacement of middle section (d) displacement of bottom section.

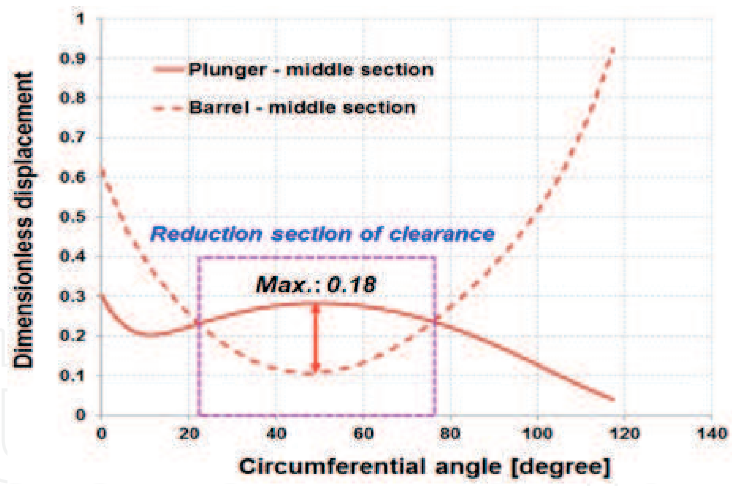

Figure 7.
Dimensionless displacement of barrel and plunger in the middle section [11].

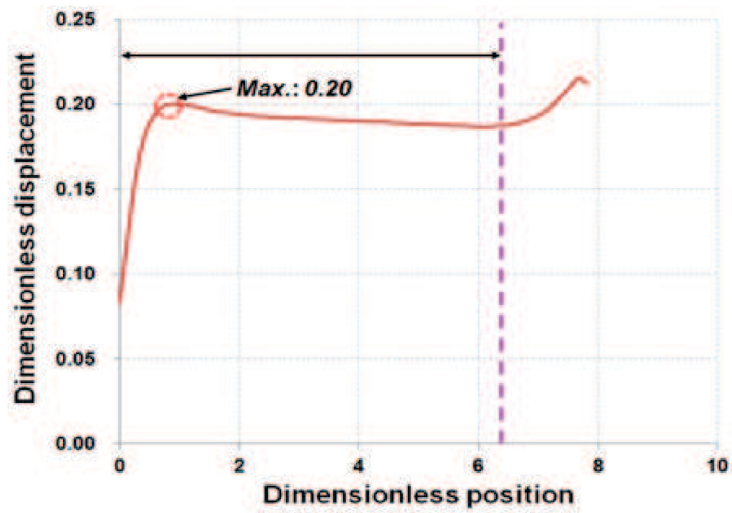

Figure 8.
Dimensionless displacement of plunger in stem part [11].

In hydrodynamic lubrication analysis, an unsteady-state two-dimensional Reynolds equation is used to model the fluid film between the barrel and the plunger, and the Reynolds boundary condition is applied. Moreover, the equilibrium equation of the moment at fixed point and the equilibrium of the forces in the vertical and horizontal directions are used. The lubrication characteristics of the pump with variation of clearance are investigated in two parts, head and stem. When the dimensionless viscosity is 0.57, **Figure 9(a)** shows the dimensionless minimum film thickness with a change in the dimensionless clearance of the head. Furthermore, when the clearance in the stem is constant, the film parameters do not change to be more than a specific clearance in the head. The film parameters obtained according to the results of the minimum film thickness are shown in **Figure 9(b)**. **Figure 10** shows the changes in dimensionless viscosity and dimensionless clearance of the film parameter. Increments in the dimensionless viscosity raise the film parameter. If a film parameter of 3 or higher shows good lubrication properties, the pump's lubrication qualities are satisfactory throughout a wide range of viscosities. While in a low viscosity state, the film parameter is less than 3 when the dimensionless clearance of the stem is 0.25, 0.32 and 0.37. This means that, at low viscosity, metal-to-metal contact between the plunger and the barrel is possible. However, a low viscosity situation results in a parameter of 4.25 for a dimensionless

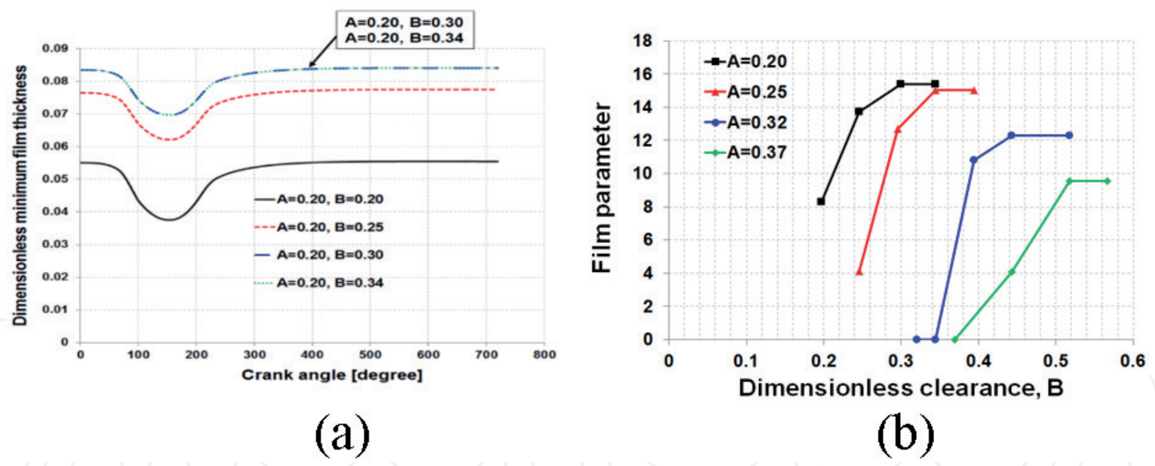

Figure 9.
Dimensionless minimum film thickness and film parameter with dimensionless clearance (B) [11] (a) dimensionless minimum film thickness (b) film parameter.

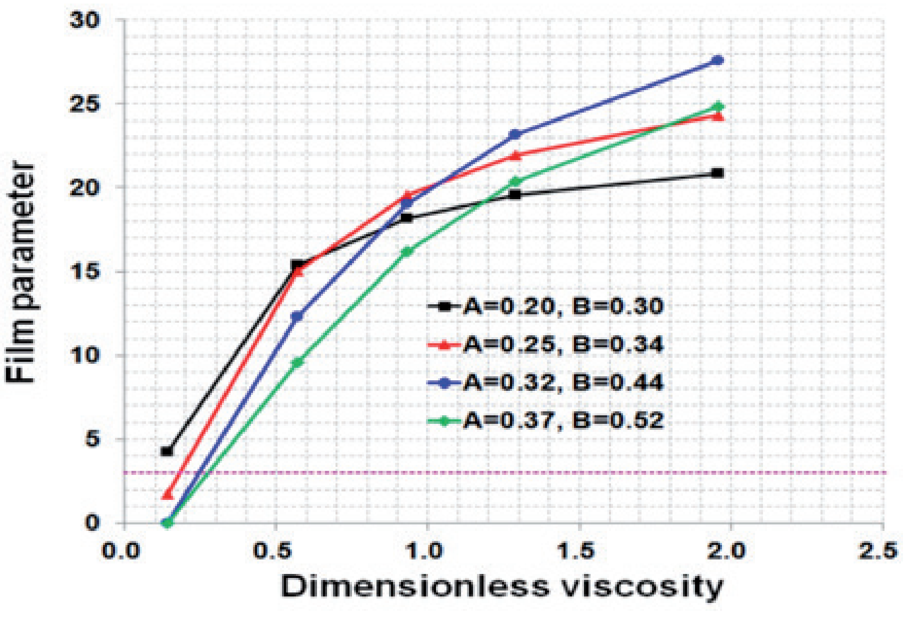

Figure 10.
Film parameter with dimension viscosity and clearance (A, B) [11].

clearance of 0.20. Since the stem of the reciprocating fuel pump system has a relatively large lubrication area, the dimensionless clearance of the stem must be small, and the clearance in the stem had a significant impact on the lubrication properties of the system. According to the hydrodynamic lubrication study of marine diesel engines, metal-to-metal contact does not occur when the stem clearance is modest and the head clearance is more than a specific clearance.

The tolerance and machining limit of the clearance are also required to establish the pump clearance. Since the tolerance of the clearance must be caused by the machining process, and the clearance is decided by two surfaces, the clearance in a genuine mechanical system is not a single number but rather a range of values. Furthermore, the manufacturer's processing capability is also taken into account while determining the clearance. Therefore, in this chapter, this concept is referred to as the "machining limit of the clearance".

In the scenario where the dimensionless clearance machining limit is 0.17 and the dimensionless tolerance maximum is 0.05, the dimensionless clearance of the pump is determined. To prevent metal-to-metal contact, the ideal dimensionless clearance in the stem and head should be greater than 0.18 and 0.20, respectively.

Assuming that the hydrodynamic lubrication analysis shows that the hydrodynamic lubrication characteristics are good when A = 0.20 and A = 0.30, the dimensionless clearance of the stem (A = 0.20) will be inadequate because the maximum dimensionless reduction in clearance induced by deformation is greater than the dimensionless clearance. Therefore, to prevent metal-to-metal contact despite having good lubrication characteristics of the pump for the hydrodynamic lubrication analysis, the dimensionless clearance must be greater than 0.20 for the stem part. However, since the dimensionless clearance is larger than the maximum dimensionless reduction in the clearance, the dimensionless of the head (B = 30) is correct. Therefore, the dimensionless clearance is determined while considering the maximum dimensionless tolerance, as shown in **Table 1**. However, there are a few limitations to this method. The disturbance generated by the cam was not taken into account when calculating the reciprocating motion. Moreover, a variety of fuel oils are used in marine diesel engines, and changing fuel oil causes a stick in the fuel pump. In this approach, the changes in fuel oil were not taken into account. Furthermore, this method is more accurate than the fluid–structure interaction (FSI) method even though structural and hydrodynamic lubrication analyses are used to estimate the proper pump clearance. Although this design method of clearance has some significant drawbacks, it improved the fuel pump durability and made it feasible to create engines with a 20% increase in power [11].

2.2.2 Application of taper shape and grooves

The grooves were used to improve the lubrication performances and heat transfer in various mechanical components. In addition, profiles (taper shape) were applied to cylindrical roller bearings to release the stress concentration at the both ends of the rollers. Previous study has suggested application of partial grooving, circumferential grooving, taper shape, and the design of optimal clearance to improve the lubrication characteristics of the fuel pump in marine diesel engines. The hydrodynamic lubrication analysis of the fuel pump performed by using two-dimensional Reynolds equation and Reynolds boundary condition to compare lubrication characteristics of the pump with variation of taper shape, groove condition, and viscosity.

Figure 11(a) shows a tapered plunger, with tapered applied to the plunger stem. In the axial and radial directions, the dimensionless taper lengths of the upper section of the plunger stem are A1 and B1, respectively. C1 and D1 are the dimensionless taper lengths of the lower section of the plunger stem, respectively. A grooved plunger is shown in **Figure 11(b)**. L_1 is the dimensionless distance between the stem's edge and the first groove. Dimensionless groove width and depth are represented by L_2 and H_2, respectively. L_3 is dimensionless distance between grooves, and N is number of grooves. The depth of groove is considered in film thickness equation in case shallow groove. On the other hand, the pressure in the deep groove calculated using the continuity of flow rate.

Dimensionless maximum reduction of clearance	Head part	0.18
	Stem part	0.20
Dimensionless machining limit of clearance		0.17
Dimensionless clearance of the pump	Head part	$0.30 \leq C \leq 0.34$
	Stem part	$0.21 \leq C \leq 0.25$

Table 1.
Determination of optimal clearance [10].

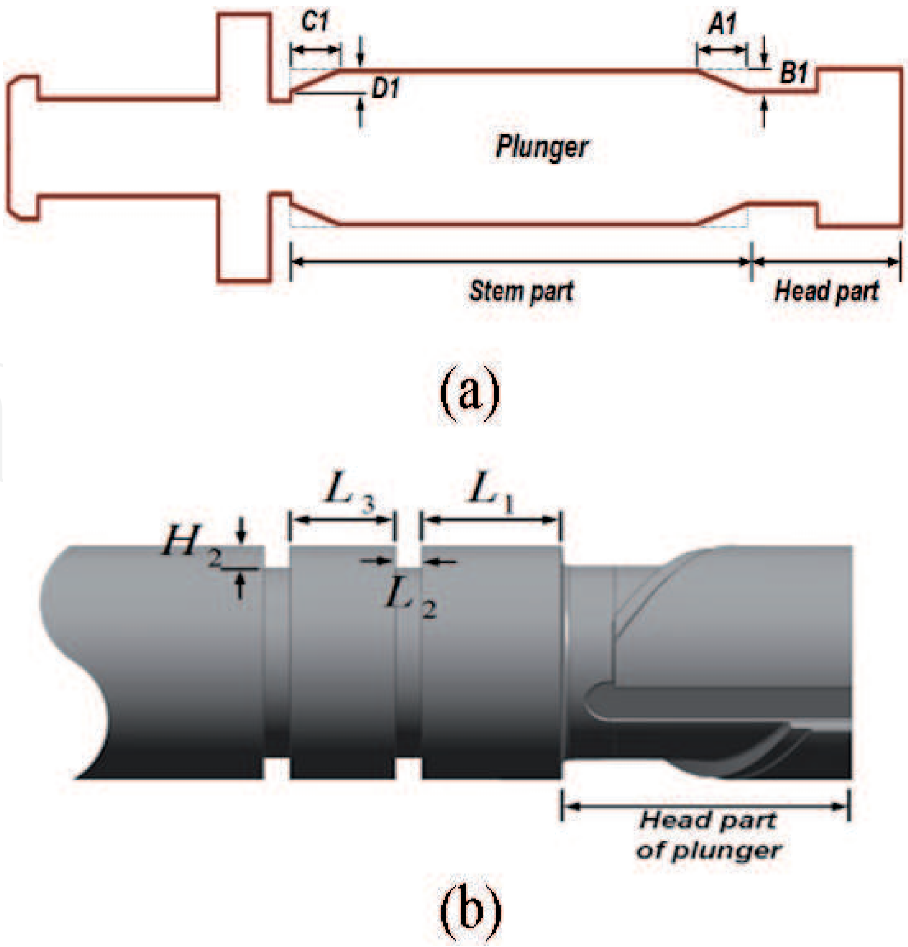

Figure 11.
Geometries of plunger with taper and groove [12]. (a) Taper shape (b) circumferential groove.

The taper to the stem part of the plunger applied to improve lubrication characteristics of the pump by using a pressure generation by the wedge effect. The generated pressure helps to restore the plunger to the barrel center during reciprocating motion. **Figure 12(a)** and **(b)** show the effect of tapering the upper part of stem. The dimensionless minimum film thickness in the tapered plunger is greater than that of an untampered plunger. When A2 increased from 0.312 to 0.938, the dimensionless minimum film thickness increases by roughly 15%. However, the minimum film thickness does not change when the A1 increases beyond 0.938. Furthermore, the change in dimensionless minimum film thickness differs by less than 50% compared to the variation of B1. According to **Figure 12(c)**, tapering the bottom section of the stem does not affect lubricating properties. During reciprocating action, the lower section of the stem pops out of the barrel for a short period due to reduced fluid film pressure in the bottom part of the stem.

The imbalance in pressure can be alleviated by applying a circumferential groove, which allows passage in the circumferential direction. These grooves equalize these pressures and restore the plunger to the center of the barrel by facilitating flow around the periphery of the plunger from the high-pressure zone to the low-pressure zone. **Figure 13** shows the film parameter with dimensionless viscosity and groove type (shallow groove, deep groove). The film parameter in the case of shallow grooving is greater than that of deep grooving. The percentage of increase in the film parameter calculated based on the film parameter of no groove condition. A plunger with a shallow groove improves the lubrication properties more effectively in low-viscosity circumstances, since deep grooves are less efficient in high-viscosity conditions because of increased viscous friction. However, it is difficult to

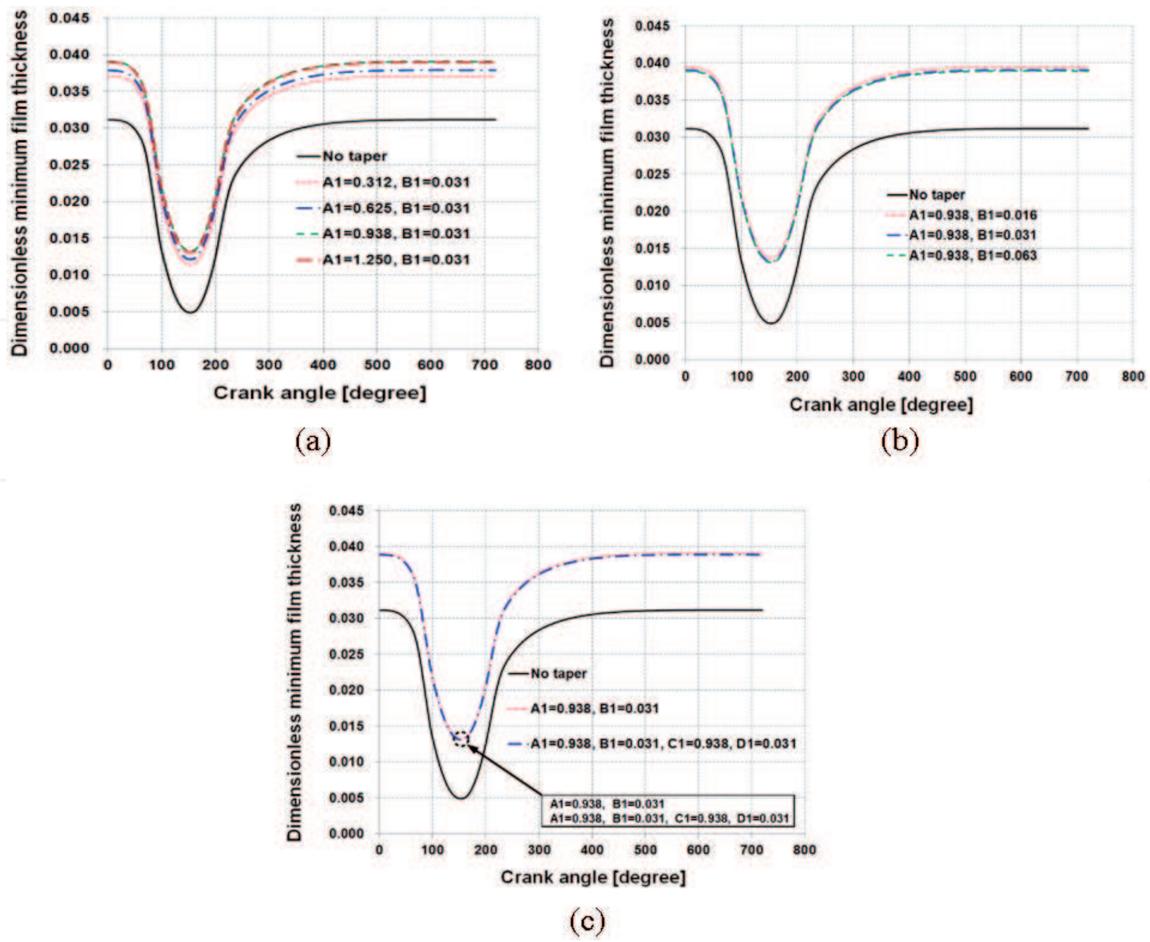

Figure 12.
Dimensionless minimum film thickness with taper geometries and crank angle [12] (a) variation of A1, (b) variation of B1, (c) effect of C1 and D1.

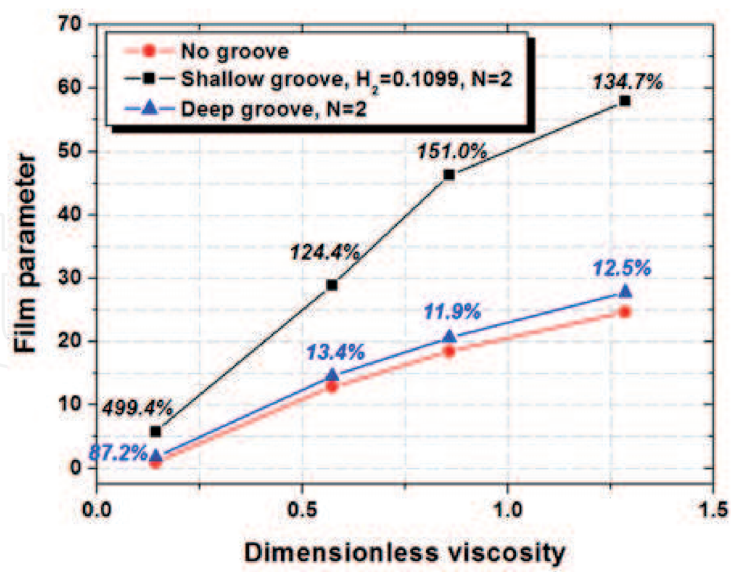

Figure 13.
Film parameter with groove types and dimensionless viscosity [12].

apply circumferential grooves with shallow depth because of the economics of the product for processing, and the reduction in ability to trap wear particles.

The effect of application both circumferential groove and taper shaper to the plunger investigated. **Figure 14** shows various types of plunger. **Figure 15(a)** compared the lubrication characteristics of the plunger applied both taper and grooves with those cases where either are applied singly. A pump's film parameter is highest

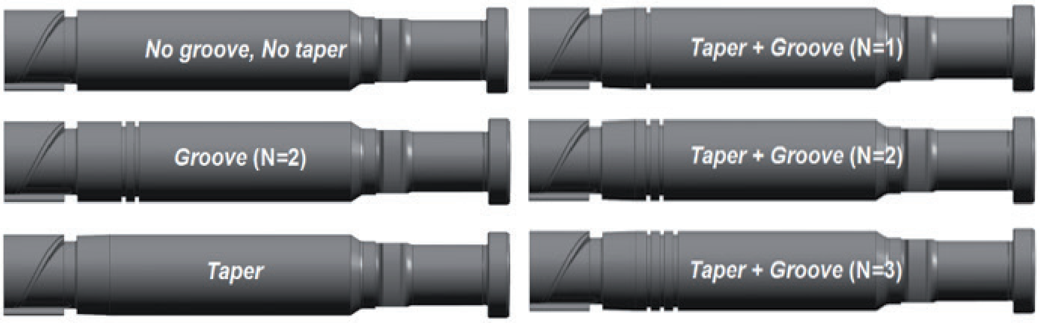

Figure 14.
Types of plunger [12].

when both taper and groove are used. In the low viscosity condition, the absence of groove and taper increases film parameters by about 390%. Furthermore, the use of a taper is more successful than the use of grooves in improving lubricating qualities. As shown in **Figure 15(b)**, the film parameter varies when the number of grooves is changed in the case of taper + groove. The difference in film parameters is less than 4% between N = 2 and N = 3. However, the lubricating properties of a pump with three or more grooves are not improved as the pressure imbalance occurs mainly in the head and upper part of the stem [12].

Besides, research on fuel pump with spiral grooves has performed in order to improve the durability of the fuel pump in marine engines. The application of spiral grooves is quite effective in the design of fuel pumps requiring high pressure for high power of the engines. This is because a spiral groove is one continuous groove and can effectively release uneven pressure distribution surrounding the plunger. In addition, spiral grooves are not machined to the both ends of the stem part because the grooves cause a pressure drop of compressed fuel [14].

2.3 Lacquering

Environmental restrictions affect the composition of marine fuel oil and the design of marine diesel engines. To significantly limit the sulfur content of fuel oil and emission, there is a requirement to strengthen environmental laws. Low-sulfur fuel oil is used in most marine engines to meet international environmental requirements. However, the use of such fuel oils can cause unexpected concerns. These concerns include higher lubricant consumption due to lacquer build-up on the cylinder lining, as seen in **Figure 16**.

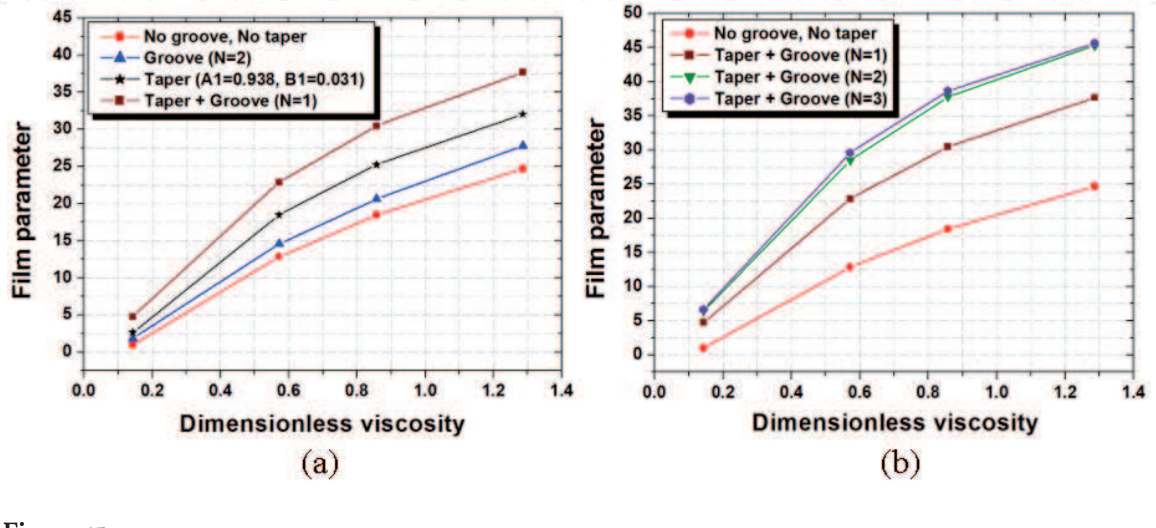

Figure 15.
Film parameter with dimensionless viscosity (a) variation of type (b) variation of N [12].

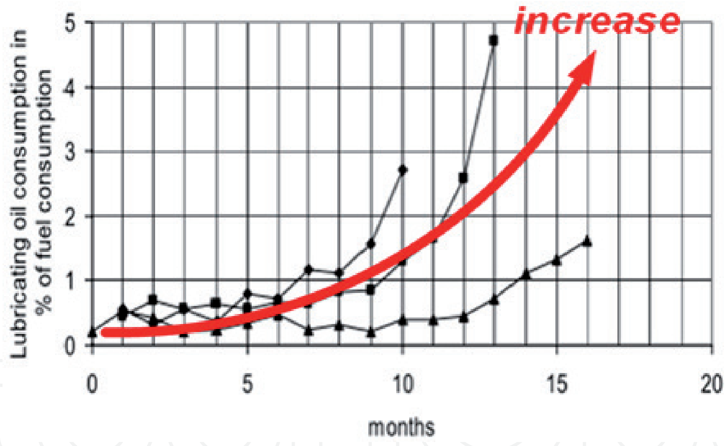

Figure 16.
Oil consumption increase in engines [15].

Lacquer forming (lacquering) in the cylinder liners of marine diesel engines has been a matter of concern for at least 20 years. Lacquer development increases lubricating oil consumption, sticks the injection pump, and causes scuffing in the cylinder liner. Cylinder liners are the most essential engine components when it comes to oil consumption and friction losses. According to studies, friction between the cylinder liner and the piston ring is responsible for up to 40% of engine friction losses. The surface of the cylinder liner consists of a mixture of deep enough valleys and smooth plateaus, which is called honing, in order for the liner to hold a satisfactory amount of lubricant oil and to reduce friction. **Figure 17** shows that cylinder liner lacquer results in deposits in the grooves. Such deposits reduce clearance to the point of contact between plunger and barrel. Sticking can occur due to reduced clearance, the lubrication characteristics of the pump are deteriorated [15, 17].

Previous studies have speculated on lacquer producing methods. Shell [18] and Alberola [19] have presented two proposals.

According to Alberola, lubricant oils are liquid polymers with low molecular weight; therefore, deposit formation due to thermal and oxidative degradation of these oils can be considered a thermosetting process. In this process, a polymeric liquid undergoes two macroscopic phase transformations, gelation and vitrification, which turn the liquid into a solid. Gelation is the production of branched molecules with a potentially infinite molecular structure that occur at a critical point in a chemical reaction. First, a paste-like gel-like coating is initially formed from the lubricating oils. Then, a vitrified or glassy solid is formed from thermosetting

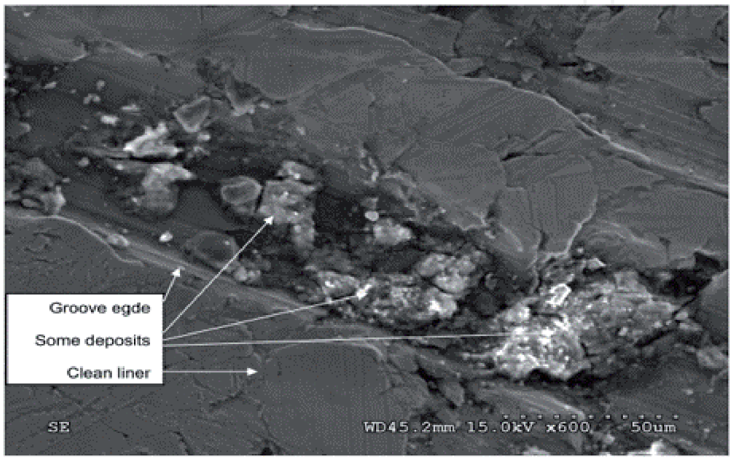

Figure 17.
Deposits in the groove of cylinder liner [15, 16].

polymers in the thermosets. During this phase, the polymeric network becomes tighter due to the chemical cross-linking processes that continue to take place. Because the thermoset structure has transformed to a vitrified glassy state, molecular segment motions are no longer feasible. Finally, the pasty properties are lost in the glassy deposit, which is commonly referred to as a lacquer or varnish [18].

According to Shell, condensation of partially combusted and cracked fuel components on the surface forms lacquer (**Figure 3**). To form the layer, these components oxidize and polymerize before mixing with the calcium and zinc salts of lubricant oil. These metal components act as catalysts in the oxidation of the surface. The layer turns into a hard glaze under high temperatures. This process results in the formation of hard and glassy layers on the surface [19].

However, the two proposals have yet to be confirmed by detained chemical analysis of the lacquer [15, 17].

There are a variety of reasons for the formation of lacquer. Lacquer can be formed due to the use of only fuel oil, or a mixture of fuel oil and lubricant oil. The boiling point and aromatic content of fuel oils also affect the formation of lacquer. Compared to fuel oils that do not form lacquer, lacquer-forming fuel oils have higher aromatics and paraffinic contents [20, 21]. A higher than normal final boiling point may indicate higher than normal content of polycyclic aromatic hydrocarbons (PAHs) in the fuel oils. Past work has also suggested that distillate fuels containing heavier ends are more prone to form lacquer [22]. The base number (BN) level of lubricant oils and sulfur content of fuel oils are directly related to lacquer formation [21]. The marine diesel engines are normally designed to burn residual fuel oils containing high-level sulfurs, and need lubricant oils with an appropriate level of BN to neutralize the corrosive combustion acids. However, higher BN and sulphated ash indicated a higher deposit risk. In addition, engine tests that the lacquer increases when either the liner temperatures or inlet air temperature are too low. This is because the low temperatures favor conditions for condensation of partially combusted and/or heavier fuel ends on the surface. Operation with a lot of idle, part-load, or combined full load (or over load) operation seems to be the most lacquer-prone.

There are many variations in the appearance of the lacquer under different conditions. Normally, amber and brown, lacquer appears darker when viewed from an angle, probably because more light is reflected from the surface where most of the deposit is located. Moreover, the term "glazing" is used to describe the appearance of lacquer. The lacquer has a strong bonding force with the surface, so it is not easily physically removed. The degree of bonding force has been evaluated through pull-off test. The allowable criteria of marine paints specified in ISO 4624 is about 3 ~ 4 MPa. The pull-off pressure of lacquer were over 9 MPa, and the values are two or more times larger than the allowable criteria of marine paints.

Anthranequinone and other quinones are also insoluble in most solvents, but they are soluble in acids, such as sulfuric acid and acetic acid as shown in **Figure 18**. The presence of quinones in the lacquer would explain why the lacquer dissolves readily in a weak organic acid. Through this phenomenon, acids are effective in removing the lacquer.

The lacquer can form due to a variety of causes, so there are various measures that can be taken to prevent or minimize the problem. The maintenances of fuel injectors, turbocharger and cooling around the liners are effective factors in preventing incomplete combustion, and also influence the prevention of lacquer formation. To prevent lacquer formation in the fuel pump, systematic control of the lubricant oil flow and periodic inspection of the pump were suggested, in order to ensure replacement of the sealing ring, and oil sediment removal. Other countermeasures are use of alternative lubricants, and of multifunctional fuel additives.

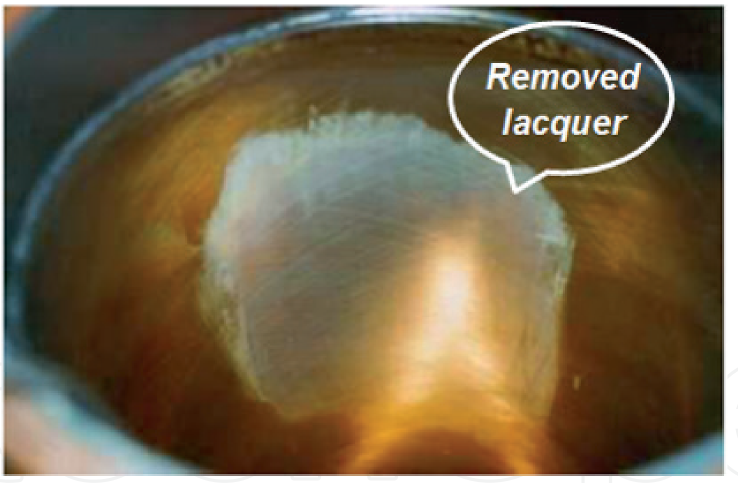

Figure 18.
Liner lacquer after partial cleaning with acetic acid [17].

Several companies have proposed an “advised range” for the BN depending on the sulfur content of fuel oil to prevent lacquer formation [17].

2.4 Bearings

The development of marine diesel engine was high-power, compact structure, and hence the bearings were required to work normally under smaller size, high load, and thinner oil film conditions. **Figure 19** shows big end bearings of connecting rod. In recent years, coupling simulations between elasto-hydrodynamic lubrication (EHL) and nonlinear multi-body dynamics (MBD) are carried out to dynamically-loaded bearings of marine diesel engines, and the coupling analysis is an effective method to investigate the lubrication characteristics of marine bearings [23].

In a marine engine, the connecting rod bearing is a key friction component. It converts the reciprocating action to rotary motion by connecting the crankshaft and pistons, as well as the cross-head slide. The bearing of the connecting rod significantly affects engine performance by improving its reliability, durability, and strength. To analyze the EHL for the large end connecting rod bearing of a low-speed two-stroke marine diesel engine, AVL Excite Power Unit software was used.

Figure 19.
Big end bearing of medium-speed diesel engine.

This software considers nonlinear multibody dynamics. Bearing lubrication can be calculated more accurately if the following factors are taken into account: friction surface roughness, elastic deformation of the bearing and journal, oil supplying qualities, and the influence of the cavity on the oil film lubricant [24].

The wear caused by insufficient lubrication is the most general cause of endurance life issues. An absence of lubrication in the journal-bearing system leads to bearing seizure, and normally, to total destruction of the part. Insufficient lubrication caused by factors such as a machining error in the manufacture of the crank pin and the bearing leads to metal-to-metal contact between the crank pin and the bearing, which results in adhesional wear. The crank pin bearing which connects the connecting rod and crank arm, converting a reciprocating motion into a rotary motion plays an important role in a marine diesel engine. Through the motion analysis of the piston-connecting rod-crank arm system, the bearing loads and lubricant velocity were calculated. The numerical algorithm for the hydrodynamic lubrication analysis coupling with the motion analysis of the piston-connecting rod-crank arm system developed to investigate lubrication characteristics. The maximum film pressure decreased with decreasing clearance and lubricant temperature, and that film thickness increased with decreasing clearance and lubricant temperature. The lubricant temperature had a higher effect on the film thickness than the clearance [25].

Fretting is phenomenon that concerns mechanical components in contact that are designed to be fixed but undergo small relative displacement due to fluctuation loads. The fretting is one of main issues of connecting rod bearing in marine diesel engines. **Figure 20** shows fretting fatigue fracture between bearing bush and small end. The fretting damage begins with local adhesion between mating surfaces and processed when adhered particles are removed from the surface, they may react with air or other corrosive environments. Surface crack can be initiated by fretting, and led to catastrophic failure after crack propagation. The fretting influenced by contact pressure, friction coefficient and relative slip motion. **Figure 21** shows numerical results about contact pressure, tangential stress and slip amplitude at certain crank angle. The fretting severity on mating surface is evaluated as fretting damage parameter (FDP). The FDP is defines in Eq. (3). The τ is the frictional shear stress at the interface and δ is the absolute slip amplitude in the tangential stress direction in Eq. (3). The potential for fretting initiated fatigue fracture on the mating surface is also evaluated using the Ruiz criterion and defined as fretting fatigue damage parameter (FFDP) in Eq. (4). The σ is the tensile tangential stress on the contact surface in Eq. (4). The greater slip amplitude and tangential stress can increase the possibility of fretting fatigue damage. Moreover, the contact pressure at the mating surface is not an effective parameter to predict the fretting damage because the areas where contact pressure is high, the FDP and FFDP are close to zero.

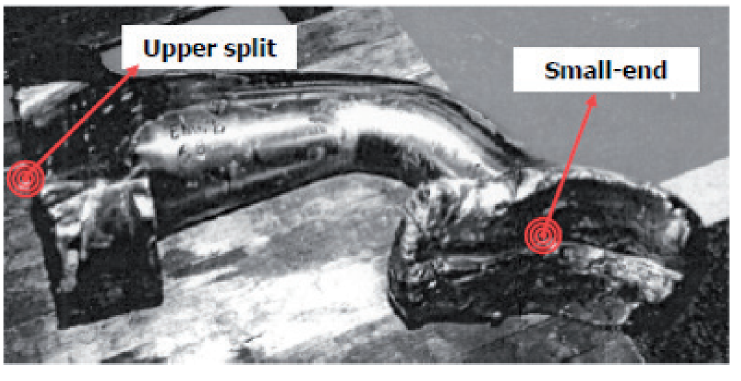

Figure 20.
Fretting fatigue failure of connecting rod [26].

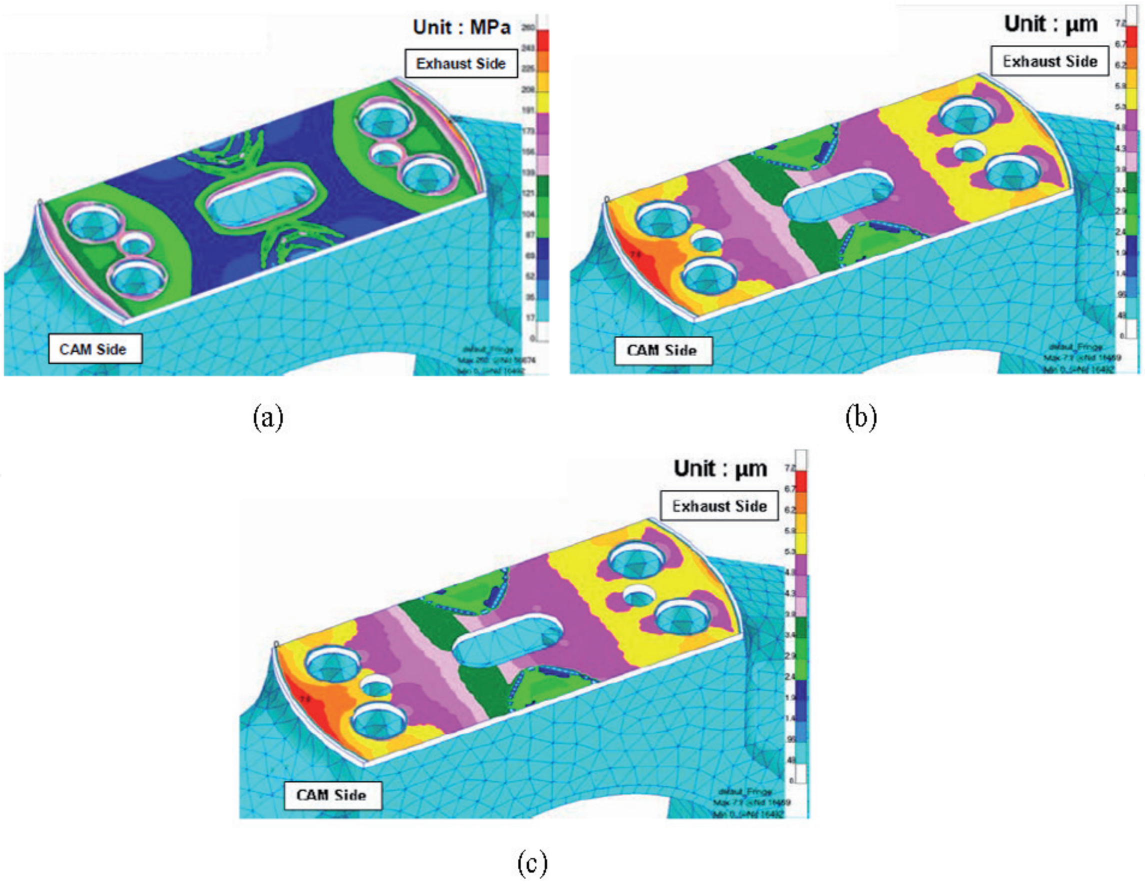

Figure 21.
Numerical results with AVL software [26]. (a) Contact pressure distribution at crank angle 490° (b) tangential stress distribution at crank angle 110° (c) slip amplitude distribution at crank angle 310°.

Therefore, the possibility of fretting damage at the upper split in connecting rod bearing was investigated using the Excite software from AVL and Ruiz criterion [26]. The Ruiz criteria is an effective empirical approach for evaluation of fretting fatigue damage parameter and has been demonstrated in two dimensional fretting studies of a typical dovetail interface problem. Moreover, this criterion is suitable for predicting the fretting damage of the connection rod bearing in marine diesel engines.

$$FDP = \tau \bullet \delta \tag{3}$$

$$FFDP = \sigma \bullet \tau \bullet \delta \tag{4}$$

Aside from crank pin bearings and connecting rod bearings, various bearings are used in marine diesel engines. Since the size of most bearings are large, it is difficult to carry out experimental studies due to cost and volume, so most researches on bearings are focusing on analytical studies. However, in order to improve practically lubrication characteristics of bearings in marine diesel engines, more experimental studies should be conducted simultaneously.

2.5 Condition monitoring with oil analysis

Maintenance strategies play a crucial role in reducing the cost of down time and improving system reliability. Consequently, machine condition monitoring plays an important role in maintaining operation stability and extending the period of usage for various machines. Machine condition monitoring through oil analysis is an effective method for assessing machine's condition and providing early warnings regarding a machine's breakdown or failure as shown in **Figure 22**.

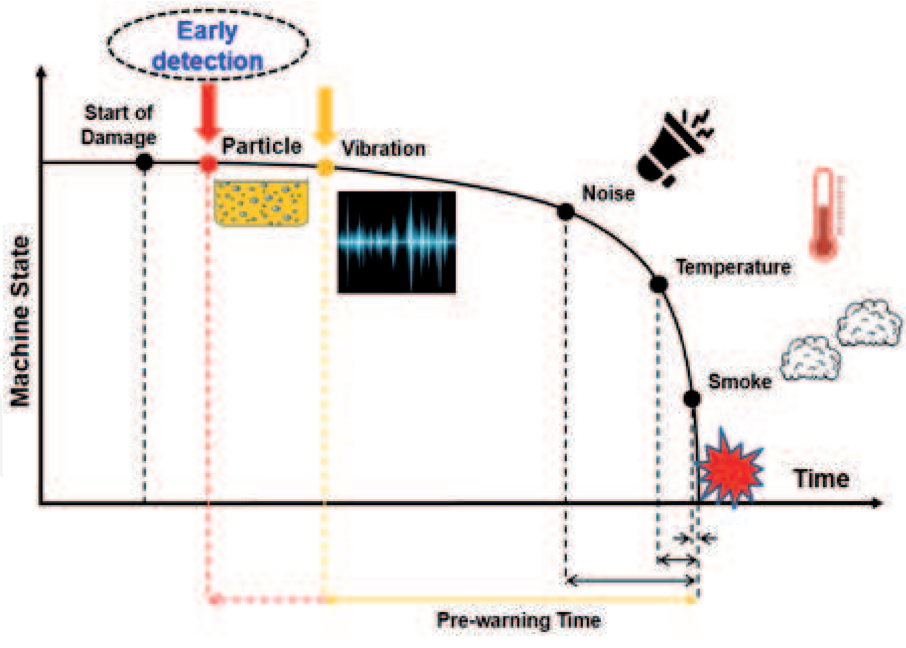

Figure 22.
Advantage of machine condition monitoring with oil analysis.

The three main methods of oil analysis are off-line, in-line, and on-line techniques as shown in **Figure 23**. The method of analyzing lubricants through oil sampling is an off-line and has been mainly utilized in the past. The in-line is the method to analyze directly where the main flow of lubricant oil occurs, and the on-line is the method to analyze the lubricant oil in the by-pass. The in-line can interfere with the flow of lubricant during the measurement process and can be difficult to measure under conditions such as high temperature and high pressure. On-line analysis is the most effective method of the three methods used for analyzing lubricant oils. This is because it can monitor the machine condition effectively using oil sensors in real time without requiring excellent analysis skills and eliminates human errors. Determining the oil quality usually requires complex laboratory equipment for measuring factors such as density, viscosity, base number (BN), acid number (AN), water content, additive and wear debris. Real-time monitoring with oil analysis is also utilized in various industries, such as manufacturing, aerospace, power plants, construction equipment, wind-turbine and marine diesel engines.

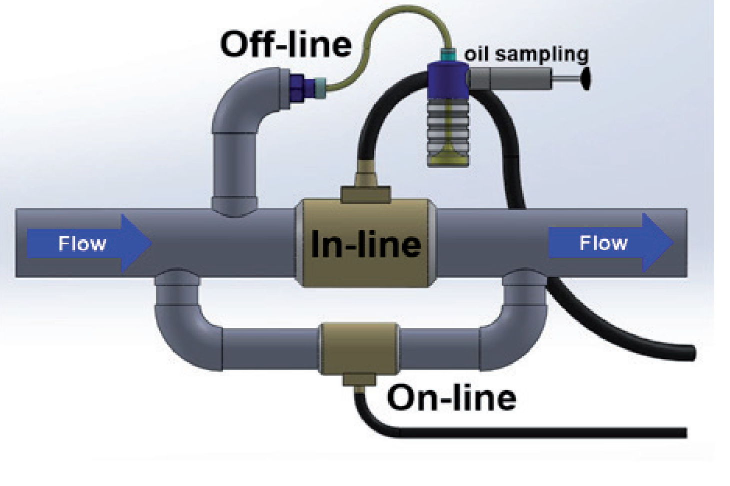

Figure 23.
Methods of oil analysis.

It is well known that faults and failures in marine diesel engines are always caused by wear in the tribo-systems. Vibration source complexity, multiphasic interference, and lower frequencies are the factors that make wear monitoring challenging. This has led to the use of oil analysis as a primary means of monitoring the status of marine diesel engines [27]. Even if the oil analysis has been applied in condition monitoring for marine diesel engines until now, there are some dissatisfactory circumstances in the oil analysis for them. This is because oil analysis takes in off-line mode, and it is not real-time. However, the on-line monitoring system with lubricant sensors is efficient to diagnose condition of marine diesel engines. Among oil properties, wear particle, viscosity, capacitance, base number, dielectric constant, water contents (relative humidity) are commonly measured [27]. The viscosity of the lubricant is its resistance to flow, and the condition of marine engines are normally diagnosed by monitoring the increment or decrement of it. However, if other lubricants with different viscosity or a large amount of fuel are not mixed, a sudden change in the viscosity of the lubricant does not occur easily in a short time.

The moisture is easily observed in lubricant oil contamination and it causes to increase the acid number, multiply microorganism and deteriorate the lubricant quality. The contamination of moisture is caused by water leakage during the operation of mechanical system. Water is a typical polar substance, and the presence of water in the lubricant oil increases the permittivity. The permittivity is a measure of the electric polarizability of a dielectric. The relative permittivity is called the dielectric constant. The dielectric constant of water is almost 80, whereas lubricant oil is about 2. A small increase in water contents of the lubricant oil caused as a sharp increase in permittivity. Therefore, it is possible to measure the amount of moisture in the lubricant oil by the dielectric constant. The Karl Fischer is a representative method for measuring moisture, but it requires a complex experimental device and skilled skill for the analyzer. Therefore, the measurement of water contents with moisture sensor or dielectric constant sensor is effective in the lubricant oil analysis [28, 29]. The oil analysis method with dielectric constant sensor is applied in marine diesel engines [27].

The base number (BN) is a property that is more associated with engine oils than industrial oils. It can be defined as the lubricant’s ability to neutralize acids that are produced during use. The lubricant with proper BN should be used according to sulfur contents of marine fuels to prevent deposit and corrosion in shown in **Figure 24**.

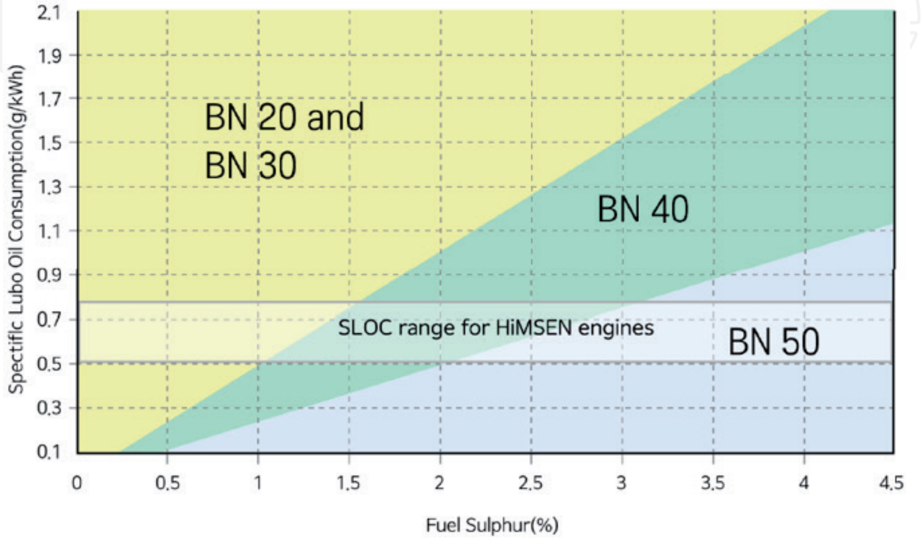

Figure 24.
BN guideline according to fuel sulpher in marine diesel engines.

Figure 25.
$CaCO_3$ deposits on the piston top.

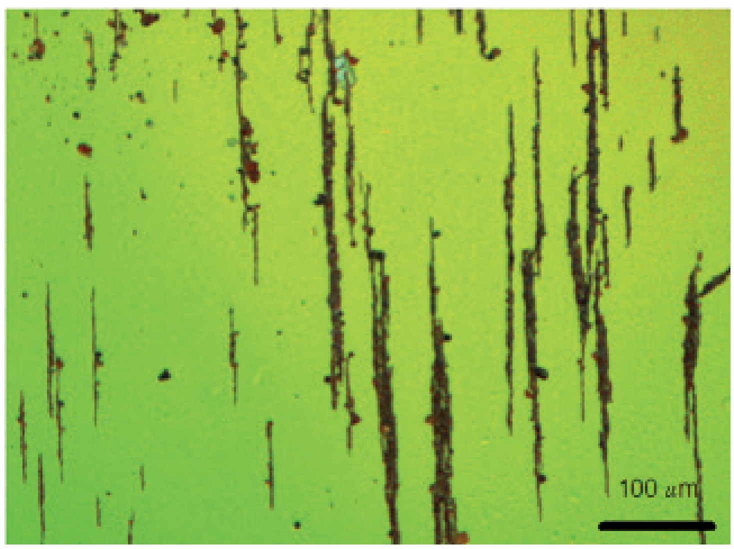

Figure 26.
Ferrogram photomicrograph of marine engine oil [30].

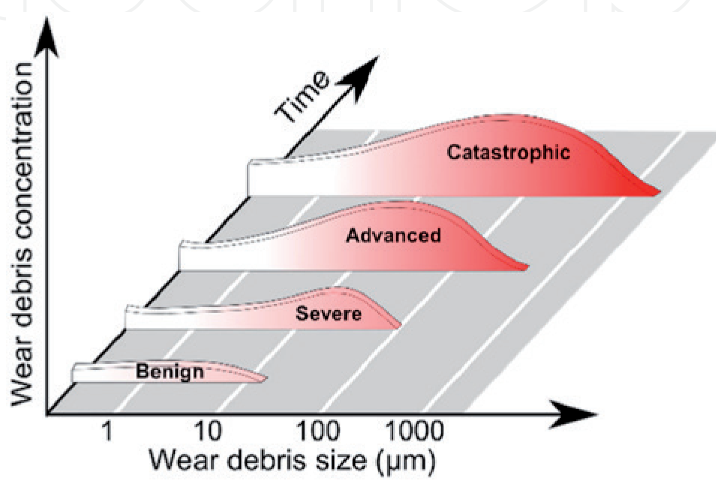

Figure 27.
Relationships of wear debris size, concentration, and machine conditions [31].

When a fuel with a high sulfur content is used and a lubricant oil with a low BN is used, there is a risk of corrosion. Conversely, when a fuel with a low sulfur content is used and a lubricant oil with a high BN is applied, deposit problem occur as shown in **Figure 25**. This is because the lubricant oil with a high BN partially neutralized sulfuric acid, and the remaining additive such as detergent react chemically, resulting in deposits. The tracking the BN of engine oil can determine how much life is remaining. The most reasons for a drop in the BN are related to low quality fuel such as residual fuels and oil oxidation. Therefore, the BN of lubricant in the marine diesel engines must be measured in order to monitor the condition of them.

Condition monitoring of machinery through analysis of wear debris is now an extensively applied as a tool in diagnostic technology. Wear debris analysis or analytical ferrography is a method of predicting the health of equipment in a non-intrusive manner by studying wear particles in lubricant oils. **Figure 26** show the ferrogram photomicrographs of marine engine oil. The shape and length of ferrous wear particles in engine oil evaluated by ferrography as shown in **Figure 26**. The correlation between wear debris, time and wear particles concentration is shown in **Figure 27**. During initial or normal operation of new engines, the wear size is normally between 1 μm and 10 μm. However, in abnormal condition, larger wear particles between 20 μm and 100 μm are detected. Thus, wear particles larger than 20 μm should be monitored in order to provide an early warning of the machine condition [28, 29]. The analysis of wear debris is important to detect critical stages of accelerated wear that precedes costly and dangerous components failures. Therefore, application of wear particle analysis and ferrography by oil sensors is essential means to keep good maintenance in marine diesel engines [32].

3. Conclusions

This chapter explained the tribology of marine diesel engines, which are the heart of a marine system. Modern marine diesel engines must satisfy stringent reliability requirement. Various researches on tribological issues in the marine diesel engines were performed, the lubrication characteristics of machine components such as bearings, cylinder liners, fuel injection pump were improved. Besides, the phenomenon of lacquer is explained in terms of generating mechanism, causes, physical and chemical properties, and prevention or removal methods. Furthermore, condition monitoring with oil analysis is introduced to keep maintenance and to reduce the downtime cost of the marine diesel engines. A variety of tribological researches are needed in the future in order to improve the reliability of the marine diesel engines.

Acknowledgements

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Appendices and nomenclature

A	Dimensionless clearance of stem part
A1	Dimensionless taper length of upper part in the axial direction
B	Dimensionless clearance of head part
B1	Dimensionless taper length of upper part in the radial diction

C	Dimensionless clearance
C1	Dimensionless taper length of lower part in the axial direction
D1	Dimensionless taper length of lower part in the radial direction
H_2	Dimensionless groove depth
h_m	Minimum film thickness
L_1	Dimensionless distance from edge of stem part to the first groove
L_2	Dimensionless width of groove
L_3	Dimensionless distance between grooves
N	Number of circumferential grooves
R_p	Maximum peak height roughness
R_q	Root mean square roughness
R_{q1}	Root mean square roughness of plunger
R_{q2}	Root mean square roughness of barrel
R_v	Maximum valley depth roughness
δ	Absolute slip amplitude in the tangential stress direction
λ	Film parameter
σ	Tensile tangential stress on the contact surface
τ	Frictional shear stress at the interface

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