IMPROVEMENT OF DEUTZ MARINE ENGINES TO REDUCE CYLINDER LINER FAILURE DURING TRANSIENT CONDITIONS

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Thesis submitted in partial fulfilment of the requirements for the
Degree of Master of Engineering in Manufacturing Systems Engineering

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ABSTRACT

The problem of cylinder liner seizure of marine engines fitted on-board Fast Attack Craft (FAC) is an outstanding issue in Sri Lanka Navy(SLN).This failure has resulted in the wastage of money, manpower, reliability and non-operation of craft fitted with these engines. The Original Equipment Manufacturer (OEM) has tried to solve this issue by extensive investigations. However, the problem has not been fully resolved.

The research is mainly focused on geometrical aspects, surface texture, parameters of cylinder liner and low load running as these are the most influential factors for a cylinder liner. The parameters of the engine were investigated experimentally by preparing an experimented cylinder liner and installing it on 04 engines, surface texture and geometry of cylinder liners were inspected by obtaining sample cylinder liners from stock, and Low Load Running was examined on 04 Nos. engines for 06 months duration.

Literature survey revealed that most of the researches are related to thermal cracking, excessive wear down and lubrication oil failure, however, less attention has been paid towards the cooling water system of the engine. Therefore, engines were operated in steady state condition and also in various transient load conditions with experimented cylinder liners to understand the behaviour of the coolant system and related parameters.

It was revealed that the coolant system of the engine is not responding properly to cater to transient load changes of the engine and as a result, the cooling water temperature as well as the cylinder liner wall temperature could not achieve the required temperature to maintain sufficient clearance between piston and cylinder liner, thus causing a risk of cylinder liner seizure. Hence, a more sensitive thermostatic control valve can be introduced, which can respond very quickly in order to maintain specified elevated temperature values of the engine.

The research is structured in 06 main chapters: (1) Introduction, referring to content, aim & objective, methodology (2) Literature Review, addressing the theoretical background of the problem (3) Results, complete analysis of the thesis (4) Conclusion (5) Recommendation and (6) Future Work, referring to limitations and proposals for future research.

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LIST OF ABBREVIATIONS

OEM – Original Equipment Manufacture

SLN – Sri Lanka Navy

1. INTRODUCTION

1.1 Content

The Deutz TBD 620 engine is a German made marine engine fitted on Fast Attack Craft of SLN. These are high speed craft, and can be manoeuvred quickly to counter any enemy attack. At present, the Sri Lanka Navy is in possession of 62 Nos. of these engines fitted on-board Fast Attack Craft and also as standby engines. The cylinder liner seizure of Deutz TBD 620 engine was reported for the first time in 2001 and thereafter, this failure occurred repeatedly. As a result, SLN faced many difficulties and it was a vulnerable situation as craft were getting non-operational at sea, during the peak of the war period. Spare parts cost for one cylinder unit replacement due to seizure is around Sri Lankan Rupees 2.2 Million which is a huge amount. During the warranty period of the engine, OEM provided spares to effect repairs to these break-downs. However after the lapse of the warranty, SLN had to bear the cost. Other than the spare part costs, huge manpower utilization was also involved which in turn incurred huge costs.

As per the OEM, this type of engines operating in other countries is operating satisfactory. Hence, they did not consider it seriously. However, most of these engines on other countries are fitted on-board patrol boats, pleasure craft and as generator prime movers. The nature of usage of this engine in other countries is for totally different purposes, whereas in the SLN it is mainly used for high speed manoeuvring.

The OEM of the engine has carried out a series of investigations to overcome this outstanding issue and after introducing all these upgrades and modifications, the rate of the cylinder liner seizure of Deutz TBD 620 engine has been reduced to a certain extent. However, the problem has not been resolved completely. Therefore, this research was conducted by analysing the operating parameters of the cylinder liner, surface texture, low load running of the engine and geometrical analysis of the cylinder liner and associated components of the engine under SLN operating conditions.

During the analysis, it has been proved that the cooling system of the Deutz TBD 620 engines fitted on Fast Attack Craft of the SLN are not functioning properly during transient load changes of the engine due to the slow response of the thermostats. When the cooling system is not responded to properly, it affected the functioning of the cylinder liner and piston as cooling water temperature has significant influence on clearance between the liner and piston. As transient load changes on-board the FAC occur frequently, there is a risk of cylinder liner seizure. This has been identified as the research gap of this thesis.

1.2 Aim and Objectives.

1.2.1 Aim

The aim of the thesis is to improve the operation of Deutz TBD 620 engines fitted on Fast Attack Craft of the SLN by reducing the cylinder liner seizure during transient load variations of the engine.

1.2.2 Objectives

It is essential to analyse the cylinder liner seizure of the Deutz TBD 620 V 16 engines fitted on Fast Attack Craft of the Sri Lanka Navy, find probable causes for this longstanding failure and also educate the OEM to manufacture defect free and reliable products to the SLN. The main objectives of this study can be summarized as follows:

a. Study about the components and associated system of the engine.
b. Analyse the operating parameters of the engine by using parameter logs and history available on board
c. Experimental analysis of parameters related to the cylinder liner
d. Analyse the clearances and tolerance limits of the moving components of the engine
e. Materials of construction with particular reference to their coefficients of expansion
f. Comparison with the data and parameters of other types of engines available with the navy
g. Based on all the above, to investigate into the existing problem

h. Remedial actions and solutions to achieve a better product for the SLN

1.3 Methodology

The research was conducted by analysing systems, components and nature of failures of the engine by using past records available with the SLN since the acquisition of these engines. The tolerance limits of moving components were investigated by referring to data sheets recorded during the major overhauls of the engines. Assistance was obtained from the OEMs of other reputed engines to find the researches they have conducted for similar nature failures. Further, the literature survey was carried out by referring to research papers, journals, relevant books and articles to find out various factors and system specifications affecting this type of failures. To conduct various practical analyses, some of the engines operating on-board SLN craft were utilized after obtaining concurrence of the SLN.

The parameters of the engine were investigated into experimentally by preparing an experimented cylinder liner and installing it on four(04) engines. The engines were operated under various transient load conditions in order to obtain parameters at real operating conditions. The surface texture was analysed while using 12 Nos. cylinder liners and comparing with other types of marine engines of similar capacity (06 No.s from Deutz and 06 No.s from MTU). The Low Load Running was examined on 04 No.s engines for 06 months duration and the geometry of cylinder liners was analysed on 32 No.s cylinder liners. The recorded data were analysed in depth to find any abnormality of the behaviour of the engine and the research gap.

1.4 Introduction to Chapters

The research consists of 06 main chapters. The first chapter is the introduction which illustrates the background of the problem and highlights the motivation of selecting this topic for the research, followed by the aim and objective and methodology. The second chapter is the Literature Review, which includes the theoretical background of various components and systems of the engine. Then it explains the existing problem of cylinder liner seizure and various research carried out for similar types of failures and remedial actions. The third chapter is the Results, which summarizes all the recorded data of the research, graphic representations, mathematical calculations, complete analysis of the thesis and deriving of the research gap. The fourth chapter is the Conclusion and it highlights the research gap of the thesis. The fifth chapter is the Recommendation, which explains the proposed modifications to be introduced to overcome the research gap and the sixth Chapter is Future Work, referring to limitations of the thesis and proposals for future research.

2. LITERATURE REVIEW

2.1 Introduction to Deutz TBD 620 V16 engine

The Deutz TBD 620 engine is a heavy-duty four-stroke, water-cooled diesel engine designed for various drive applications, both in the marine and industrial fields such as ships, pleasure craft, generators etc. It has been designed with turbochargers and charge air coolers to obtain optimum engine performances. The SLN acquired Deutz TBD 620 engine fitted Fast Attack Craftfor the first time in 1998 from anIsraeli Shipyard, IAI Ramata and thereafter from the Colombo Dockyard Limited in the years 2000 and 2002. These are high speed craft, and can be manoeuvred quickly to counter any enemy attacks.The basic specifications of the engine are as follows:

Make/ Model	:	DEUTZ TBD 620 V 16
Power output	:	1935 Kw at 1800 RPM

T	-	Exhaust Turbocharger
B	-	Charge Air Cooler
D	-	Four-stroke diesel engine
620	-	Engine Series
V	-	V-cylinder Arrangement
16	-	Number of cylinder

Cylinder Number	-16
Total piston displacement	-70.8dm ^{3}
Cylinder Arrangement	- 90^{o}V-angel
Cylinder bore	- 170 mm
Piston stroke	- 195mm
Piston area per cylinder	- 227 cm^{2}
Displacement per cylinder	- 4.43 L
Compression ratio	- 14.2:1
Direction of rotation	- Anti-clockwise
Combustion system	- Direct injection

2.2 Cylinder Liner Seizure of Deutz TBD 620 engines

2.2.1 History and Background of the Cylinder Liner seizure problem.

The first incident of the cylinder liner seizure was reported on 19th April 2001 and A3, A4 units of Starboard main engine P 412 were subjected to heavy seizure during operation of the engine. Thereafter, this failure occurred repeatedly on Deutz TBD 620 V 16 engines. During the warranty period of the engine, OEM provided spares to attend repairs for this breakdown. However after the warranty lapsed; the SLN had to bear the costs. Other than the spare part cost, man power utilization was also involved enormously and thus incurred a lot of manpower costs. The details of the cylinder liner seizure of the Deutz TBD 620 engines can be summarized as follows:

Table 2.1: Statistics of cylinder liner seizure

Sr No	Year	No of failures
1	2001	2
2	2002	3
3	2003	3
4	2004	7
5	2005	2
6	2006	11
7	2007	37
8	2008	32
9	2009	28
10	2010	3
11	2011	1
12	2012	3
13	2013	4
14	2014	5
15	2015	2
16	2016	4
17	2017	5
Total		**152**

Figure 2.1: Cylinder Liner and Piston subjected to heavy seizure

Through this data, it appeared that cylinder liner seizures have taken place mostly during the period of 2006 to 2009. During this period the Fast Attack Craft fitted with these engines have heavily engaged in sea battles against the LTTE terrorist and the engines have been operated with frequent/sudden rpm variations to cater to the scenario. This matter was informed to the OEM on several occasions and it appeared that the OEM did not contribute fully to overcome this problem, but rather introduced a few upgrades on an experimental basis. Due to some modifications, the engine performances have improved, however, the cylinder liner seizure problem was not fully resolved.

2.2.2 Nature of cylinder liner and piston seizure

When examined, the nature of most of the liner seizure, patterns and scrape marks are similar to each other. The seizure marks can be found on both the pressure side and on the counter pressure side of the cylinder liner. Around the skirt of the piston there are several different areas of seizure marks which are all identical in nature. The only difference is that some cylinder units are subjected to heavy seizure while some are at the initial stages. The following photographs indicate some common pattern of seizure of this engine.

Figure 2.2: Nature of seizure

2.2.3 Upgrades done by OEM to overcome Cylinder Liner seizure

As the initial step, the OEM advised to amend the maintenance schedule, as the initial schedule contains removal of cylinder heads after every 2,000 operating hours. The OEM suspected that during this activity, if any minor deviation occurred to the original position of the cylinder liner, then it can cause a cylinder liner seizure. Hence, the OEM needed to minimize the disturbance to the cylinder unit during the maintenance schedule. Also, the OEM advised to introduce a liner locking arrangement during any cylinder head removal of the engine to ensure that the original position of the cylinder liner remained the same. This is an acceptable advice as any minor changes to wear down the pattern of cylinder liner and piston rings may lead to liner seizure.

The OEM has introduced black coated cylinder liners, which has an extra lubricating coating on the liner and the same helps to reduce frictional forces between cylinder liner and piston rings. These cylinder liners have been introduced to both main engines of the P 419 Fast Attack Craft to overcome the cylinder liner failure. Further, maximum RPM of all other Fast Attack Craft fitted with Deutz TBD 620 engines were restricted to 1,700 rpm till the performance of the black coated liners were checked. However, even after introducing the black coated cylinder liners, the A3 and B5 units of the Port main engine of P 419 craft were subjected to cylinder liner seizure after approximately 200 operating hours.

After a series of studies, the OEM has recently introduced a cylinder liner with an APR (Anti Polishing Ring) and its construction is as follows:

Figure 2.3: Cylinder Liner without APR and with APR

The function of the APR is to prevent bore polishing and thus reducing the lube oil consumption. The APR is fitted at the top of the cylinder liner and it can prevent the formation of coke layer, which smooth the running surface of the cylinder liner. Based on the recommendation, the SLN introduced APR Cylinder liners for all the Deutz TBD 620 engines. Simultaneously with this modification, other components of the cylinder unit such as piston, connecting rod and small end bearing of connecting rod were also upgraded by the OEM as follows:

a. **Piston** – Introduced a new piston by re-arranging the lubricating channels inside the piston and reducing the number of holes underneath the piston, which were there to drain on to the top of the connecting rod to lubricate small end bearing and Gudgeon pin. The purpose of reducing the number of holes is to maintain sufficient lubricating oil pressure in the piston. With these modifications, the OEM has tried to improve the cooling of piston to maintain sufficient clearance between piston and cylinder liner and also to prevent overheating of piston as these factors also can support cylinder liner seizures. By introducing additional lubricating holes towards the connecting rod small end side, the OEM has shown concern on the lubrication of the Small End Bearing of the connecting rod, since during some cylinder liner seizures the SLN observed that the Gudgeon Pin was subject to seizure with the small end bearing of the connecting rod.

06 Nos. Lubricating holes underneath piston.Reducednumber of holes to one

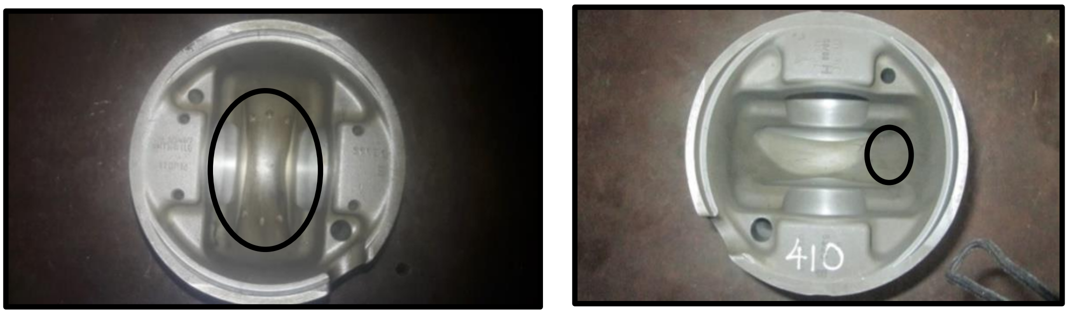


*Figure 2.4:*Initial arrangement and Upgraded piston

b. **Connecting Rod -** Even though the number of holes underneath the piston was reduced, the OEM has tried to improve lubrication of the connecting rod small end bearing and Gudgeon pin by introducing a slot on the connecting rod. By doing this, the OEM has focused on the proper lubrication of the Small End Bearing of the connecting rod. Because, during some cylinder liner seizures, the SLN observed that the Gudgeon Pin suffered seizures with the small end bearing of the connecting rod.

Figure 2.5: Initial arrangement and Upgraded Version

c. Connecting rod small end bearing. – Bearing lubrication was improved as shown in the photographs by enhancing the lubrication groove length around the full circumference of the bearing. Initially, it was only a slot on the top side of the bearing.

Oil groove is only a slot Oil groove around the shell


Figure 2.6: Initial Bearing and Upgraded bearing

When introducing the cylinder liner with the APR, the OEM has slightly increased the inside diameter of the cylinder liner in order to address the cylinder liner seizure problem. The comparison cylinder liner diameter is as follows. The reason behind this modification is to provide sufficient clearance between cylinder liner and piston at elevated temperatures.

Table 2.2: Increasing Cylinder Liner Bore Diameter

Description	**Initial Diameter 170mm+**	**Upgraded Diameter 170mm+**
Diameter of cylinder liner	0.025mm to 0.340 mm	0.043mm to 0.61mm

After introducing all these upgrades and modifications, the rate of the cylinder liner seizure of the Deutz TBD 620 engine has been reduced to some extent. It is understood that the OEM has done all these modifications on an experimental basis and without deep analysis. As per the OEM, these types of engines in other countries are operating satisfactorily. However, most of these engines in other countries are fitted on-board patrol boats, pleasure craft and as generator prime movers. The nature of usage of this engine in other countries is for totally different purposes, whereas in the SLN they are mainly used for high speed manoeuvring. Even after all the modifications done by the OEM, the cylinder liner seizure problem still exists in Fast Attack Craft operating in the SLN and it has become a menace to the Sri Lanka Navy.

2.3 Various systems of the Deutz engine incorporated with the cylinder liner

The descriptions related to the systems of the Deutz TBD 620 engines, which has been elaborated in subsequent paragraphs are based on the experience gathered by studying the systems, and some real photographs have been opted from a dismantled engine, from their operation manual and workshop manuals.

2.3.1 Lubrication Oil System

The primary cooling method of the engine is the lubrication system. The system includes lubrication of all the moving components of the engine. The system comprises the following main components:

a. Sump
b. Lubrication oil gear type pump
c. 03 Nos. Pressure relief valves
d. Centrifugal filter
e. Lubrication oil cooler
f. Lubrication oil filter
g. Main gallery

In this system, lubrication oil is sucked from the sump by a gear type pump and sent to the lubrication oil cooler via a centrifugal filter. The centrifugal filter removes the carbon deposits present in the lubrication oil.

Gear type lubrication oil pump

A gear pump uses the meshing of the gears to pump fluid by displacement. Gear pumps are positive displacement, meaning they pump a constant amount of fluid for each revolution. As the gears rotate they separate on the intake side of the pump, creating a void and suction which is filled by fluid. The fluid is carried by the gears to the discharge side of the pump, where the meshing of the gears displaces the fluid. The tight clearances, along with the speed of rotation, effectively prevent the fluid from leaking backwards.

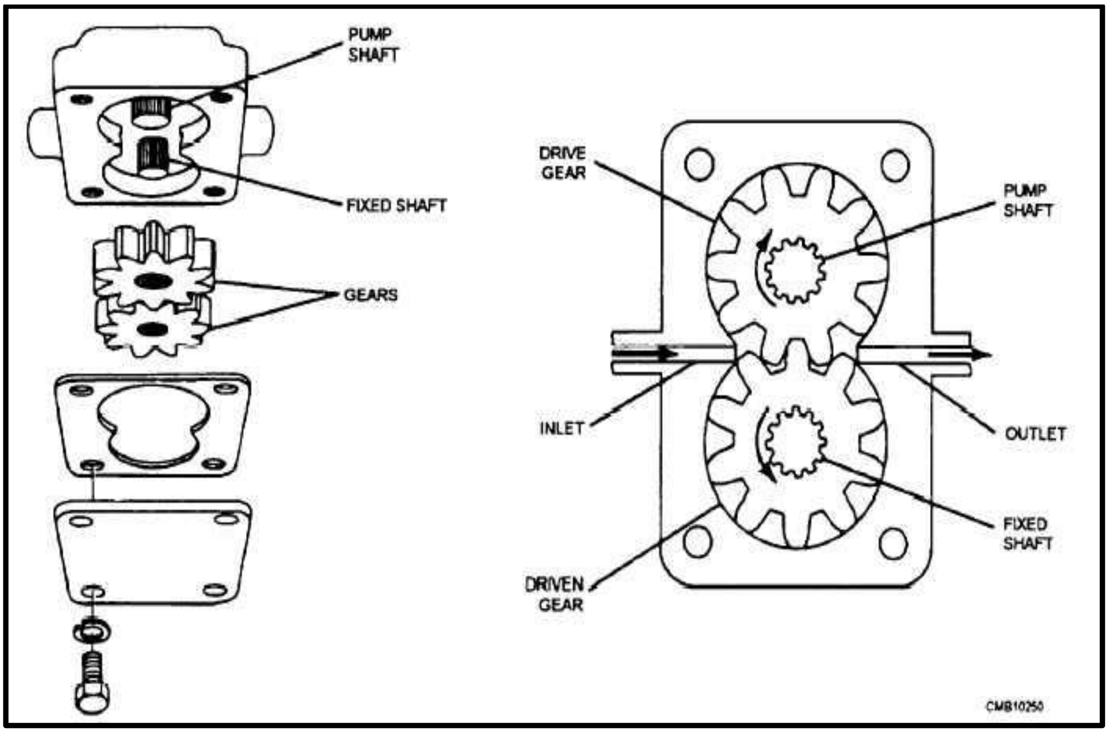


Figure 2.7: Gear wheel type lubrication oil pump [1]

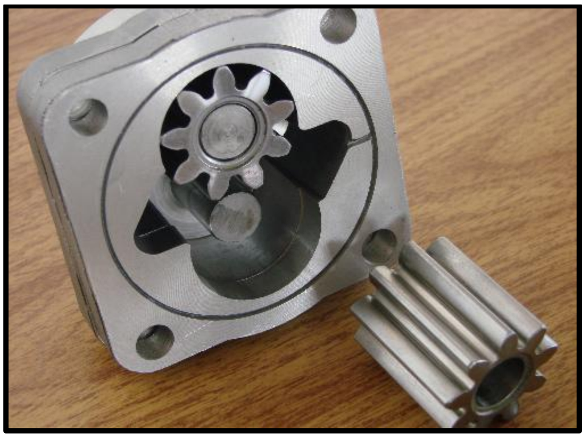


Figure 2.8: Lubricating Oil pump fitted on Deutz TBD 620 engine

Lubrication oil cooler

The Lubrication oil cooler is a shell and tube type heat exchanger, in which lubrication oil exchanges heat with the coolant.

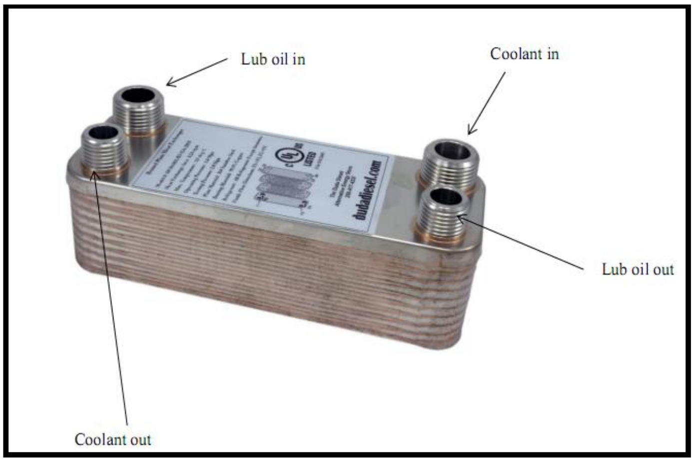


Figure 2.9: Lubricating Oil Cooler

Pressure relief valves are used for releasing the additional pressure generated inside the main lubrication oil gallery to the oil sump, when it increases beyond 10 bar. After the oil cooler, lubrication oil flows to main lubrication oil gallery through lubrication oil filters. From the main lubrication oil gallery the oil flows to the following locations:

a. Piston cooling nozzles
b. Liners
c. Cam shaft
d. Main bearings
e. Turbochargers
f. Sea water pump
g. Coolant pump

On completion of lubricating the above components, the lubrication oil returns to the sump. Important specifications of lubrication oil used in the engine are as follows:

Temperature

a. Engine inlet lube oil temperature, at full load, after cooler < 90 °C
b. Engine inlet lube oil temperature, at full load, before oil cooler approx.100 °C

Pressure

a.	After filter with warm engine (Engine speed 1500 rpm)	4.4 – 4.8 bar
b.	After filter with warm engine (Engine speed 1800 rpm)	4.6 – 5.0 bar
c.	Differential pressure before / after filter	0.2 bar

Filter characteristics

a.	Paper inserts	35 µm
b.	Strainer jacket	100 µm

Lube oil pump characteristics

a. Type of pump Gear type

b. Delivery rates of lube oil pump at maximum rpm 1800 – 41.0 m^3/h

Lubrication System Diagram

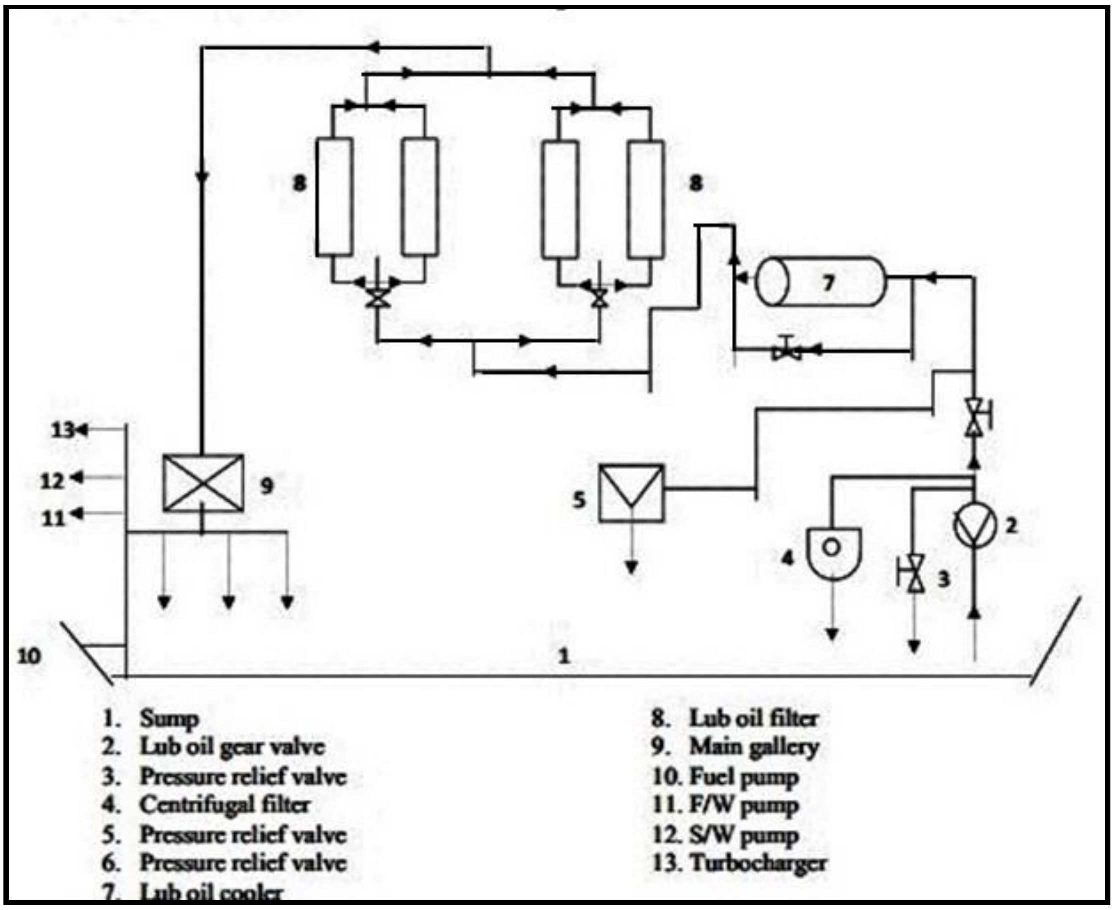


Figure 2.10: Lubricating oil system diagram [1]

Lubrication of Pistons, Cylinder Liner and Small End of the Connecting Rod

After the main gallery, the lubricating oil flows into the pistons and also splash around the inner surface of the cylinder liner through the cooling nozzles. Excess lubricating oil on the inner surface of the cylinder liner is scraped down and falls to the sump by using the oil scraper ring of the piston.

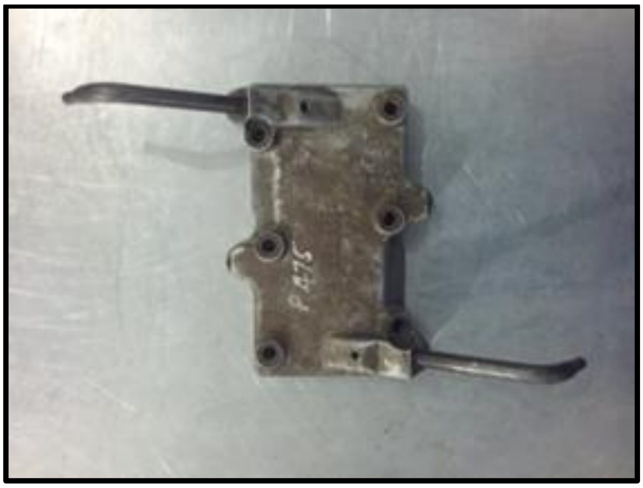


Figure 2.11: Piston cooling nozzles

Piston cooling and connecting rod small end lubrication

There are 02 holes in the piston (Lubricating oil inlet hole and outlet hole). Lubricating oil continuously flows through the cooling nozzles. When the piston reaches the BDC position, lubrication oil enters the piston through the oil inlet hole, circulates inside the piston and is discharged from the lubrication oil outlet hole on the piston. At the same time, once the lubrication oil passes through the piston, it flows through the hole under side of the piston and drops on the small end of the connecting rod.

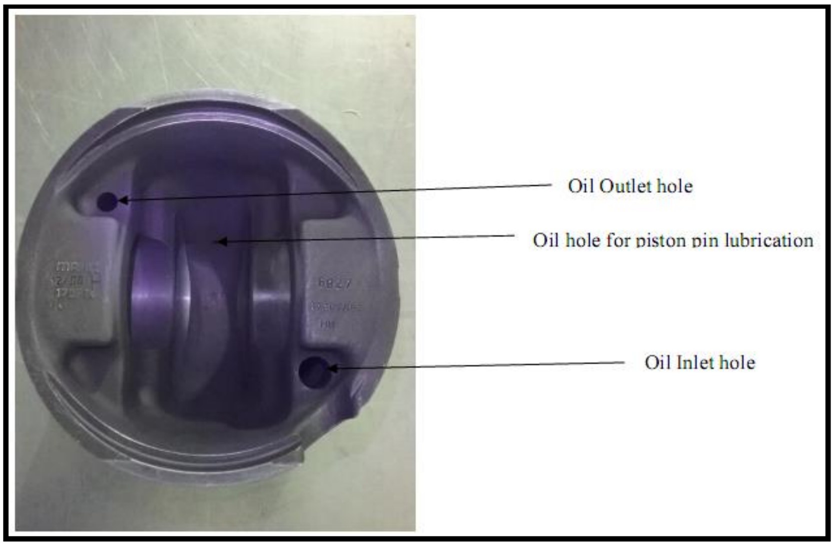


Figure 2.12: Lubrication oil holes in the Piston

Figure 2.13: Connecting rod lubrication

Cylinder liner lubrication

Lubricating oil continuously flows through the cooling nozzles and splashes on the cylinder liner surfaces. There are three piston rings on the piston. The oil scraper ring wipes off the splashed lubricating oil so as to lubricate the liner for the next stroke. The cylinder honing helps to form an appropriate thick lubrication oil layer for cylinder lubrication.

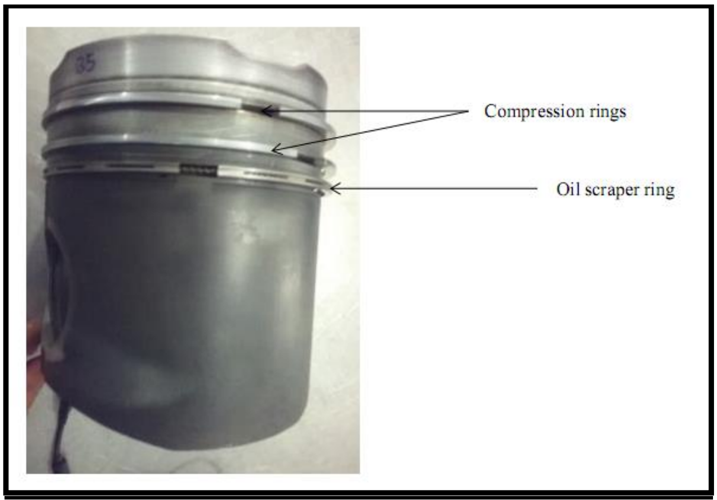


Figure 2.14**:** Piston with Piston Rings

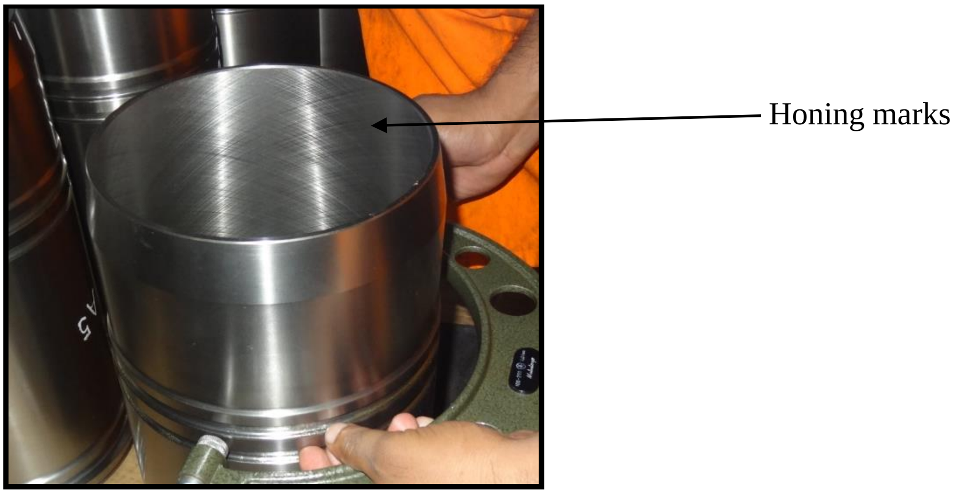


Figure 2.15**:** Cylinder liner with honing

2.3.2 Fuel System

The fuel system is one of the paramount and most important systems of the engine. The followings are the main components of the fuel system:

a. Fuel tank
b. Non return valve
c. Fuel filter
d. Fuel Manifold
e. Fuel pump
f. Lift pump
g. Separator
h. Injection pump
j. Injectors
k. Flow monitor
l. Overflow line
m. Fuel distribution block

An engine-driven pump helps to circulate fuel in the entire fuel system. Fuel is transferred to the fuel pump suction side from the fuel tank through a non-return valve and fuel-water separators. The fuel pump discharges the fuel to the fuel filters and on completion of filtration, thereafter delivers to the Fuel Injection Pump and then to the injectors. There is a separate lift pump for priming the fuel system manually.

The fuel distribution block plays a major role in the fuel system that distributes fuel appropriately for the required components. There are 02 Nos. non-return valves in the fuel distribution block, which helps to maintain the required pressure of the fuel system.

Fuel System Diagram

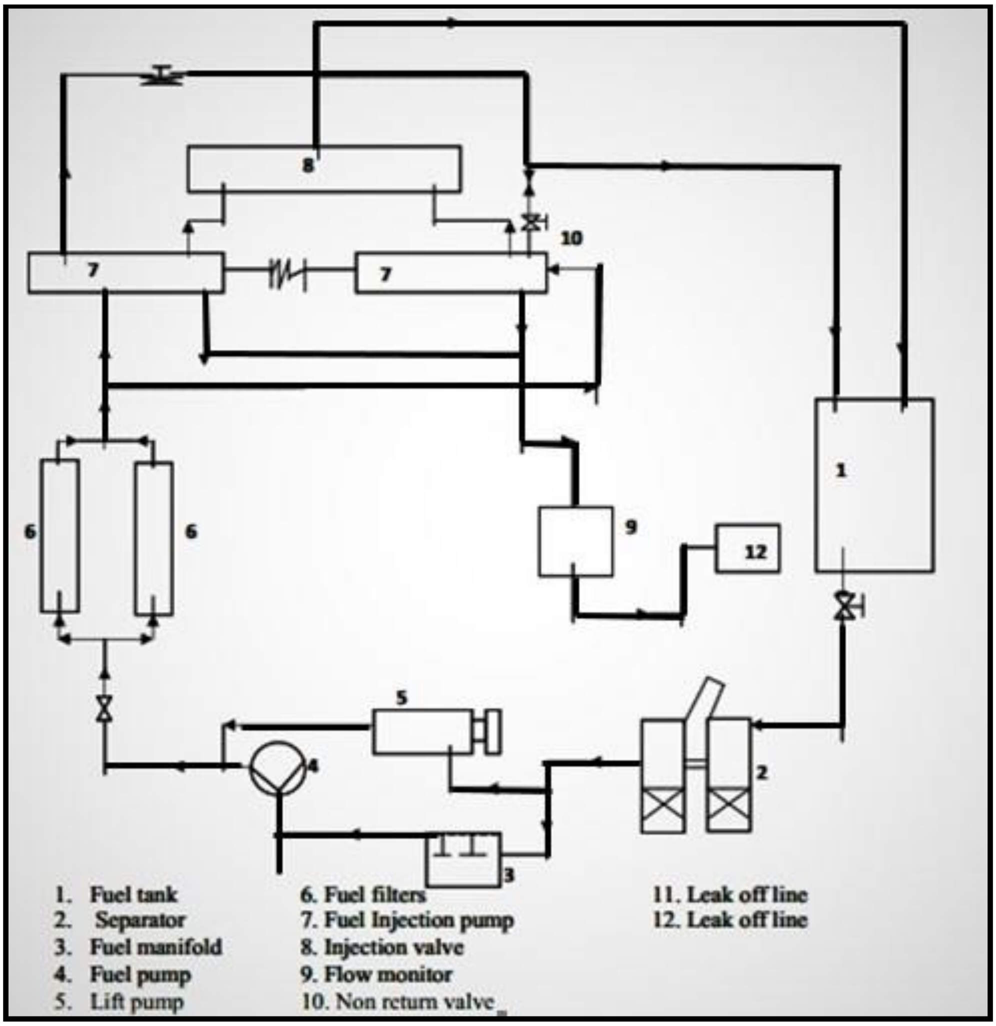


Figure 2.16: Fuel system diagram [1]

The Fuel Injection Pump delivers fuel to the injectors as per the demanded quantity at a demanded time. The Elements decide the amount of fuel being delivered to the injector as per its position of rotation.

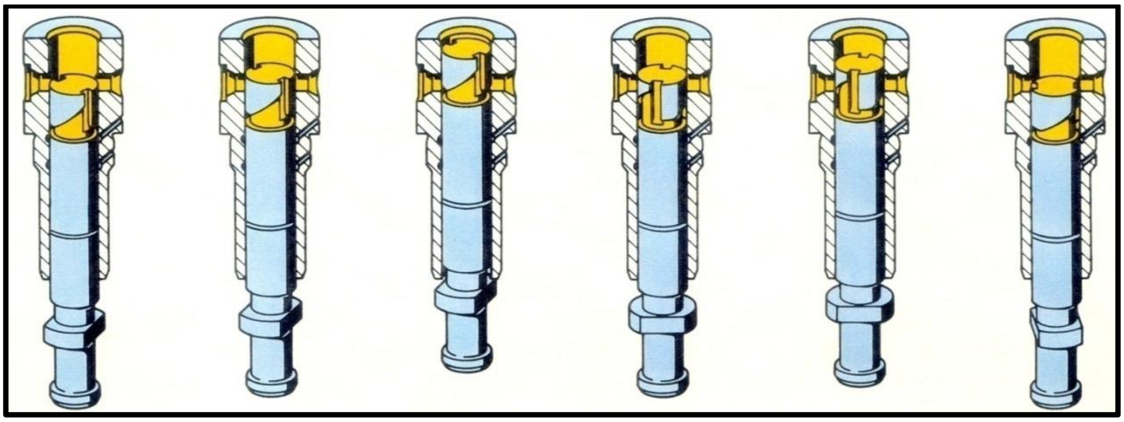


Figure 2.17: Injection Pump Nozzle Elements [1]

2.3.3 Coolant System

The Coolant system is also a paramount and important system since it is the main cooling method of the engine. The followings are the major components of the coolant system of the main engine:

a. Expansion tank
b. Coolant Pump
c. Cooling water
d. Collecting pipe
e. Thermostat
f. Plate cooler

In this system, the coolant is circulated by a pump. First the coolant flows to the lubrication oil cooler and absorbs the heat of the lubricating oil. After that the coolant flows to the cylinder block and then on to the cool liners and cylinder heads. Thereafter, the coolant flows to the thermostat through the turbocharger and bearing housing. If the temperature of the coolant is less than 75^0C, the coolant is sent back to the suction side of the coolant pump by closing the thermostat. If the temperature of the coolant is more than 75^0C, then the thermostat is opened and the coolant is sent through the plate cooler and therein it is cooled by the help of sea water.

Thermostat

The thermostat used in the DEUTZ TBD 620 V16 main engine is a “Wax pellet type” thermostat. When the thermostat is in cold condition the wax is in a solid form. Once the thermostat starts gaining heat the wax melts and expands.

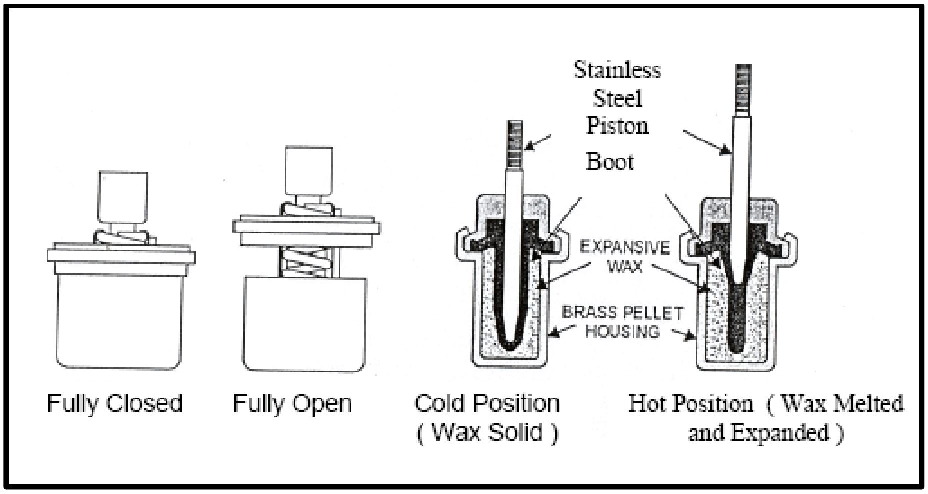


Figure 2.18: Operation of thermostat

Figure 2.19: Thermostat of DEUTZ TBD 620

Plate cooler

The Coolant of the DEUTZ TBD 620 V16 main engine is cooled by sea water via a 'plate type' heat exchanger. The theory behind this heat exchanger is interchanging heat through metal plates. In this case, the surface area of the fluids in contact is increased and therefore, the heat transfer rate is increased. Individual plates consist of rubber gaskets in order to prevent leakages of fluids.

The configuration of the 'plate type' heat exchanger is as follows:

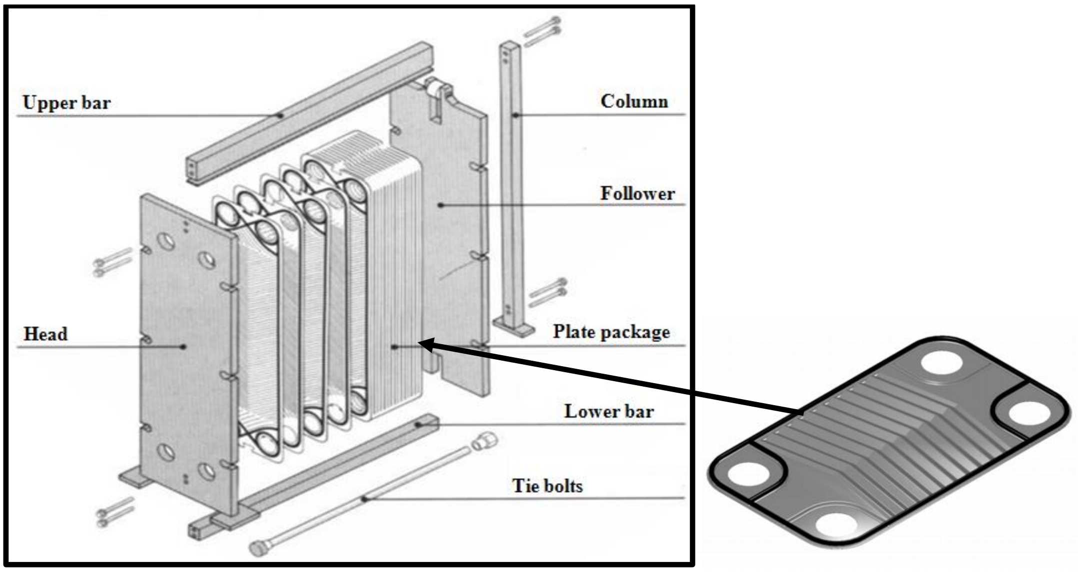


Figure 2.20: Configuration of 'plate type' heat exchanger

Coolant pump

The Coolant pump is a centrifugal type pump which circulates the coolant within the entire Engine.

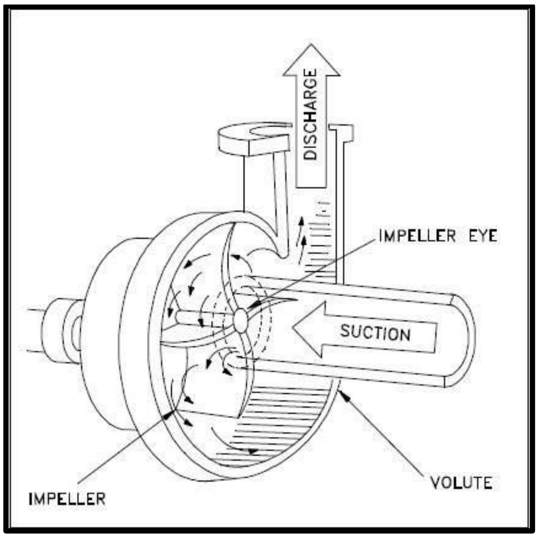


Figure 2.21: Centrifugal type Coolant Pump

Coolant System Diagram

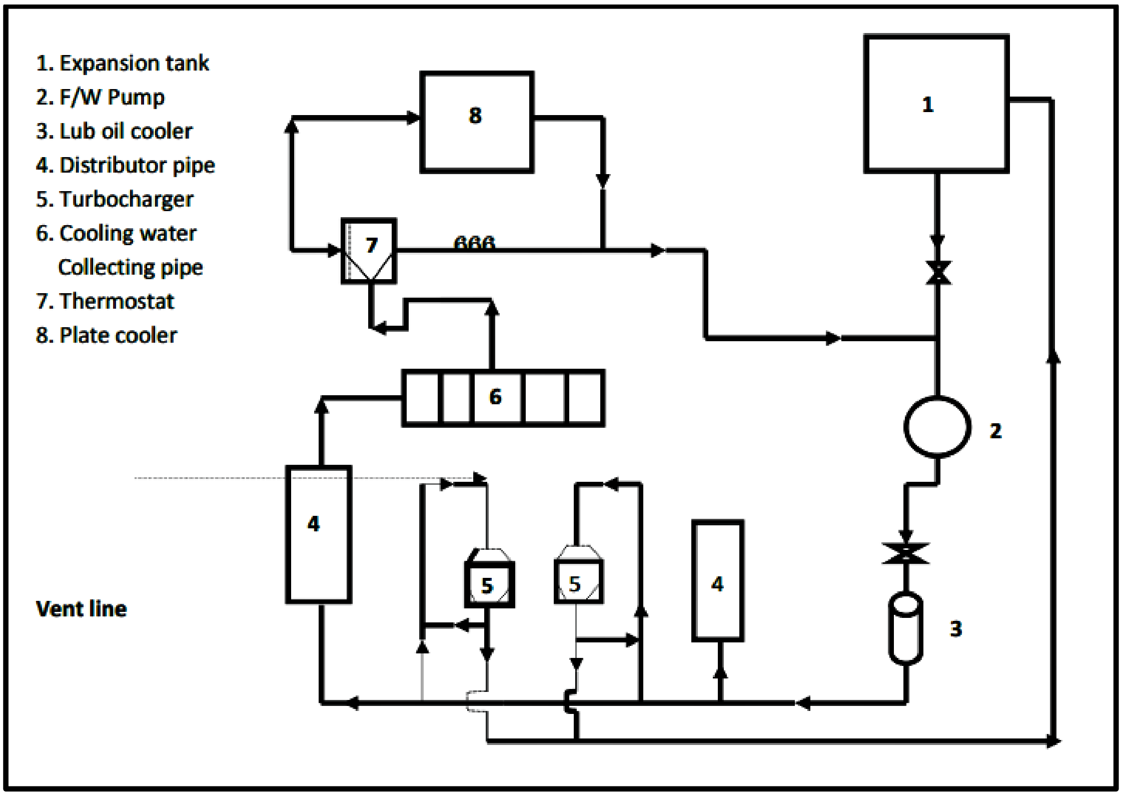


Figure 2.22: Coolant system diagram [1]

2.4 Various Components Related to Cylinder Unit

2.4.1 Cylinder liner

The Cylinder liner is an enclosed space where combustion takes place and it is made according to the centrifugal casting process. There are wet liners and dry liners, and wet liners are common in marine diesel engines, in which the coolant comes directly in contact with the cylinder liner's outer surface. Sealing of the cooling water jacket is affected at the top and bottom by o-seals, and in Deutz engines the cylinder liner collar itself is designed to seal the top position. In these cylinder liners, the design of the collar with regard to fatigue limit and heat transfer is free from compromises due to the sealing elements. The cylinder liner is subjected to heavy stresses during the reciprocating movements of the piston and combustion. Therefore, the construction of the cylinder liner is with all symmetrical support in the crankcase and its purpose is to form a stable unit. This is a basic requirement for low wear rates and low lubricating oil consumption. There are practical reasons for the manufacture of the cylinder liner separately from the crank case.

a. The liner can be manufactured by using a superior material to the cylinder block. The cylinder block is made of a Grey Cast Iron, the liner is manufactured with a cast iron alloy containing chromium, vanadium and molybdenum. (Cast iron contains graphite, which is also a lubricant. The alloy elements are used to help resist corrosion and improve the wear resistance at high temperatures.)

b. The cylinder liner is subjected to wear because of its continuous usage and therefore it may have to be replaced within a specific time period. The cylinder block lasts with the life of the engine.

c. The liner ishotter than the jacket at working temperature, and it is free to expand more diametrically and lengthwise. If they were cast as one piece, then it would cause unacceptable thermal stresses, causing the fracture of the material.

d. Less risk of defects. It is difficult to construct a homogenous structure with less residual stresses [2].

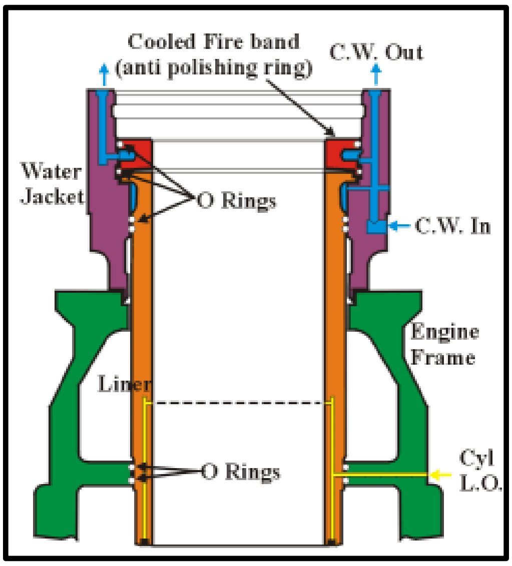


Figure 2.23: Cylinder Liner of Four stroke engine

(**http://www.marinediesels.info/4_stroke_engine_parts/The_4_stroke_cylinder_liner.htm**)

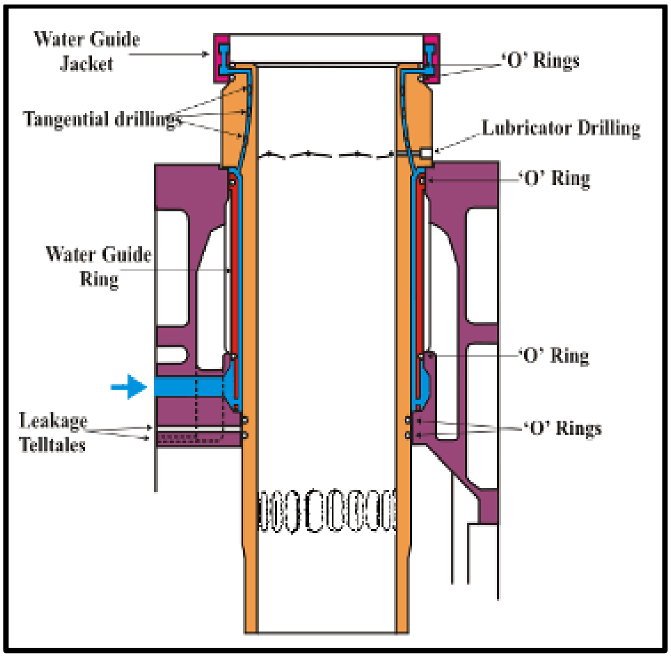


Figure 2.24: Cylinder Liner of two stroke engine

(http://www.marinediesels.info/2_stroke_engine_parts/liner.htm)

2.4.1.1 Cylinder Liner Materials

The cylinders can be manufactured by using cast iron which contain Phosphorus, Manganese, Chromium, Molybdenum, Vanadium and Titanium as alloying elements or Steel or Aluminium. Cast Iron is the most commonly used material when manufacturing the cylinder liners of larger engines. Material specifications for cast-iron cylinder liners are presented below. Nodular Cast-Iron cylinders with Cermet (ceramic-metal composite) sliding surfaces have been used in some low-speed two-stroke diesel engines. A chromium layer can be coated to enhance the wear resistance of the cylinder liner[3].

Table 2.3: Material Composition of Cast Iron cylinder liner [3]

	Composition [%]							
	C	Si	Mn	P	Cr	S	Mo	Ni
Standard 45	2.8–3.2	1.7–2.4	0.5–0.8	0.4–0.45	0.25–0.4	< 0.03	-	-
Standard P	2.8–3.2	1.7–2.4	0.5–0.8	0.6–0.8	0.25–0.4	< 0.3	-	-
HE G40	2.6–2.8	1.1–1.6	< 0.8	< 0.08	-	< 0.08	1.0–1.5	1.0–1.5
ASTM 247	3.1–3.4	1.85–2.3	-	< 0.12	< 0.35	< 0.18	0.25	0.50

2.4.1.2 Surface Texture of Cylinder Liner

The purpose of the cylinder bore surface finishing is to achieve some specific factors such as reducing oil consumption, increasing the durability and increasing the resistance against wear and scuffing. The surface roughness of the cylinder liner should have a Ra value between 0.25 and 0.4 µmandRz value between 3 and 6 µmto limit hydrocarbon emissions and particles by reducing the oil consumption[3]. This can always be achieved with optimally honed bore surfaces. There is a consensus for a surface texture of clean-cut surfaces with smooth, flat plateau and a regular, consistent arrangement of primary oil-retaining valleys[4]. The scuffing resistance, which depends on the surface finishing of the cylinder liners and scuffing is due to the high friction, which can lead to polishing of the liner [5]. The honing process is applied for finishing the surface of the cast-iron cylinder liners. The cutting marks of the honing form a pattern of diagonal valleys on the liner surface. The honing groves control the amount of oil, the volume and the direction of the valleys control, by keeping the oil on the liner surface and by improving the spreading of the oil. Since the requirements of good sealing properties and optimal lubrication are contrary to each other, the demands on the topography of the cylinder liner are exacting [6][4].

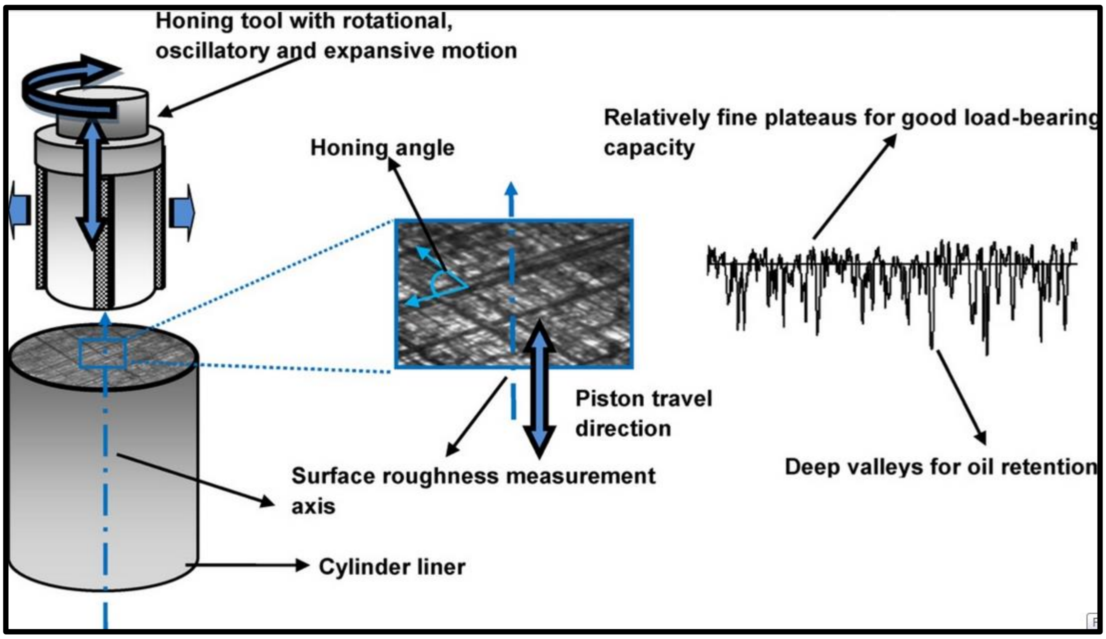


Figure 2.25: Honing details of cylinder liner [7]

The quality of the honed surface is affected by the bore geometry and diameter tolerances, the surface roughness of the fine bored surface, the machining operations and the number of machining stages, the material, hardness and type of the honing stones. The liner is usually machined to obtain the specific surface in the following four steps:

a. Step 1- Rough boring– for the basic geometry.
b. Step 2- Rough honing – for alignment.
c. Step 3- Fine honing – for desired surface roughness.
d. Step 4- Plateau-honing – for surface smoothing.

Step 3 removes all the traces of the first two steps. Step 4, the plateau-honing, partly replaces the running in process of the liner surface, which improves the dimensional tolerance of the cylinder, increases the engine efficiency and decreases oil consumption[6]. Many parameters have been used to characterize the plateau-honed surface.

2D-parameters, such as

1. Ra (mean deviation of the surface roughness)
2. Rz (mean surface roughness)
3. Rmax.

Parameters describing the shape of the surface, such as

4. Skewness (Rsk)
5. Kurtosis(Rku).

Functionally characterized 2-dimensional Rk and 3-dimensional Sk parameters

6. Rpk and Spk (Reduced peak height, addresses the running-in properties)
7. Rk and Sk(Core roughness depth, addresses the wear and load-carrying capacity)
8. Rvk and Svk (Reduced valley depth, addresses the oil volume for lubrication).

The depth of the surface deformation created by different machining operations is as follows:

Table 2.4: Depth of surface deformation [4]

Machining operation	Depth of deformation
Coarse machining	70–80 μm
Fine machining – Ceramic	50–60 μm
Base hone – Diamond	25–35 μm
Base hone – CBN	10–15 μm
Base hone – SiC	10–15 μm
Plateau hone –Diamond base	15–20 μm
Plateau hone –CBN base	5–10 μm
Plateau hone –SiC base	5–10 μm

2.4.2 Piston Rings

Piston rings are metallic seals which function as sealing medium in the combustion chamber from the crankcase and assure the flow of heat from the piston to the cylinder. Further, the piston rings help to achieve a uniform oil film thickness on the cylinder liner and also avoid passing the lubricant to the combustion chamber. To achieve this, the piston rings must always be kept in contact with the cylinder wall and the piston groove side. The inherent spring force of the ring will help achieve radial contact [8].

The piston rings are associated with the piston to reduce the slapping motion of the piston. This is mainly during cold starts of the engine and in this condition clearances are greater than during running conditions. In most engines the piston has compression rings and oil control rings. The top rings are compression rings (keystone rings) and the rings below are oil control rings. All rings are opened at one location, called the ring gap, which is an indication of wear down and also help to assemble the piston rings onto the piston.

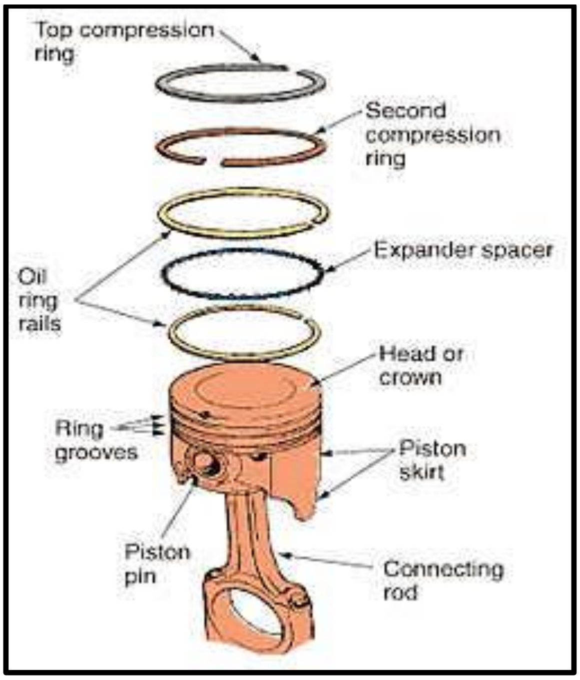


Figure 2.26: Piston rings[8]

2.4.2.1 Types of piston rings

Piston rings come as a ring pack and generally consist of two to five piston rings. The number of rings varies upon the engine type, normally with two to four compression rings and zero to three oil control rings. High speed four-stroke diesel engines are generally equipped with two or three compression rings and a single oil control ring. Apart from that, there are scraper rings for the purpose of sealing and scraping off oil from the cylinder liner wall. These Scraper rings have a beak intended to scrape the oil.

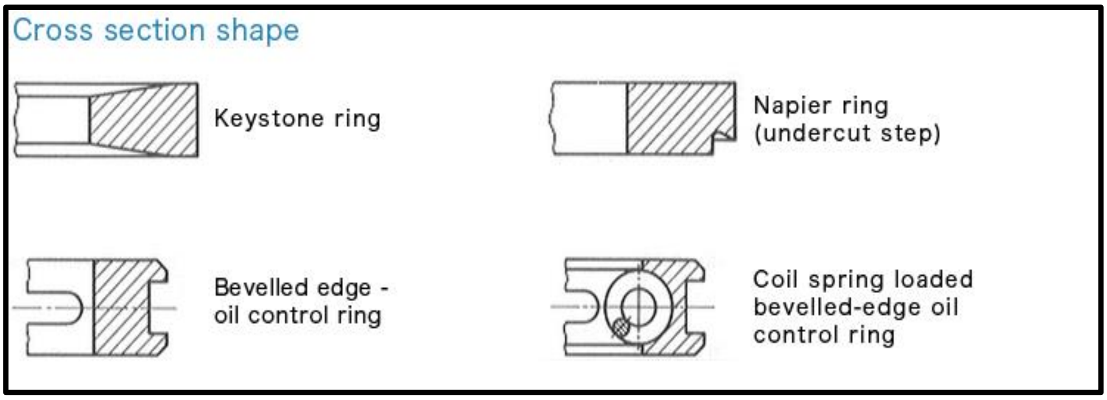


Figure 2.27: Piston ring types [9]

2.4.2.2 Piston ring materials and coatings

A piston ring material is selected to suit to the demands of the running condition of the engine. Further, elasticity, good thermal conductivity, resistance to damage and corrosion are essential requirements for the ring material. Most of the piston rings are made of Grey cast iron, because of the lubrication effect of the graphite phase.

Chromium coating of piston rings are widely used, mainly in corrosive and abrasive conditions. Hard chrome plating is particularly relevant for the compression ring. Other than the Chromium plating, piston ring surfaces are plasma sprayed with metal or ceramic composites like molybdenum's a uniform coating [10]. Experiments have been done on thermal spray of new powder compositions which include molybdenum/nickel/chromium alloys, chromium oxide (Cr_2O_3) with metallic chromium binder, alumina and titania (Al_2O_3-TiO_2), tungsten carbide (WC) with metallic cobalt binder, ($MoSi_2$andCrC-NiCr) [11][12]. Hard chromium layers can be improved by plasma spraying chromium ceramic on the ring face, thereby increasing the thermal load capacity [13]. Furthermore, thin, hard coating compositions like Titanium nitride (TiN), Chromium nitride (CrN) are used, but this type of coating is currently used exclusively for small series production and for selected production engines[14][15]. Multilayer Ti/TiN coatings have been experimentally deposited onto castiron piston rings, and this coating is claimed to be more wear resistant than the chromium plated or phosphate surface, especially when the number of layers is high[16]. Haselkorn and Kelley have done experiments to find coatings for use in low heat rejection engines and revealed that high carbon Iron/Molybdenum blend and Chrome/Silica composite applied by plasma spray, and also chrome nitride applied by low temperature arc vapour are coatings which can meet the demands in low-heat rejection engines[17].

2.4.2.3 Blow-by prevention

Blow-by is common in worn out engines, low load operations and due to defective piston rings, where combustion gases goes from the combustion chamber through the ring pack to the crankcase. The combustion gases flow past the piston ring at various locations:

a. at the piston ring gap
b. past the front side of the piston ring at starved lubrication conditions
c. Past the backside of the piston ring when the ring is not in contact with either of the ring-groove walls

The hot blow-by combustion gases cause the piston and piston rings to overheat. The blow by disturbs the piston and ring lubrication by affecting the oil film and combustion gases contaminate the lubricant and cause the oil to entrain in them. When the combustion gas reaches the crankcase it pollutes the lubrication oil. Blow-by cannot be totally prevented as long as the rings have gaps and move in their grooves. This means that some blow-by will always have to be allowed. The blow-by directly or indirectly affects the engine power (fuel) efficiency: the blow-by consumes some of the combustion power and increases the friction as a result of less favorable lubrication conditions. The gap between the piston and liner wall is greater on the anti-thrust side of the piston than on the thrust side. This requires that the gap between the back-side of the ring and the ring groove is quite large and thus has a large gas-flow area. Measurements have shown that the twist of the piston rings affects the amount of blow-by past the ring pack. A negative twist on the second ring can cause instability of the ring, which results in an increase in the blow-by. A positive twist on the second ring can, in turn, cause high land pressure, which may result in radial collapse or axial movement of the ring [18].

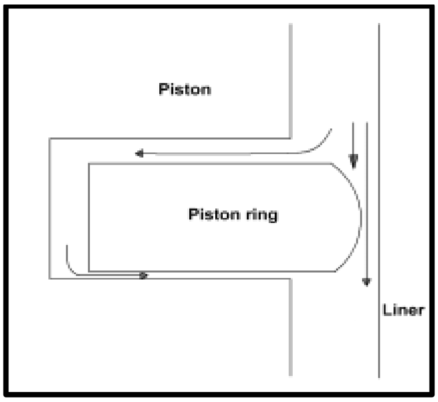


Figure 2.28: Combustion gas blow-by past the piston ring

2.4.3 Piston

There are many piston types developed according to the operating requirements of different engine types. The piston types are commonly categorized by their cooling arrangement, by their primary field of application or by their structure. Well-defined descriptions of different piston types can be found in the Refs.[19][10]. Four classifications of piston types, according to their structure, are presented as follows:

a. Un-cooled or oil spray-cooled, cast or forged mono-metal light-alloy pistons for high-speed automotive and small utility vehicle engines
b. Un-cooled or oil spray-cooled cast light-alloy pistons with ring-groove insert for high speed and heavy duty diesel engines
c. Single-piece or composite pistons with a cooling gallery for high speed with heavy duty and medium speed diesel engines
d. Pistons for two stroke with low speed diesel engines.

The most essential areas of the piston are the piston top, the ring belt including the top land, the pin support, and the skirt. The geometry of these areas can vary significantly incompliance with the field of application.

In a large number of piston designs, the piston ring belt consists of three ring grooves. The piston rings are situated in the grooves between the ring-groove flanges. Since the ring groove and the flanges are part of the piston sealing system, affecting the blow-by of the combustion gases and the oil consumption; the surfaces of the flanges have to be of very high quality. To achieve low mass forces, high resistance against deformation and fatigue failure, and good sliding properties, the piston materials need to have the following requirements.

a. low density
b. high strength for temperature variations
c. good heat conductivity
d. high wear resistance
e. suitable heat expansion

Most of the piston are made of light alloys of cast iron, nodular cast iron, and alloyed steels. The pistons for high speed engines are generally made of aluminium silicon alloys. In addition to aluminium, it also consists of 11% to 13% silicon and each of copper, nickel and magnesium in 1%. Heat expansion and wear can be reduced by enhancing the silicon content to approximately 18% to 24%, however, the strength falls. In large, low-speed two-stroke engines, pistons of high-grade iron based materials and primarily nodular cast iron with a paralytic base structure, are still employed today [19].

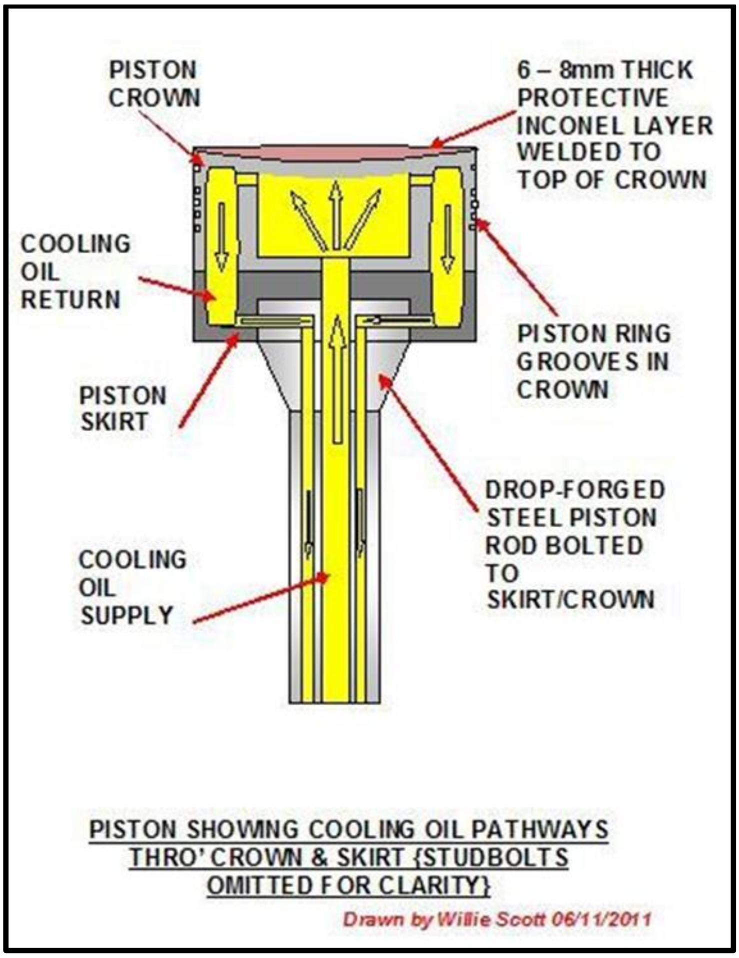


Figure 2.29: Piston
(http://www.brighthubengineering.com/marine-engines-machinery/58129-piston-of-large-marine-engine/)

2.5 Theories behind the problem

2.5.1 Cylinder lubrication

The piston moves to and fro within a cylinder and there are several piston rings inserted in the grooves on the piston which perform multiple tasks including sealing of pressure inside the combustion chamber, lest it leaks from below. The speed of rubbing between these piston rings and the cylinder liner is quite high and apart from that there are extreme conditions of temperature, pressure and corrosive gases inside the combustion chamber to which any lubricating fluid between the liner and the piston would be subject to. The essential properties of cylinder lubricants are as follows:

a. It should be able to minimize friction between the piston rings and the liner and frictional wear.
b. Should maintain sufficient viscosity for temperature variations and good oil film.
c. It should maintain good seal with the assistance of piston rings and thereby avoid blow by, burning oil film and inadequate of compression.
d. Should possess a clean burn without ash deposits.
e. It should able to neutralize the corrosion effects of mineral acids which are developed during fuel combustion.

The type of lubrication found inside the cylinder is of the thin film or boundary type of lubrication for most parts of the piston motion except for the upper and lower extremes of motion where this changes to imperfect lubrication as the speed reduces to zero at these points and the direction of motion is reversed.

When the moving parts are at relatively low speed boundary lubrication can occur and as a result, metal to metal contact can occur by reducing the oil film thickness. This phenomenon can happened when starting to rotate, at low speed, high load and due to inadequate viscosity.

2.5.2 Friction in cylinder liner assembly

Generally about 20% of the mechanical losses in an Internal Combustion engine are due to the frictional losses. Hence, when reducing the frictional losses, the engine can achieve high efficiency, less emission and fuel consumption. Experiments done by Takiguchi and co-workers with two rings and three ring pistons have indicated that the number of rings can also influence the frictional behavior of the piston ringpack, however, ultimately frictional losses are determined by the total tension of the piston rings in the piston ring pack[20]. Hence, during the design stage various modifications and experimental approaches are done to reduce the frictional losses. In such an experiment, Bedajangam S. K and and co-workers have tried to develop a piston ring to enhance the engine efficiency without adverselyaffecting the cost, blow-by, wear and oil consumption[21]. In this study Bedajangam S. Kand co-workers have shown that a significant contribution of the total power lossin an IC engine is due to the frictional interaction between the upper compression ring and cylinder wall as shown in the figure below.

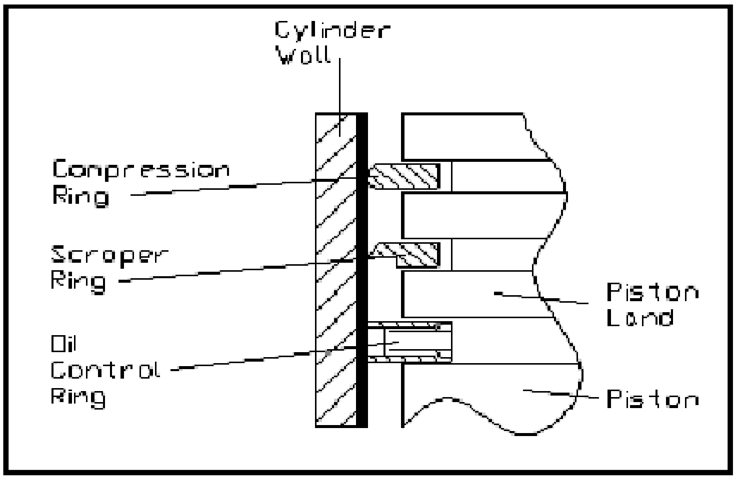


Figure 2.30: Assembly of lubricating Piston Ring and Cylinder Wall [21]

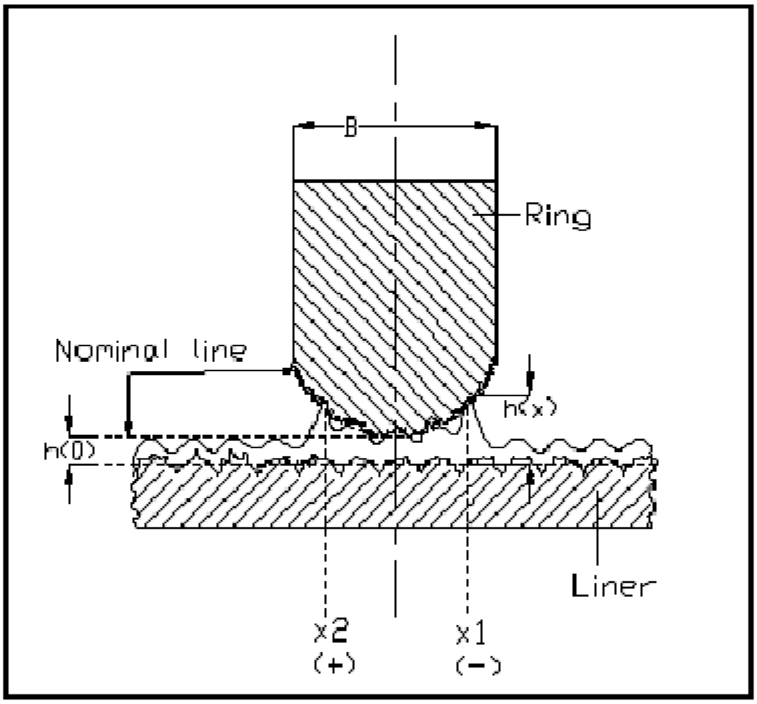


Figure 2.31: Lubrication Conditions Encountered by Piston rings[21]

Due to the roughness of the surfaces in contact between the compression ring and cylinder wall, it is possible for certain portions of the two surfaces to have asperity contact and for other parts to be sufficiently lubricated so that the ring is supported by the load from the oil film. To simplify this situation, the modes of lubrication are typically characterized by the spacing between nominal lines that define smooth surfaces representing the average of the asperities. Depending on the distance between the nominal lines, h(x), three different modes of lubrication are possible

a. Complete hydrodynamic-lubrication
b. Mixed-lubrication
c. Complete boundary-lubrication

In pure hydrodynamic lubrication, an adequate quantity of oil separates the two surfaces to achieve no asperity contact between them. The transition from pure hydrodynamic lubrication to mixed lubrication occurs when the following criteria is met [21]:

$$\boxed{\frac{h}{\sigma} < 4}$$

Where, $\sigma = \sqrt{\sigma_{ring}{}^{2} + \sigma_{liner}{}^{2}}$ is the combined roughness between the ring surface and the liner. In mixed lubrication, there is oil between the two surfaces in contact. However, there is another portion of the ring and liner surfaces between which the spacing is sufficiently small that satisfy the above equation [21]. Therefore, these parts of the surfaces are also considered to be in boundary contact. There is a transition between mixed lubrication and pure boundary lubrication, which occurs when the wetting between the ring and the liner completely disappears, and there is therefore no more oil between the ring and the liner.

The friction force, for the mid stroke region of the piston movement can be useful for the assessment of variations in the lubrication conditions following from different tribological parameter combinations. Weighted average friction measurement results, in terms of the coefficient of friction or the friction force, for the entire working cycle contain less information for use in lubrication analyses but they are more useful for the assessment of the frictional power losses of the engine. For enabling calculations of the frictional power loss on the basis of a friction coefficient curve for a piston/liner pair, the normal force between the piston assembly and the cylinder liner needs to be known in detail. Weighted average friction force measurement results can furthermore be expressed as the friction pressure, p_f, which is the difference between indicated mean effective pressure and brake mean effective pressure [21].

2.5.3 Effect of cylinder liner surface finishing and coating

A plateau-honed cylinder liner surface profile, which consists of a fairly flat base surface with a network of deep scars, has been found appropriate for the lubrication of the piston assembly. The oil-retaining volume of the honed cylinder liner surface is of substantial relevance for the tribological performance of the system. For some decades, surface roughness parameters were considered as

- R_{SK}(profile skewness)
- Rvk (reduced trough depth)
- Ra (arithmetic average)

They have been used as measures on the oil-retaining capability of the groove pattern that has been produced onto the liner surface by honing. Durga and co-workers have investigated into the effect of different surface roughness values of a Cast Iron cylinder liner on the coefficient of friction. Plasma-sprayed coatings, even with a fairly coarse surface finish (Ra 0.3 µm), gave lower coefficients of friction than the honed cast-iron cylinder liners under identical test conditions[27]. According to the work by Galligan and co-authors, the coefficient of friction in the beginning of an oscillating test is lower (0.1 against 0.13) with a highly polished cylinder liner than with a liner with a standard surface finish. However, after a certain sliding distance the coefficient of friction is at the same level irrespective of the difference in initial surface quality [5].As described below, in the sections concerning scuffing, the surface quality of the cylinder liner largely determines the scuffing resistance of the cylinder and piston assembly combination, from which point of view too, a smooth a surface liner may be unfavorable. From the above investigations on the effect of the cylinder liner surface quality for the engine friction, it can be concluded that an optimum surface quality can be established, which considers the aspects of both friction reduction and scuffing suppression.

2.5.4 Effect of cylinder liner out-of-roundness

Deviations from cylindricity of cylinder liners cause local variations in the contact pressure between the piston rings and the cylinder. The wear of the cylinder liner and the likelihood of bore polishing and piston ring scuffing [28] is likely to be pronounced on areas subjected to higher contact pressure. In the case of severe roundness errors, areas of particularly low contact pressure between ring and liner are subjected to increased risk of combustion gas blow-by, particularly with stiff piston rings and high crankshaft speeds.

2.6 Cylinder Liner Failure

2.6.1 Types of Cylinder Liner failure

In general, cylinder liners are subject to two dominant failures, wear down and thermal cracking. Generally, the wears down mostly occur in the top portion of high power marine Diesel engines and this is the area subjected to maximum mechanical and thermal loads. Other reasons of deepening destructing effects of friction and enlarged intensity of the wear of the cylinder liner sliding surface on the upper parts are mentioned below:

a. High unitary pressure in the space limited with the first pair of piston's gas rings and cylinder walls, while beginning of the exhaust stroke and at the end of the compression stroke, when the piston takes position of TDC– compressed air and gaseous combustion products being forced through ring clearances in the pistons grooves cause rings' expanding, what increases a side thrust on the sliding surface
b. High temperature of the combustion chamber walls – during engine running a temperature of the upper part of a cylinder liner achieves 523 – 533 K[56]. Hence, metal particles of cylinder liner's material demonstrate increasing mobility. This phenomenon is loaded in favour of a plastic deformation of the tip layer because of friction forces' reaction.
c. Unfavorable lubrication conditions – in the close vicinity of the combustion chamber, lubrication oil evaporates and burns out making hard carbon particles along with combustion products
d. Corrosion impact of the combustion products on the cylinder liner sliding surface at high temperature.

2.6.1.1 Liner Wear Down

The wear down of marine diesel engines mainly happens due to the high quantity of abrasive particles on the piston surface, which occurs as a result of combustion of heavy fuels and oil degradation (soot). The wear of cylinder liner is mainly due to the following causes:

a. As a result of friction
b. As a result of corrosion
c. Abrasion
d. Due to scuffing

The cylinder liner wear due to corrosion is caused mainly due to the high Sulphur contents in the fuel, especially with heavy fuel, low load operation and when the cylinder oil quantity is not matched with the load, it may lead to corrosion of liner. Abrasion type of cylinder liner wear is mainly due to the hard particles present and formed during combustion and it is normally high at TDC and BDC of the liner. Scuffing is a form of local welding between the particles of piston rings and the liner surface and when the piston is moving inside the liner, the welding which has occurred breaks and leads to the formation of abrasive material. Due to this type of wear the liner loses its properties to adhere cylinder oil to the surface and polishing of the surface caused by scuffing, gives the liners a mirror finish.

2.6.1.2 Thermal cracking

Cylinder liner's cracks are the most dangerous failures for the engine's reliability. They are usually caused by exceeded mechanical and thermal stresses occurring within the walls creating the combustion chamber. The covering cylinder liner's external walls with boiler scale disturbing hit abstraction from cylinder liners and pistons represents the most frequent reason of the cracks existence in the cylinder liners cooled with water. As a consequence, considerable thermal gradients within the cylinder walls and excessive thermal deformations occur. Additionally, lubrication conditions get worse and it causes cracks of the structural material

[61].Thermal deformations of cylinder liner in the vicinity of TDC achieve more than 100μm[62].

Investigations into the cracking in cylinder liners installed in ALCO 251 diesel engines in nuclear standby service have identified the conditions which contributed to cracking of the upper cylinder liner flange. It is demonstrated that the cracks are initiated via high cycle fatigue, but that most cracks can be arrested. A cylinder liner with an arrested crack of this type would be expected to continue to provide reliable service. For worst case conditions, low-cycle thermal fatigue may cause crack growth to continue beyond the normal arrest point, ultimately causing catastrophic failure of the cylinder liner flange[59].

Subodh Kumar Sharma& co-workers have carried out an experimental thermal analysis of a diesel engine piston and cylinder wall. The purposes of this investigation are to measure the distortion in the piston, temperature, and radial thermal stresses after thermal loading. The method has been experimented on a water cooled, four stroke, direct injection diesel engine to check the temperature of piston and cylinder wall temperature. As per the theory of conversion of energy, heat rejected to water and air need to be equal to the heat entering into the piston from gas at steady state condition. For steady state conditions, the heat transfer rate increases with engine loading condition, with a maximum observed at full load. From the analysis, the author found that the variations in temperature are mainly responsible for the development of temperature stress, causing the appearance of cracks in the body of the piston[63].

2.6.1.3 Similar type of failures on other engines

As per the history of MTU engines, the German origin engine widely used for marine and industrial applications, cylinder liner scraping or seizure has been occurred in MTU 20V TB63 engines mainly utilized for cruisers around the world. After a series of investigations into the OEM of the engine, MTU Friedrichshafen has found that during most of these occasions the ships are operated at low load conditions. As a solution the OEM has introduced a cylinder liner with Coke Ring, to emit carbon deposits during the operation of the engine. The concept of the Coke Ring is similar to the Anti Polishing Ring (APR).

2.7 Summary

Even though many researches have been conducted in the field of liner wear analysis, comparatively less attention has been paid to the area of thermal expansion of the cylinder liner during the cylinder liner cooling process. Moreover, the effects of the thermostats in cooling water temperature variation also have not been addressed properly in the researches. Therefore, attention towards the thermal behavior of the cylinder liner during its operation and possible alterations and adjustment recommendations will be paid in this research.

3. ANALYSIS OF RESULTS

3.1 Introduction to the analysis

The aim of this analysis is to find the root cause for the prevailing cylinder liner/piston seizure of Deutz TBD 620 engines operating in the SLN and find a suitable solution to overcome this problem. During this analysis, existing components of several engines and also brand new spares from stocks were examined. To obtain experimental data, especially to examine parameters of cylinder unit which are not indicated on the panel boards, two Fast Attack Craft fitted with Deutz TBD 620 engines was utilized after obtaining permission from the Sri Lanka Navy. In addition, several seizure components were also analysed to check the pattern of seizure. The SLN has already carried out metallurgical analysis of defective components with the assistance of the OEM and has found no deviations from the standard specifications. Hence, during this analysis, attention was not paid to the metallurgical aspects of the cylinder liner.In order to analyse this critical problem, the following analysis will be carried out in the proceedings:

a. Analysis of running parameters of cylinder liner
b. Surface texture of cylinder liner.
c. Geometrical analysis of cylinder liner and associated components.
d. Low load running of the engine

3.2 Analysis of Running Parameters of Cylinder Liner

It is important to study the various parameters related to the cylinder liner of the engine. However, only the basic parameters of the engine are indicated on the operating panel of the engine. Hence, two engines were selected which have operated around 4,000hours after the first major overhaul and the measurement set up has been arranged to obtain all the internal temperatures of the cylinder liner, which are not visible on the control panel. The results of this measurement test can be used to analyse whether there are structural abnormalities when the performance is compared to the original test bed protocol of the engine. While arranging the instrumented cylinder liner, concurrence was obtained from the OEM whether the cylinder liner is not subjected to heavy stress by this arrangement, during the operation of the engine.

On 12th May 2017, a measurement test took place at the port side engine (left side) of the Fast Attack Craft pennant number P473. The craft is fitted with 02 Nos. Deutz TBD 620 engines, which has subjected to 03 cylinder liner seizures since commissioning of the engine. Thereafter, the same experimental test was carried out on engines fitted on-board the P424 craft.

To get detailed information about the performance and especially the water temperature behaviour in the engine, a special instrumented liner with three thermocouples was installed in unit A3. The units on the A bank, and in particular A3, A4 and A5 have shown the most sensitive units and suffered from piston seizures many times.

First a reference test was carried out, with different load steps. Out of this data, the reference data was collected. As the second part of the measurement test, various load changes were carried out during sea trials. With these tests, the dynamical behaviour of the cooling system is to be investigated.

3.2.1 Set up of the measurement test

Instrumented Liner

In order to record the line temperatures, an instrumented liner was prepared as follows, and equipped with three thermocouples. These thermocouples were distributed evenly on the water side of the liner, where the tip of the thermocouple was installed approximately halfway into the liner wall.

Figure 3.1: Instrumented Cylinder Liner

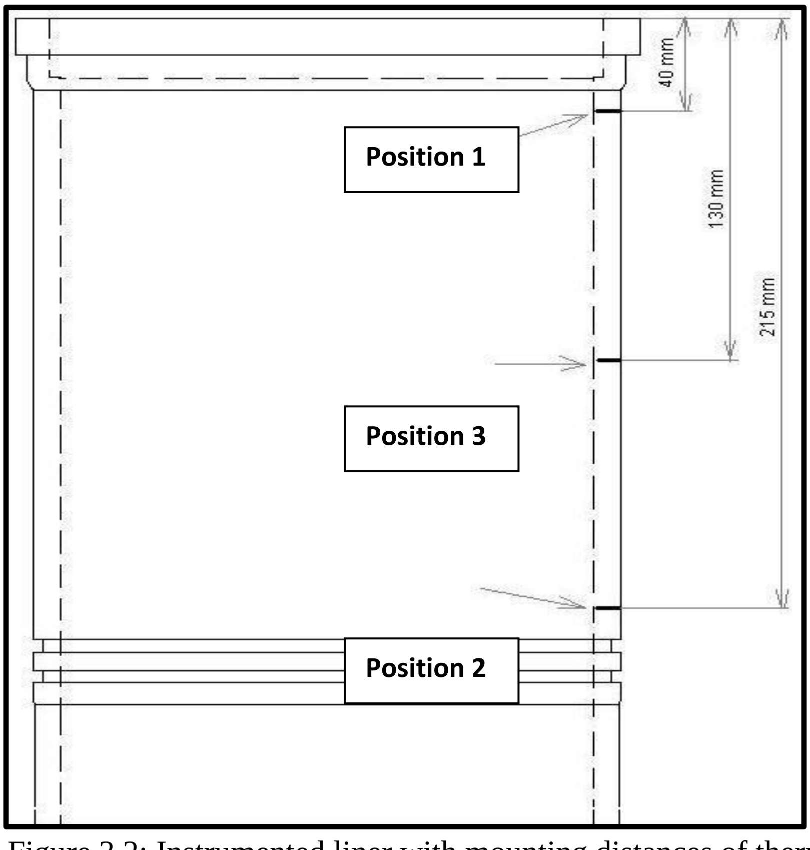


Figure 3.2: Instrumented liner with mounting distances of thermo couples

Data logger

The Data Logger has specially been designed with data log equipment to record a maximum of 40 parameter signals from the engine, like pressures and temperatures. This equipment collects the signals and sends these via processors to the laptop in which the data will be stored. From these data, graphs can be generated as well.

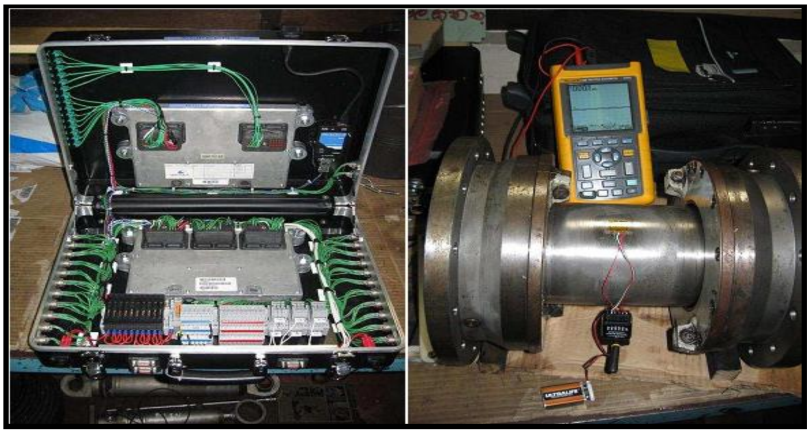


Figure 3.3: Data log suitcase and strain gauge on the drive shaft

3.2.2 Measurement Schedule

After installation of the test liner, three load steps were recorded for reference during the running programme:

Table 3.1: Load Steps of the Experiment (first stage)

Step	RPM	Load according propeller curve [%]	Running time [min]
1	1620	75 %	20
2	1710	85 %	20
3	1800	100 %	20

For the second part of the measurement test, the purpose was to monitor the load changes and reaction of the systems to the load changes. Therefore, the following load steps were applied:

Table 3.2: Load Steps of the Experiment (second stage)

Step	RPM [min*-1]	Load according propeller curve [%]	Running time [min]
4	1440	51 %	20
5	1200	30 %	20
6	1800	100 %	10
7	1200	30 %	10
8	700	7 %	45
9	1800	100 %	10

3.2.3 Measurement data

<u>Recorded data at steady state conditions:</u>

Engine speed : 1800 rpm

Engine load : 1620 kW, with 2.4% gearbox losses

HT cooling water A bank out : 81 °C

HT cooling water B bank out : 81 °C

HT cooling water in : 56 °C

Average exhaust gas T cyl.headA bank : 525 °C

Average exhaust gas T cyl.head B bank : 528 °C

Average receiver pressure : 1334 mbar

Lubricating oil temperature : 79 °C

Inlet air temperature : 52 °C

Liner thermocouple 1 : 125°C

Liner thermocouple 2 : 95 °C

Liner thermocouple 3 : 102 °C

Average exhaust gas T after TC : 565 °C

The following Graph showsthe recorded parameters when the engine is run at 1800 rpm, steady state condition at 100% load.

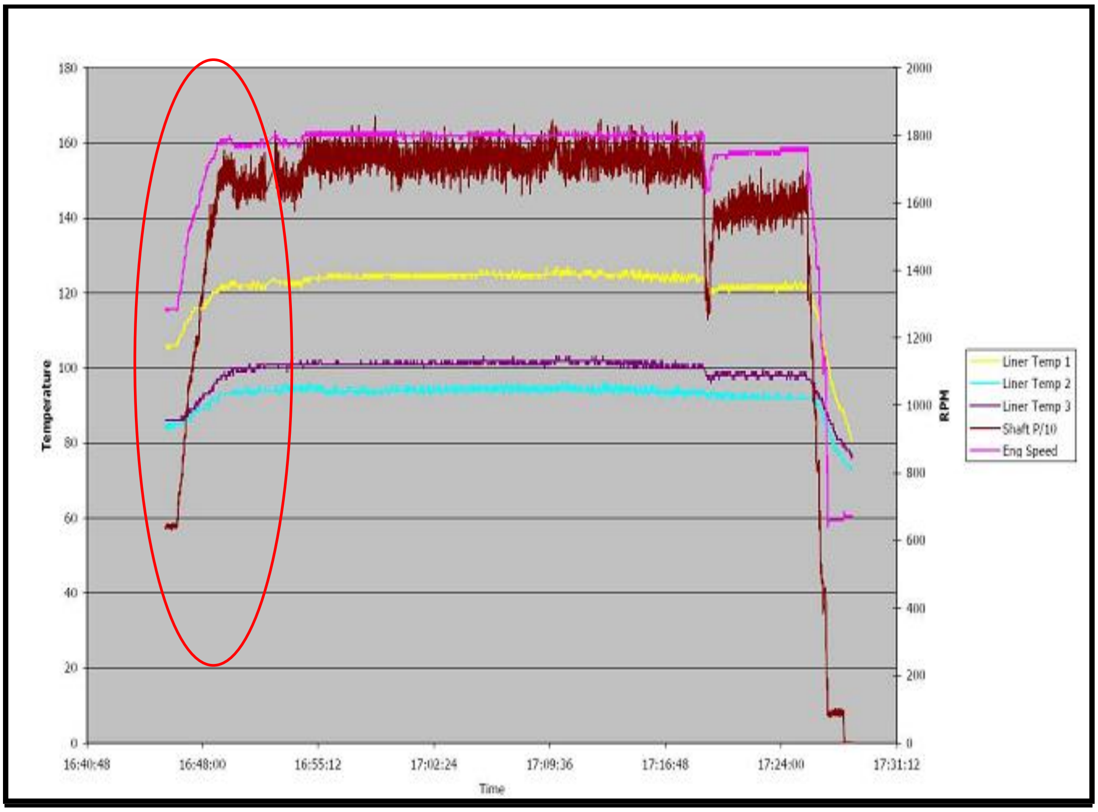


Figure 3.4: Liner temperatures, shaft speed and engine speed (steady state)

On the X – axis is the time and on the Y- axis is the temperature on the left and the RPM on the right side. At around 17.18 hrs, there is a sudden change in rpm, probably because of manoeuvring or turning of the craft. At around 17.28 hrs the rpm was back at idling with no load. When the rpm of the engine increased gradually from 700rpm to 1800rpm, the cylinder liner temperature also reached its elevated temperature simultaneously (the particular area is marked above in red colour).

The following pair of Graphs shows the measurement step 9, from 700 rpm rapidly increasing until 1800 rpm. In this condition, even though the rpm of the engine reached its maximum 1800 rpm, the cylinder liners have not reached the elevated temperature and the time taken to reach this temperature is considerably high (this area of the graph is marked in red colour).

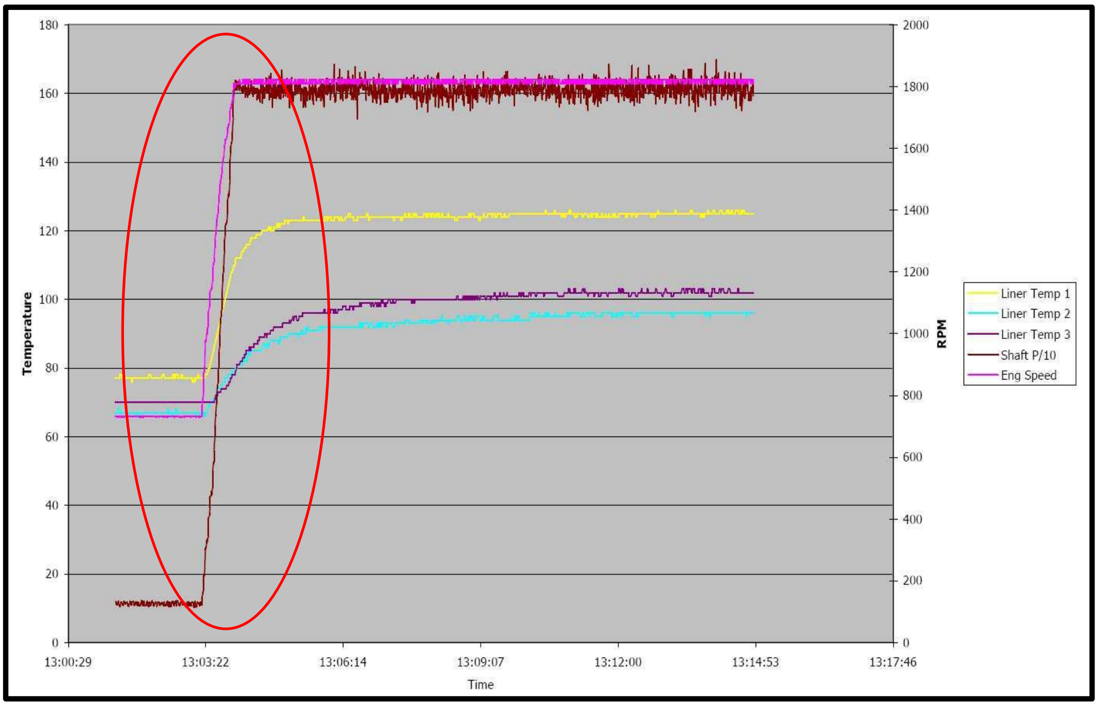


Figure 3.5: Liner temperatures, shaft speed and engine speed (rapid increase)

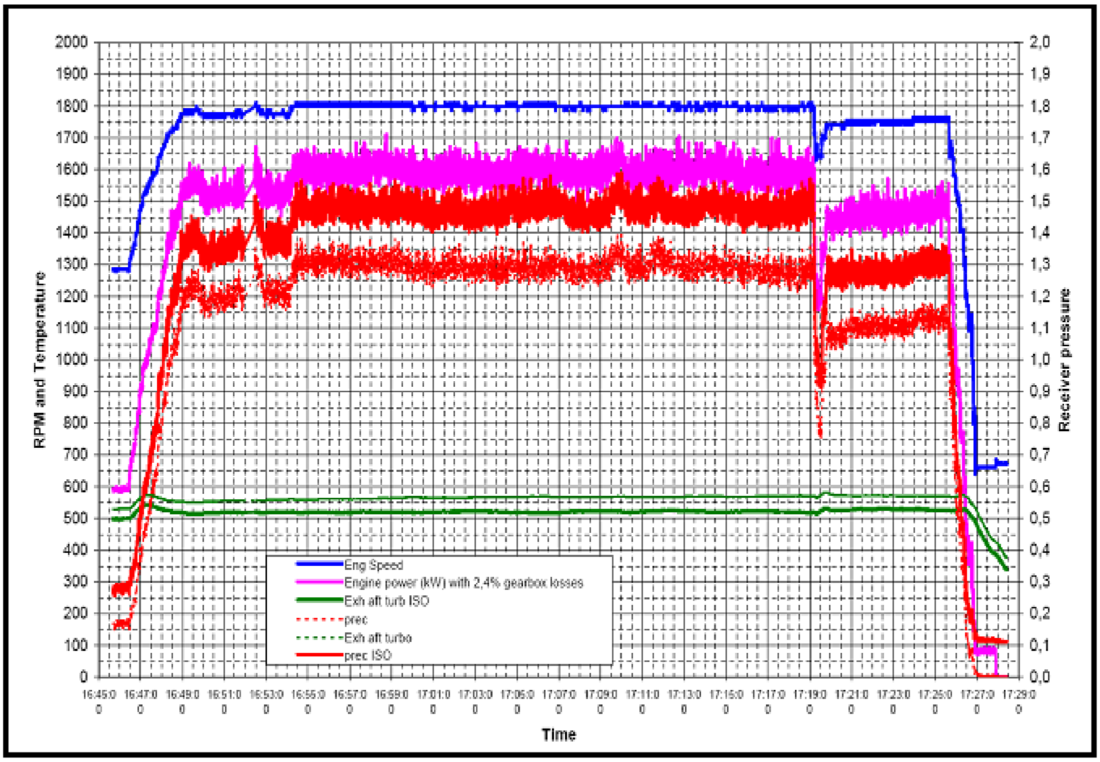


Figure 3.6: Engine speed, Exhaust gas and receiver pressures

The exhaust gas temperature showed an slight increase compared to the factory acceptance test, new condition of this engine, but that is a phenomenon to be expected (normal wear of the engine with age).

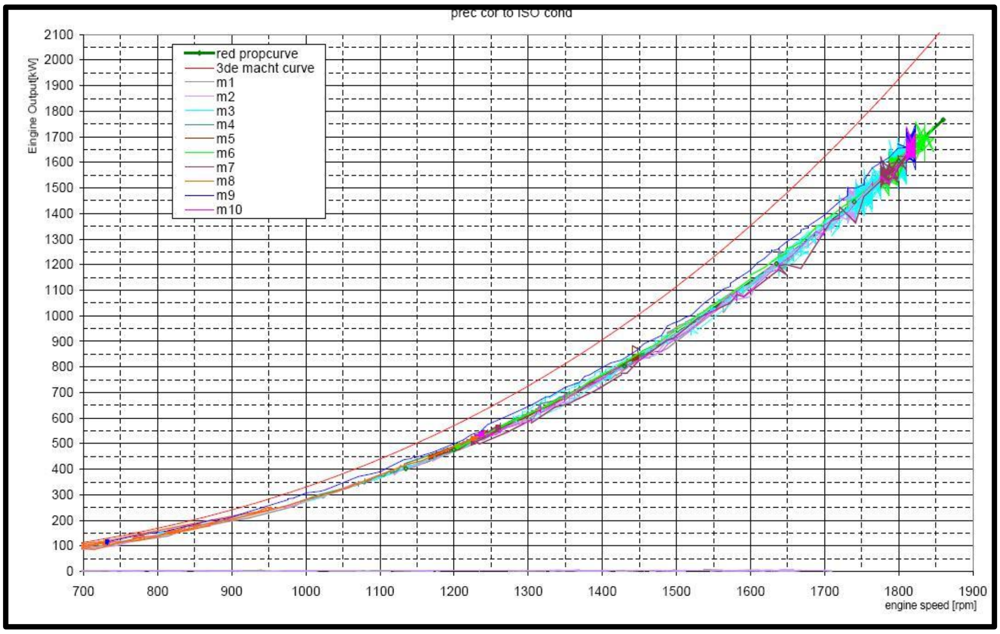


Figure 3.7: Comparison measurement step 1-9 with theoretical power curve

The Graph shows the theoretical power curve, combined with measurements step 1–9. All measurements are in the safe area and not exceeding the theoretical curve, proving that the engine is not overloaded, neither statically or dynamically during acceleration.

Comparison of recorded data (summary)

Running time with 700 rpm : 45 min

Steady state conditions at 700 rpm

HT cooling water out average : 63 °C

HT cooling water in : 56 °C

Liner thermocouple 1 : 77 °C

Liner thermocouple 2 : 67 °C

Liner thermocouple 3 : 70 °C

Conditions during acceleration from 700 – 1800 rpm:

Acceleration time 700 – 1800 rpm : 33 seconds

HT cooling water out average : 64 °C

HT cooling water in : 56 °C

Liner thermocouple 1 : 110 °C

Liner thermocouple 2 : 79 °C

Liner thermocouple 3 : 80 °C

Steady state temperatures @ 1800 rpm:

HT cooling water out average : 81 °C

HT cooling water in : 56 °C

Liner thermocouple 1 : 128 °C

Liner thermocouple 2 : 95 °C

Liner thermocouple 3 : 102 °C

Evaluation of the measurements

Within these measurements, no significant deviations have been discovered, other than the cooling water temperature, which could directly affect the performance of the cylinder liner. When increasing load of the engine rpm from 700-1800, even though it has taken 33 seconds, the HT cooling water temperature out still at approximately 63 °C. This load variation of the engine was carried out in 45 seconds and still HT cooling water temperature out was increased only up to approximately 66°C.The cooling water out of the engine has to be approximately (80-82)°C at this stage within (30-40) seconds duration. However, it took much longer, approximately 60 seconds to reach this value.

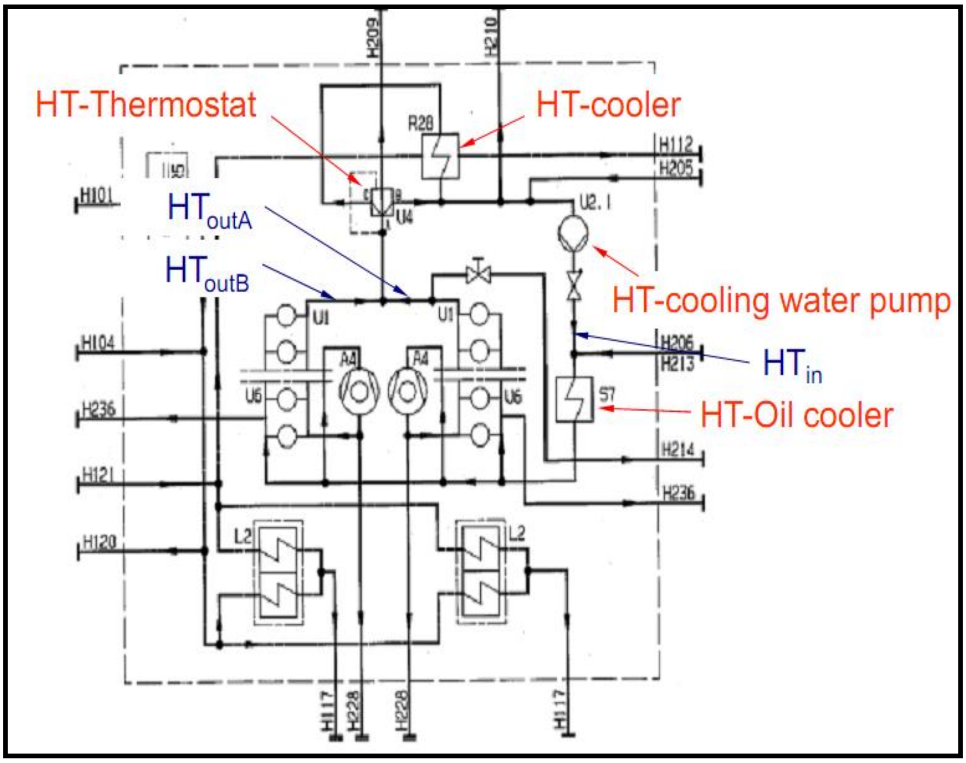


Figure 3.8 HT in and out of the engine [1]

Cooling water temperature logs available on-board shows the majority fluctuation HT temperature under different loads. Especially with low loads, the temperature of

HT drops below the minimum level. In normal operation, a temperature of 75°C should be maintained minimally. This is also the operating temperature of the thermostats.

In addition, the temperature difference between HT cooling water temperature in and out is also very high (13-17°C in both craft) and standard value should be within (7-12) °C, indicating that the thermostatic control valves operation needs to be improved. In order to further clarify, the HT cooling water temperature difference of the engine was recorded at various rpm range of the engine and the recorded values are as follows:

Table 3.3: HT cooling water difference of the engine

Measurements						
Step	Engine speed (min -1)	Engine load (%)	HT Cooling water °C			
			in	out A-bank	out B-bank	dHT
1	1620	75	57.5	73.5	73.5	16
2	1710	85	57.5	73.5	73.5	16
3	1800	100	57	74.5	74.5	17
4	1440	51	56	73	73	17
5	1200	30	54	70	70	16
6	1800	100	57	74	74	17
7	1200	30	55	69	68	14
8	700	7	50	63	63	13
9	1800	100	56	74	74	18
10	1200	30	54	69	69	15

(7 – 12)°C is the normal value of dHT

The recorded values indicated that the difference of HT cooling water in and out temperatures are considerably high. As per the recommendation of the OEM, the standard limitations of HT cooling water temperature and HT cooling water temperature difference applicable for Deutz TBD 620 engine are as follows:

Table3.4: Specified limitations by the OEM [1]

PAGE 24 01	ISSUE 05 - 00	TBD 620	Technical Data			DEUTZ
Coolant						
Construction type		Dim	V8	V12	V16	Remark
Content						
Cooling water content of engine with LJI cooling	without heat exchanger	dm^3	70	100	140	
	with heat exchanger		85	120	160	
	additional filling volume with LJI cooling					
Temperature						
Engine - cooling water outlet				82		
Engine monitoring	Pre - warning			85		
	Shut - off			90		
Temperature difference between Engine inlet and outlet				7 to 12		Including oil cooler

These HT cooling water temperatures were compared with the initial parameters recorded during the commissioning of the engines in year 2001, at the Test Bench. It appears that the temperature difference between HT cooling water temperature in and out of the engine is well within the limit and maximum HT is also above 75°C.

.

Table 3.5: Test bench recorded data of the engine

Acceptance test									
Load		Speed (min-1)	HT cooling water (oC)				Lub oil cooler	
[kW]	%		in	out A-bank	out B-bank	d(HTout-HTin)	inlet	out
195	10	1800	73	75	75	2	71	73
484	25	1800	74	77	76	3	71	74
968	50	1800	74	78	78	4	71	74
1451	75	1800	75	79	79	4	71	75
1935	100	1800	75	81	81	6	71	75
195	10	1800	73	75	75	2	70	73
573	30	1200	74	79	78	5	71	74
729	38	1300	75	79	78	4	71	75
910	47	1400	75	80	79	5	71	75
1935	100	1800	75	81	80	6	71	75
1935	100	1800	75	81	81	6	71	75
2129	110	1860	76	82	81	6	71	75
1742	90	1740	75	81	80	6	71	75
1645	85	1530	75	82	81	7	70	75
1451	75	1635	75	81	80	6	71	75
968	50	1430	74	80	79	6	71	74
484	25	1135	74	78	77	4	71	74
72	4	1200	74	77	76	3	71	74
110	6	588	73	75	74	2	71	73

3.2.4 Conclusion of the analysis

From the measurement test, the following findings can be highlighted:

a. No significant deviation of exhaust temperature, and engine is not overloaded during operation and acceleration.

b. The most significant abnormality was the low HT cooling water temperature and low response of HT cooling water temperature during load variation of the engine. Observed that the HT temperature is about 16°C low and the HT in and out difference is also considerably high. These deficiencies will contribute to the dangerous clearance area where seizure of piston/cylinder liner could occur. This needs further attention.

c. The instrumented liner has given some input data by means of temperature and temperature differences. These data will be used to calculate and review the component clearances of piston and cylinder liner combination.

3.2.5 Examine the behaviour of cooling water temperature during the load variation

In the Deutz TBD 620 engine, a thermostatic valve senses the temperature at the engine outlet and in steady state of the engine it can control the coolant temperature satisfactorily. However, as per the recorded parameters, it hardly compensated the coolant water temperature during the transient load changes. During manoeuvring of the Fast Attack Craft under various conditions and situations, especially during the humanitarian operation, the operators were compelled to increase or decrease the rpm of the engine very quickly. Even after the humanitarian operation, Fast Attack Craft are heavily engaging against the poaching of Indian fishermen and to counter illegal activities.Hence, fast manoeuvring of craft is done very often. When the cooling system does not respond properly, it affects the functioning of the cylinder liner and piston as the cooling water temperature has significant influence on clearance between liner and piston. Hence, there is a risk of cylinder liner seizure.

When analyzing the coolant system of the engine and low response of cooling water with the load variation, thermostat valve is an important element as it maintains the temperature of the engine at its optimum condition by adjusting the flow of coolantto the heat-exchanger (in this engine plate cooler). The Deutz marine engine consists with 02 Thermostatic valves, which are operated at 75^0C. The thermostat uses wax pellets inside a sealed chamber for its operation. The sealed chamber operates a rod which opens a valvewhen the operating temperature is exceeded. The following are possible factors affecting the low response of cooling water temperature:

a. When the thermostat is not sensitive enough and it does not response quickly and ultimately affected the cooling water temperature.

b. Risk of sticking element, which can delay the response resulting in fluctuating cooling water temperature and also results in achieving low HT temperature.

The arrangement of thermostatic valves of the Deutz 620 engine is as follows:

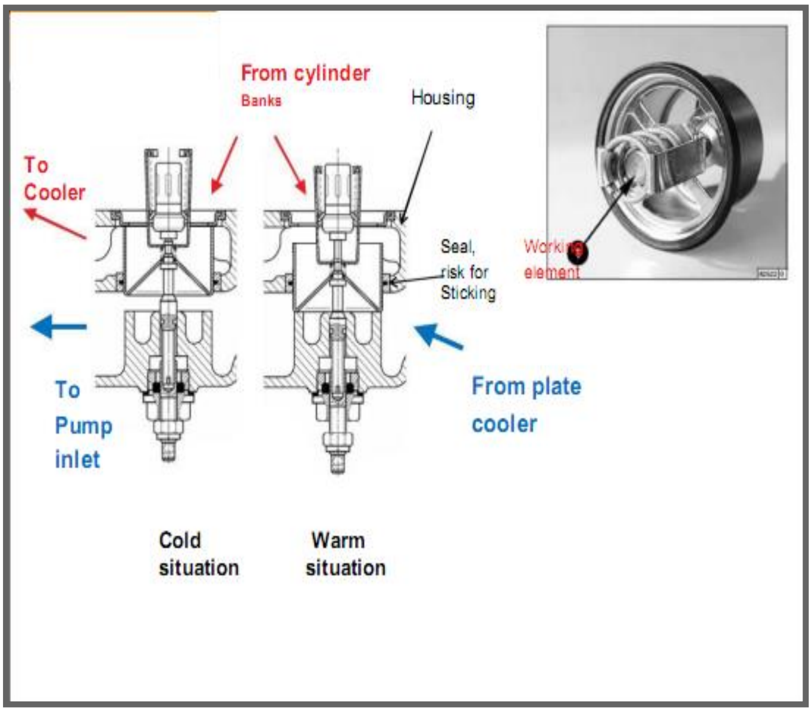


Figure 3.9: Operation of the thermostat and working element (wax)

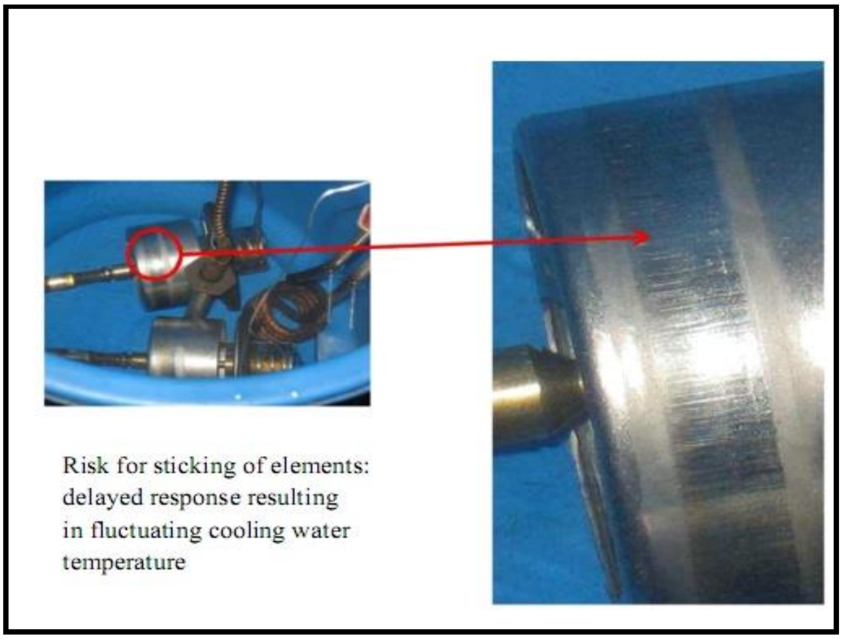


Figure 3.10: Risk of sticking element

In order to get an idea about the operation of the existing wax element thermostats, a few thermostats were removed and their operation examined by immersing them in water. The water sample gradually heated and it was observed that it started to operate at around 70^0C and fully opened at 75^0C. This operation consumed around 20 seconds. But during the real operation of the engine, the load changes has to first create more heat or less heat in the cooling water before it is sensed by the wax elements. The element responds by closing or opening only after sensing this. For the sensing activity too the thermostat consumes time and ultimately the total response time during transient load changes of the engine is considerably high. Therefore, the wax element always lags behind the engine performance, thus creating much more deviations in the required cooling water temperature.

3.3 Analysis of Geometrical Aspects of Cylinder Liner

It is important to analyse the dimensions and clearances of the moving parts of the cylinder unit as these clearances are helpful for lubrication and to maintain a sufficient amount of oil between the two surfaces to achieve pure hydrodynamic lubrication. Due to incorrect dimensions or high thermal loads, the operation clearance between the piston and cylinder can be reduced to a permissible limit or completely eliminated.

When considering the piston and cylinder liner, thermal expansion of the piston is higher than the cylinder liner. Further, thermal expansion of Aluminium materials is approximately twice that of grey cast iron and these facts are important during design stage of an engine.

During the analysis, 16 Nos. brand new cylinder units from the stock and 16 Nos. cylinder liners, pistons and connecting rods removed from an engine which has operated approximately 4500 running hours were measured to analyse the running clearances of the components.

3.3.1 Inner diameter of cylinder liner

The first measurement taken was the inner diameter. These measurements have been performed in an air conditioned room at 20°C. The inside diameter off all liners have been measured fully circumferential at depths of 30mm, 110mm, 260mm and

340mm from the top of the liner. An example of an inner diameter measurement is shown in the figure below.

According to the drawing, the inner diameter of the liners is defined as 170G5, meaning a nominal diameter with a tolerance field (G5). This tolerance is +0,025, to + 0,340 mm. In the following figure, the measurement are presented based on the nominal radius of the liner (170/2 = 85 mm). Therefore the tolerance field is also based on the nominal radius, resulting in a tolerance field of +0.0125 to + 0.17 mm. As shown in the following figure, at the full circumferential liner diameter, 08 measuring points have been taken on each cylinder liner.

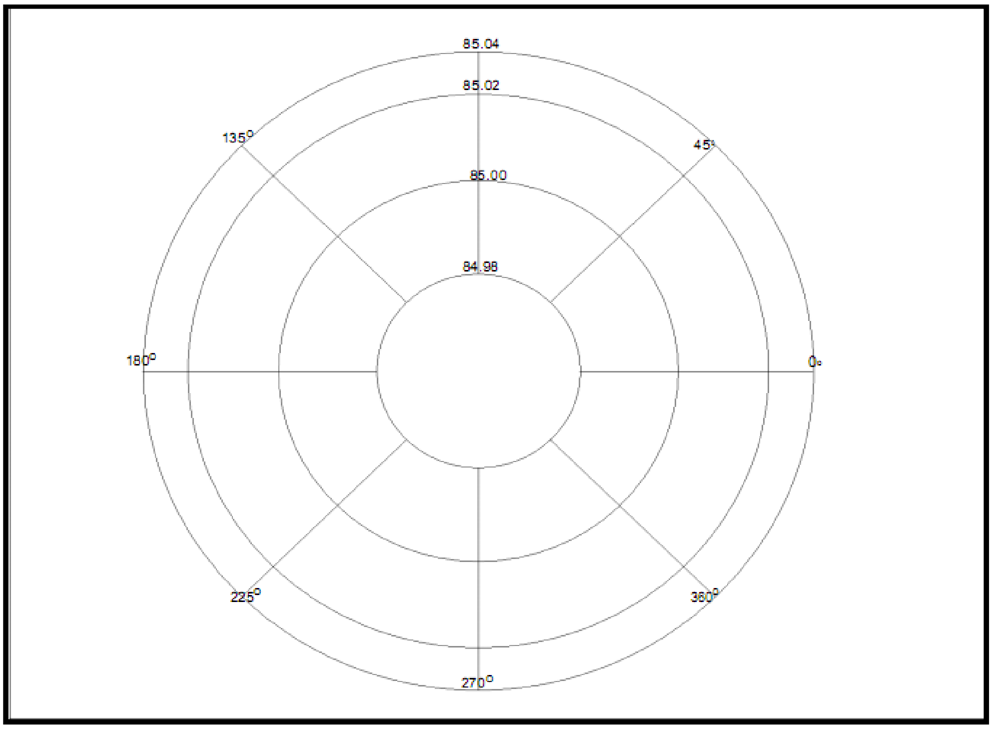


Figure 3.11: Measuring planes cylinder liner inner diameter

Table 3.6: Measurement Records of Cylinder Liner Bore

CYLINDER NO		DIMENSION(85+/-)			
		MEASURING PLACES FROM TOP 30mm 110mm260mm340mm			
A1	X	85.03	85.02	85.00	84.98
	Y	85.03	85.02	85.00	84.98
A2	X	85.03	85.02	85.00	84.98
	Y	85.03	85.01	84.99	84.99
A3	X	85.03	85.02	84.99	84.98
	Y	85.03	85.01	85.00	84.99
A4	X	85.03	85.02	84.99	84.98
	Y	85.03	85.01	84.99	84.99
A5	X	85.03	85.02	85.00	84.99
	Y	85.03	85.02	85.99	84.99
A6	X	85.03	85.02	85.00	84.98
	Y	85.03	85.01	84.99	84.99
A7	X	85.03	85.01	84.99	84.98
	Y	85.03	85.02	84.99	84.98
A8	X	85.03	85.01	84.99	84.98
	Y	85.03	85.02	84.99	84.99

CYLINDER NO		DIMENSION(85 +/-)			
		MEASURING PLACES FROM TOP 30mm110mm260mm 340mm			
B1	X	85.03	85.02	85.00	84.98
	Y	85.03	85.02	85.00	84.98
B2	X	85.03	85.02	85.00	84.98
	Y	85.03	85.01	84.99	84.99
B3	X	85.03	85.02	84.99	84.98
	Y	85.03	85.01	85.00	84.99
B4	X	85.03	85.02	84.99	84.98
	Y	85.03	85.01	84.99	84.99
B5	X	85.03	85.02	85.00	84.99
	Y	85.03	85.02	85.99	84.99
B6	X	85.03	85.02	85.00	84.98
	Y	85.03	85.01	84.99	84.99
B7	X	85.03	85.01	84.99	84.98
	Y	85.03	85.02	84.99	84.98
B8	X	85.03	85.01	84.99	84.98
	Y	85.03	85.02	84.99	84.99

3.3.2 Clearance calculations between Cylinder Liner and Piston at elevated temperatures

Based on the designed data and measured data, transient calculations have been performed in order to determine the influence of the cooling water temperature on the clearance between cylinder liner and piston during the operation of the engine. Further data such as liner temperatures are based on measurement data obtained during the previous analysis.

The circumference of the cylinder liner can be expressed as

$c_0 = 2 \pi r_0$ (1)

where

c_0 = initial circumference (m)

$\pi = 3.14$

r_0 = initial radius (m)

The change in circumference due to temperature change can be expressed as

$dc = c_1 - c_0$

$=2 \pi r_0 \, dt \, \alpha$ (2)

Where,

dc = change of circumference (m)

c_1 = final circumference (m)

dt = temperature change (^oC)

α = linear expansion co-efficient (mm/m^oC)

The final circumference can be indicated as

$c_1 = 2 \pi r_1$ (3)

where

r_1 = final radius (m, inches)

Equation 1, 2 and 3 can be expressed as

$dc = 2 \pi r_1 - 2 \pi r_0$

$= 2 \pi r_0 \, dt \, \alpha$

or transformed to

$r_1 = r_0 \, dt \, \alpha + r_0$

$= r_0 (dt \, \alpha + 1)$(4)

Equation 4 can be modified with diameters to

$d_1 = d_0 (dt \, \alpha + 1)$ (5)

$d_1 = d_0 (dt \, \alpha + 1)$

As per the experimental data obtained during the analysis Para 4.1, the recorded temperature values of cylinder liner are used for the calculation.

Average temperatures at TDC and BDC of cylinder liner when increasing load from 700rpm to 1800rpm

TDC= 110^oC

BDC=80^{o}C

Average temperature at TDC and BDC of cylinder liner at steady state of1800rpm

TDC = 128^{o}C

BDC = 100^{o}C

Ambient temperature of measurement room = 20^{o}C

Linear expansion coefficient of Cast Iron (m/m^{o}C) = 10.6 x 10^{-6}m/m^{o}C

Calculation of diameter of cylinder liner at TDC and BDC, when increasing load from 700rpm to 1800rpm

Final diameter of the cylinder liner at 1800rpm of the engine and at TDC level, when increasing load from 700rpm to 1800rpm.

$= 170.08 \text{ X } 10^{-3} [((110\,^{o}C) - (20\,^{o}C)) (10.6 \text{ X } 10^{-6}) + 1)$

= 170.242m

Final diameter of the cylinder liner at 1800rpm of the engine and at BDC level in same load condition when rpm increasing from 700rpm to 1800rpm.

$= 169.99 \text{ X } 10^{-3} [((80^{o}C) - (20\,^{o}C)) (10.6 \text{ X } 10^{-6}) + 1)$

=170.098 m

Calculation of diameter of cylinder liner at TDC and BDC, at steady state of 1800rpm

Hence, final diameter of the cylinder liner at steady state of 1800rpm of the engine and at TDC level,

$= 170.08 \text{ X } 10^{-3} [((128\,^{o}C) - (20\,^{o}C)) (10.6 \text{ X } 10^{-6}) + 1)$

= 170.275m

Final diameter of the cylinder liner at steady state of 1800rpm of the engine and at BDC level

$= 169.99 \text{ X } 10^{-3} [((100^{o}C) - (20\,^{o}C)) (10.6 \text{ X } 10^{-6}) + 1)$

=170.134 m

Accordingly, the piston diameter was also calculated at the elevated temperature at 1800rpm of the engine as follows:

Table 3.7: Measurement of Piston Diameter

PISTON NO		POSITION				
		CROWN	SKIRT		CROWN	SKIRT
A1	X	168.50	168.45	A1	168.52	168.55
	Y	169.65	169.60		169.67	169.60
A2	X	168.50	168.48	A2	168.44	168.45
	Y	169.68	169.60		169.65	169.68
A3	X	168.52	168.50	A3	168.53	168.50
	Y	169.60	169.70		169.72	169.68
A4	X	168.45	168.40	A4	168.47	168.43
	Y	169.66	169.44		169.63	169.66
A5	X	168.55	168.60	A5	168.50	168.55
	Y	169.65	169.64		169.76	169.72
A6	X	168.55	168.50	A6	168.54	168.56
	Y	169.70	169.68		169.75	169.68
A7	X	168.50	168.52	A7	168.50	168.52
	Y	169.68	169.75		169.70	169.76
A8	X	168.52	168.50	A8	168.56	168.55
	Y	169.70	169.75		169.72	169.75

Linear expansion coefficient of Aluminium alloy MSFC-388-T5 = 23.1 x 10^{-6}m/m^{o}C

Average temperature of the piston = 140^{o}C

Average diameter of a piston = 169.50 mm

Hence, final diameter of the piston at 1800rpm of the engine would be,

= 169.500 X 10^{-3} [((140ºC) - (20 ºC)) (23.1 X 10^{-6}) + 1)

=169.96mm

Therefore, during operating a Deutz TBD 620 engine at maximum rpm of 1800 rpm and during load variation from 700 - 1800 rpm, the clearance values between cylinder liner and piston can be summarized as follows. As per the calculated data, the clearance between piston and cylinder liner is marginal at the TDC and out of limit at the BDC as the standard limit is (0.30- 0.40) mm.

Table 3.8: Calculated data of cylinder liner clearance

Position	**Dia. of cylinder liner (700 to 1800 rpm)**	**Dia.of cylinder liner (at 1800 rpm)**	**Dia. of piston(mm)**	**Clearance(mm) 700 to 1800 rpm**	**Clearance(mm) 1800 rpm**
TDC	170.242	170.275	169.969	0.273	0.306
BDC	170.098	170.134		0.129	0.165

The calculated data can be compared with the standard tolerance values and the details are as follows:

Table 3.9: Comparison of cylinder liner clearance

Specified tolerance limit	**Clearance(mm) 700 to 1800 rpm**	**Clearance(mm)1800 rpm**
0.295 - 0.395	0.273	0.306
0.160-0.260	0.129	0.165

As per the calculated data, the clearance between piston and cylinder liner is out of tolerance limit during increasing of rpm of the engine from 700rpm to 1800rpm for a considerable period of time and also this time period exceed the specified time indicated by the OEM. When operating the rpm at a steady state at 1800rpm, the clearance between piston and cylinder liner is within the tolerance limit; however it is very marginal. This limited clearance between liner and piston is a critical fact during high rpm and load changes of the engine.

This matter was communicated to the OEM and they indicated that with the modification of the cylinder liner with Anti Polishing Ring, the inner diameter of the cylinder liner has also been increased in order to maintain a safety margin to meet any eventuality.

3.4 Analysis of Surface Texture of Cylinder Liner

The aim of this analysis is to examine the surface texture of cylinder liners of Deutz TBD 620 engines. For this analysis, the honing roughness of the Deutz engine was compared with the other types of engines available on-board a FAC in the SLN's MTU 396 engines. During the analysis, six new cylinder liners from the stocks for each brand of engines were examined and the Surface Texture Measuring Tool was used to obtain the readings.

Figure 3.12: Surface Texture Measuring Tool

3.4.1 Honing profile

The roughness of the honing was measured and compared as well. Of every liner, 04 Nos. roughness readings have been taken from which the Abbott Curves have been defined.

An example of the determination of a honing roughness according Abbott is given in the following figure. This figure shows the roughness of a surface. This roughness can be divided into 3 areas, Rpk, Rk and Rvk.

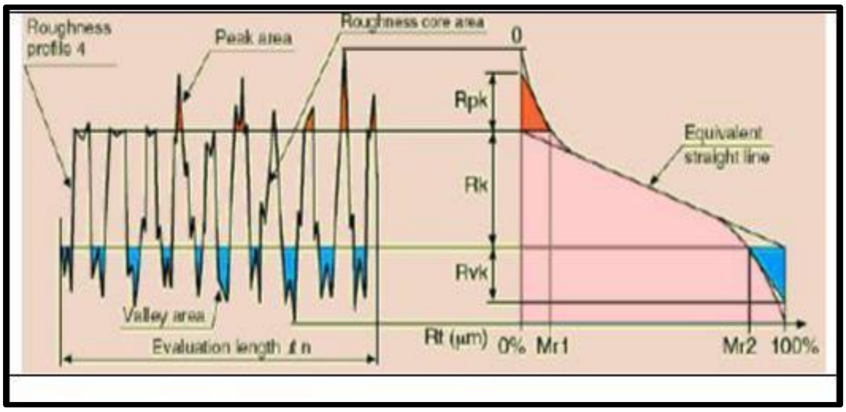


Figure 3.13: Abbott Curve -Profile to explain Rk, Rpk, Rvk[64]

Rpk is the average height of protruding peaks above the roughness of the core profile. The definition of this value is of high importance during the running in of the liner. The higher this value, there are more chances that peaks will penetrate through the lubrication film during the running in process of the liner, causing direct material contact between liner and piston which could lead to seizures.

Rk is the depth of the core roughness profile while Rvk is the average depth of the valley's projecting through roughness core profile. Also the value of Rpk is of importance representing the lubrication oil volume of the honed surface.

3.4.2 Results of the honing measurements

The measurements of the honed liner surfaces of Deutz cylinder liners and MTU cylinder liners are given in the following Tables:

Table 3.10: Roughness Measurements on Deutz cylinder liners

Roughness	Rpk (µm)	Rk(µm)	RvK(µm)
Minimum	0.57	3.21	1.58
Maximum	2.44	6.24	4.84
Average	1.44	4.62	2.79

Table3.11: Roughness Measurements on MTU cylinder liners

Roughness	**Rpk (μm)**	**Rk(μm)**	**RvK(μm)**
Minimum	0.64	3.40	2.57
Maximum	1.38	5.69	5.14
Average	1.02	4.60	3.60

Although Deutz cylinder liners as well as MTU cylinder liners are within the honing specifications, the honing roughness of the Deutz cylinder liners shows a higher absolute and average Rpk value than the MTU liners. Also the Deutz cylinder liners show a lower absolute and average Rvk value than MTU liners.

3.4.3 Evaluation of Analysis

When comparing the honing roughness of the Deutz and MTU cylinder liners, it is observed that average higher Rpk value combined with an average absolute and lower Rvk on Deutz cylinder liners.

A higher Rpk value is of importance in relation to the possibility of material contact between liner and piston. A higher Rpk value leads to more material contact between liner and piston. A lower Rvk value is related to a reduced “oil containing volume” of the honing roughness which could result in a weaker oil lubrication performance. Further, the honing pattern may not be in line with the demands of the load changes of the engine. Especially the Rpk value may be too high while the Rvk value may be too low.

3.5 Analysing load profile of Deutz TBD 620 engines operating in SLN

The rpm range of Deutz TBD 620 engines operating in the SLN is between idling rpm of 600 to maximum 1800. Generally, diesel engines need to be operated at least about 60% to 75% of their maximum rated load, however, due to the nature of operations of the Fast Attack Craft, operators are not concerned about the operating pattern. Short periods of low load running are permissible. However, operators must make sure to operate engines more than 80% at 1700 rpm for one hour periods, when the engines are operating for a prolong period at low load below 30%, that is below 1200rpm for six hours duration. In order to analyse the load profile of Deutz TBD 620 engines, running hours for the duration of 06 months of 04 Nos. Deutz engines operating in the SLN were recorded and details of each engine are as follows:

Table 3.12: Running hours of P421 (Port) and P412 (Starboard)

RPM Range	Operating HRS
600 - 800	220.55
800 - 1000	72.30
1000 - 1200	194.30
1200 - 1400	46.45
1400 -1600	95.25
1600 - 1800	49.30
	679.35

RPM Range	Operating HRS
600 - 800	260.55
800 - 1000	60.30
1000 - 1200	150.30
1200 - 1400	70.45
1400 -1600	83.25
1600 - 1800	40.30
	666.35

Craft - P 421 (Port engine)

Duration - (30/04/2017 to 30/10/2017)

Craft – P412 (Starboard engine)

Duration – (30/04/2017 to 30/10/2017)

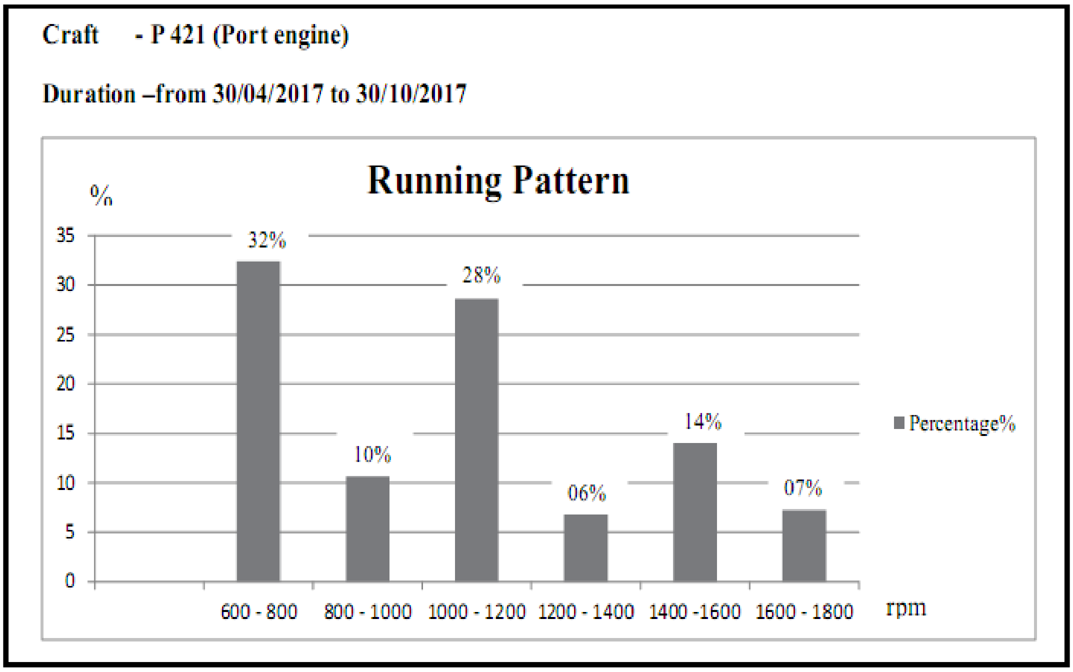


Figure 3.14: Load profile of P 421 port main engine

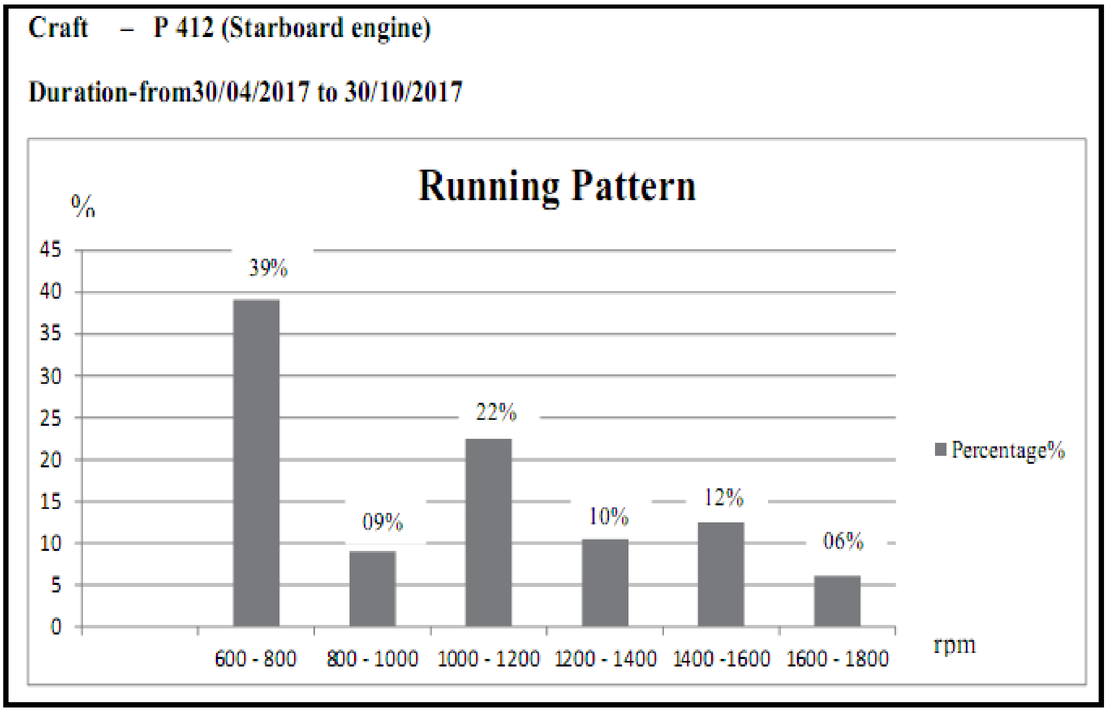


Figure 3.15: Load profile of P 421 Starboard main engine

Table 3.13: Running hours of P436 (Port) and P433 (Starboard)

RPM Range	Operating HRS	RPM Range	Operating HRS
600 - 800	233.10	600 - 800	185.10
800 - 1000	72.15	800 - 1000	112.15
1000 - 1200	193.00	1000 - 1200	200.00
1200 - 1400	45.15	1200 - 1400	65.15
1400 -1600	94.10	1400 -1600	70.10
1600 - 1800	50.10	1600 - 1800	30.10
	688.00		663.00

Craft -P 436 (Port engine)

Duration – (30/04/2017 to 30/10/2017)

Craft –P 433 (Starboard engine)

Duration- (30/04/2017 to 30/10/2017)

P 436 (Port engine)

Duration –from 30/04/2017 to 30/10/2017

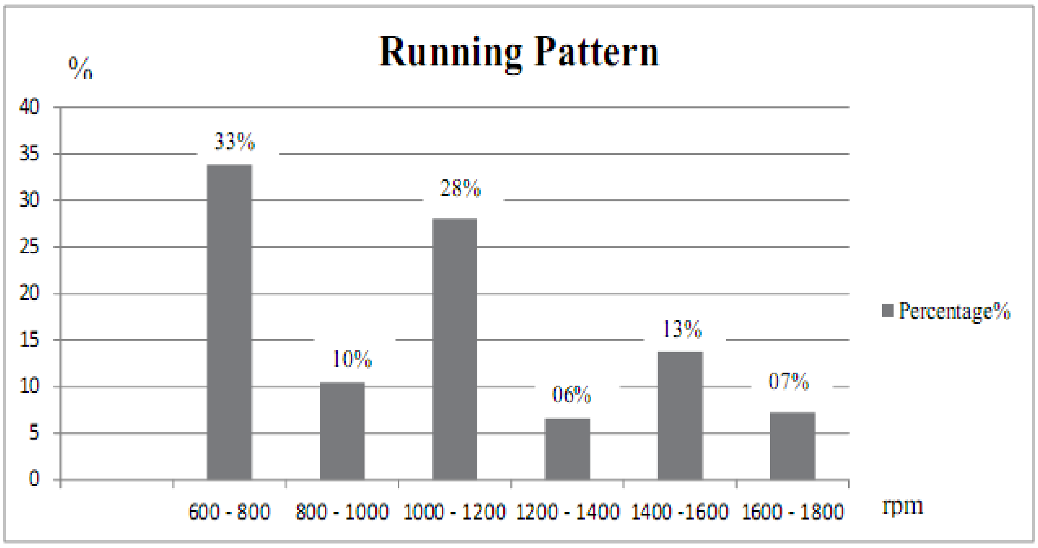


Figure 3.16: Load profile of P 436 Port main engine

P 433 (Starboard engine)

Duration-from30/04/2017 to 30/10/2017

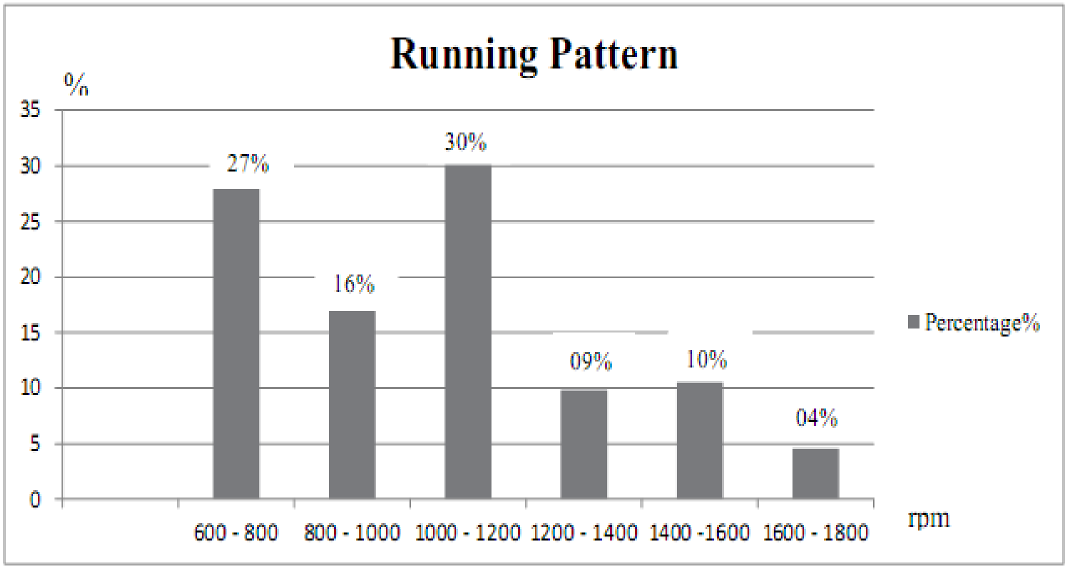
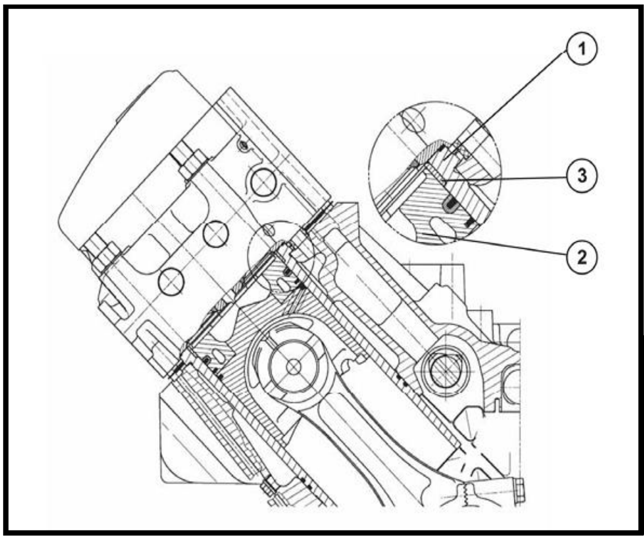


Figure 3.17: Load profile of P 433 Starboard main engine

By analysing the recorded data, it revealed that approximately 72% of operating hours are accounted for low load running of the engine that is 1200 rpm and below. Only approximately 20% of operating hours are operating between (60-80) % load and only approx. 5% running hours in maximum rpm. These statistics indicated that engines are operating at low load for longer periods of time. This is a very critical situation as running an engine under low load causes heavy carbon deposits in the combustion area and lead to sticky piston rings. Prolonged operation in this condition can cause cylinder liner seizures. Bore polishing is also a severe consequence which causes lower cylinder pressures and consequently poor piston ring sealing.

During investigation of cylinder liner failure, very recently the OEM has introduced a cylinder liner with an Anti Polishing Ring (APR). Details of the newly introduced cylinder liner with APR are as follows: The APR is installed in the upper area of the cylinder liner. The purpose of which is to remove possible carbon deposits and carbon residue from the upper portion of a piston, ensuring proper cylinder function, no bore polishing, stable lube oil consumption and low wear of the liner.



Cylinder liner	1
Piston	2
Coke scraping ring	3

Figure 3.18: Cylinder Liner with APR in Deutz TBD 620 engines

Table 3.14: Dimension of APR comparing with other area of cylinder bore

Cylinder No:		Cylinder Liner Bore Dimension (measure from top)				
		APR	35mm	50mm	150mm	250mm
A1 unit	X	**169.68**	170.055	170.050	170.050	170.055
	Y	**169.68**	170.050	170.055	170.055	170.055

As shown in the figure, in the anti-polishing ring solution, the inner diameter of the ring is slightly smaller than the diameter of the other area of cylinder bore. Hence during operation of the engine, carbon accumulated on the top part of the piston can be easily removed and goes out from the engine with the exhaust gas, since clearance in that area is lower than the other area.

4. CONCLUSION

It was revealed that the OEM has not addressed the cooling system and coolant temperature behaviour of the engine, especially during the transient load changes of the engine. During the literature survey it was also found that less attention has been paid to the behaviour of the cooling system of diesel engines, when analysing defects related to the cylinder liner or piston.

During the analysis, it has been proved that the cooling system of Deutz TBD 620 engines fitted on the Fast Attack Craft of the SLN do not behaving properly during transient load changes of the engine due to the slow response of the thermostats. When the cooling system does not responded properly, it affects the functioning of the cylinder liner and piston as the cooling water temperature has a significant influence on the clearance between liner and piston. As transient load changes on-board FACs happen frequently, there is a risk of cylinder liner seizures. This has been identified as the research gap of the thesis.

5. RECOMMENDATION AND IMPLEMENTATION

The main finding of the research is that the fresh water cooling system of the engine is not responding properly, especially during the transient load changes of the engine. Therefore, attention was drawn to enhance the response time of thermostat by introducing a more sensitive arrangement and monitor performance of the engine. Simultaneously, other findings of Surface Roughness of cylinder liner will be undertaken separately in liaison with the OEM. There are various types of thermostats and control systems are available in the market to install as existing thermostat. However, it should able to response quickly and should able to install to the existing layout and control system. Market survey has been carried out and three-way HT valve was selected which is controlled by Pleiger Temperature Controller. (Model: Pleiger Multi-function controller, type –362MC) The valve itself is a mechanical 03 way valve, operated by an electrical signal comes through the controller, Pleiger 362MC.

The controller uses a load signal of the engine which can be the rpm, or charge air pressure or from fuel rack. In this arrangement, the engine rpm is taken as the load signal and also the HT temperature IN and OUT as input to the controller.

The basic operation of the Pleiger 362 MC controller

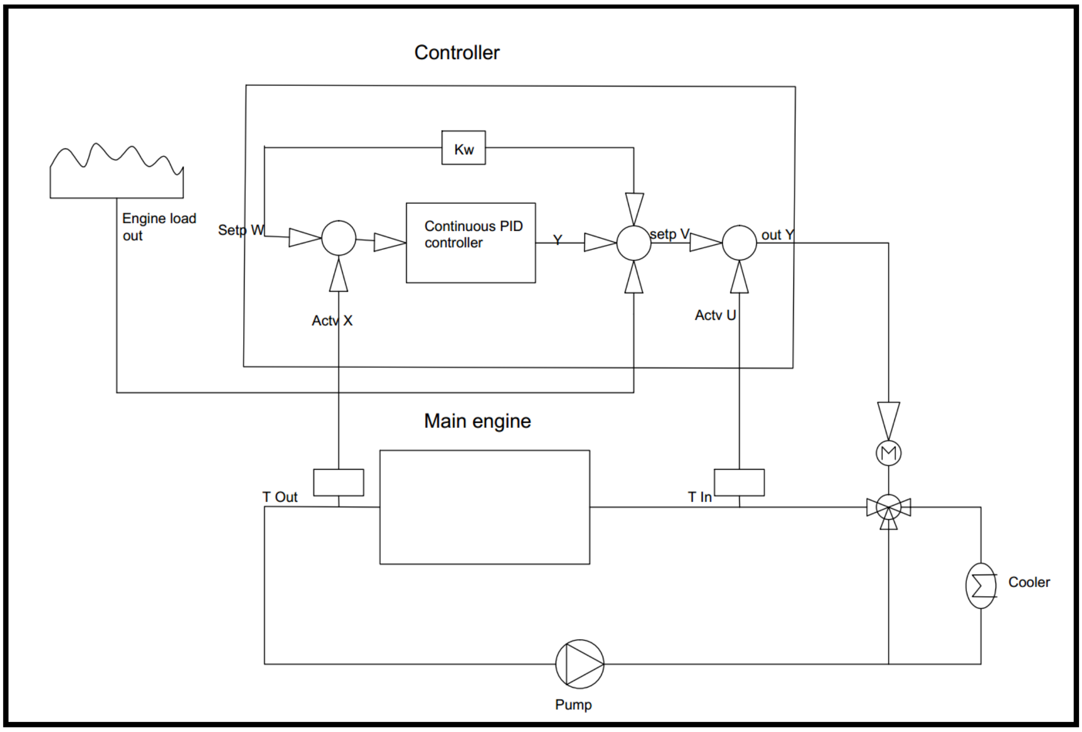


Figure 5.1: Operation of Pleiger controller with 3-way valve

In the diagram, the cooling water temperature IN and the cooling water temperature OUT is on the left and right side of the engine respectively. The three way valve M is the thermostatic valve, actuated by the controller and not by the wax elements. Total on the right, the cooler (heat exchanger) and the coolant pump at the bottom.

The controller is the PID continuous loop controller, with a certain standard set point for the cooling water temperature (W). X is the measured cooling water temperature of the engine out.

The controller calculates a certain temperature Y, which is compared with the actual value engine in (T in). This calculated value is corresponding with a certain position of the valve, which is dividing the flow of cooling water to circulate on the engine, and a certain flow is going over the heat exchanger.

The engine load out, is representing a certain load change on the engine (either higher load, or lower load). This load difference is a signal which is going into the controller, where the controller already pro-actively can change the position of the valve to compensate for the load change. In this way, the effect on the cooling water temperature is already compensated and therefore much more on a constant level.

For example: In case of a load increase of the engine, the temperatures of the combustion will create more heat to be absorbed by the cooling water. In a regular system with wax elements, first the combustion temperature will heat up the cooling water, after that the wax elements start to react (opening) and only after more opening, more flow is generated over the cooler. When the load decreases again, the wax elements first have to sense the lower temperature before they close again. The result will be a much more fluctuating temperature, because the wax elements are always lagging behind.

Function of 3-way valve

The operation of this 3-way valve also has a response time of approximately 20 seconds, the same as the wax element thermostat. However, the response point is much earlier as it is triggered by the load changes of the engine. With the standard wax element, the load change first has to create more heat or less heat in the cooling water before it is sensed by the wax element. Only after sensing by the element, the element will respond by opening and closing. The function of the 3-way valve can be explained as follows:

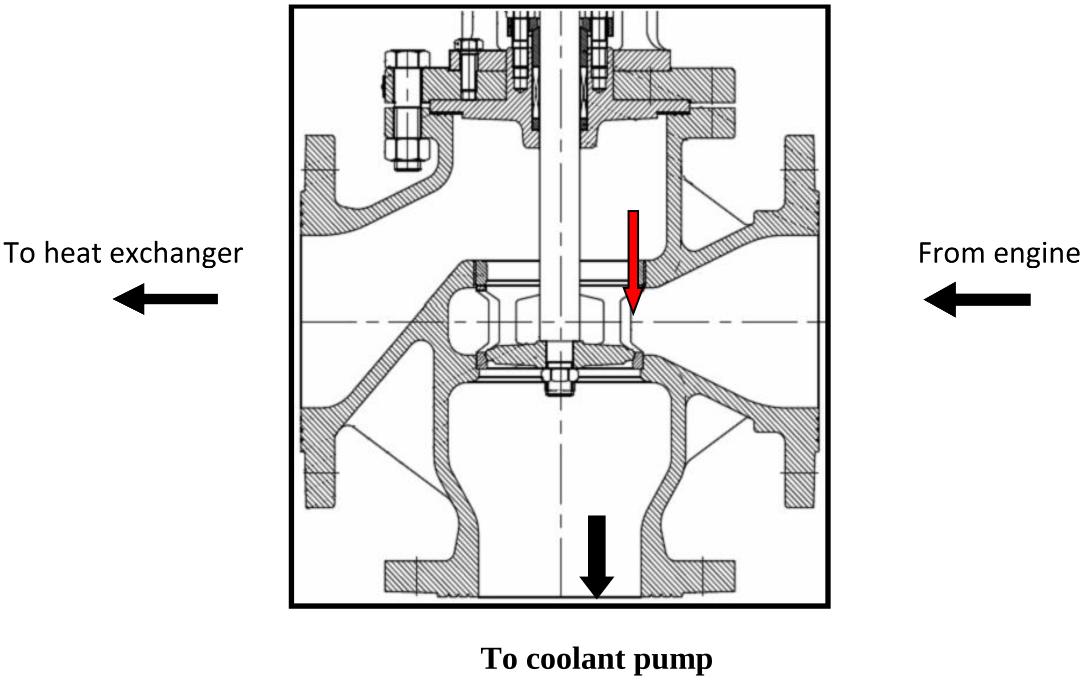


Figure 5.2: Operation of 3-way valve
(hot condition ,valve open position)

The proposed Thermostatic valve can be fitted in to the system with minimum disturbance to the existing system as follows:

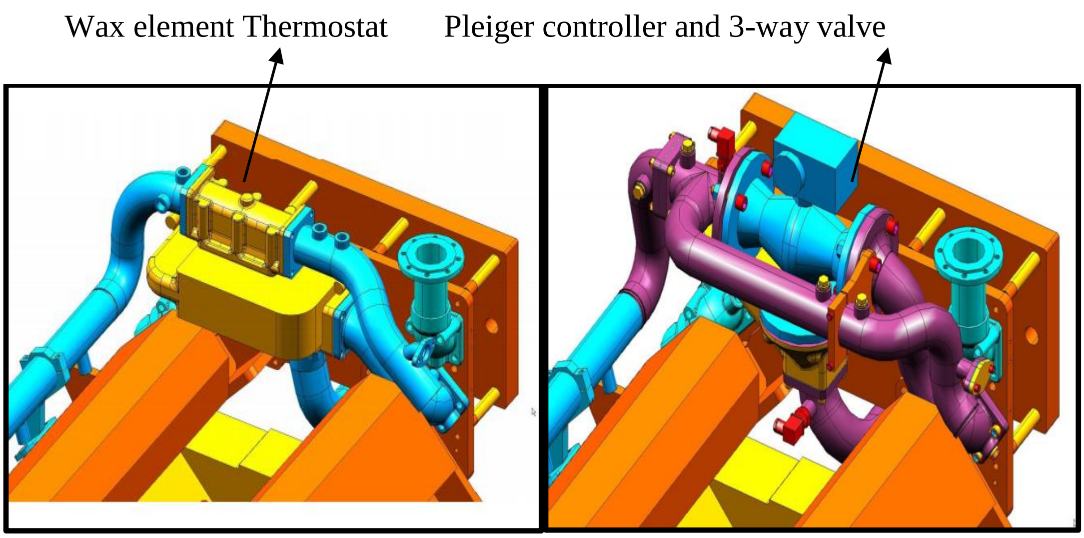


Figure 5.3: Existing (left side) and proposed (right side) arrangement of thermostatic valve

The Pleiger control Valve was obtained after presenting experimental data to the OEM through the local agent. The OEM is also enthusiastic to solve this outstanding problem as early as possible. Hence, it has given its fullest support to undertake this research work. The other general material, ‘O’ rings and gaskets were managed from the SLN spare parts distribution centre. This new thermostatic arrangement was installed in one of the engine utilized to obtain experimental data for better comparison. Photographs obtained during the modification are as follows:

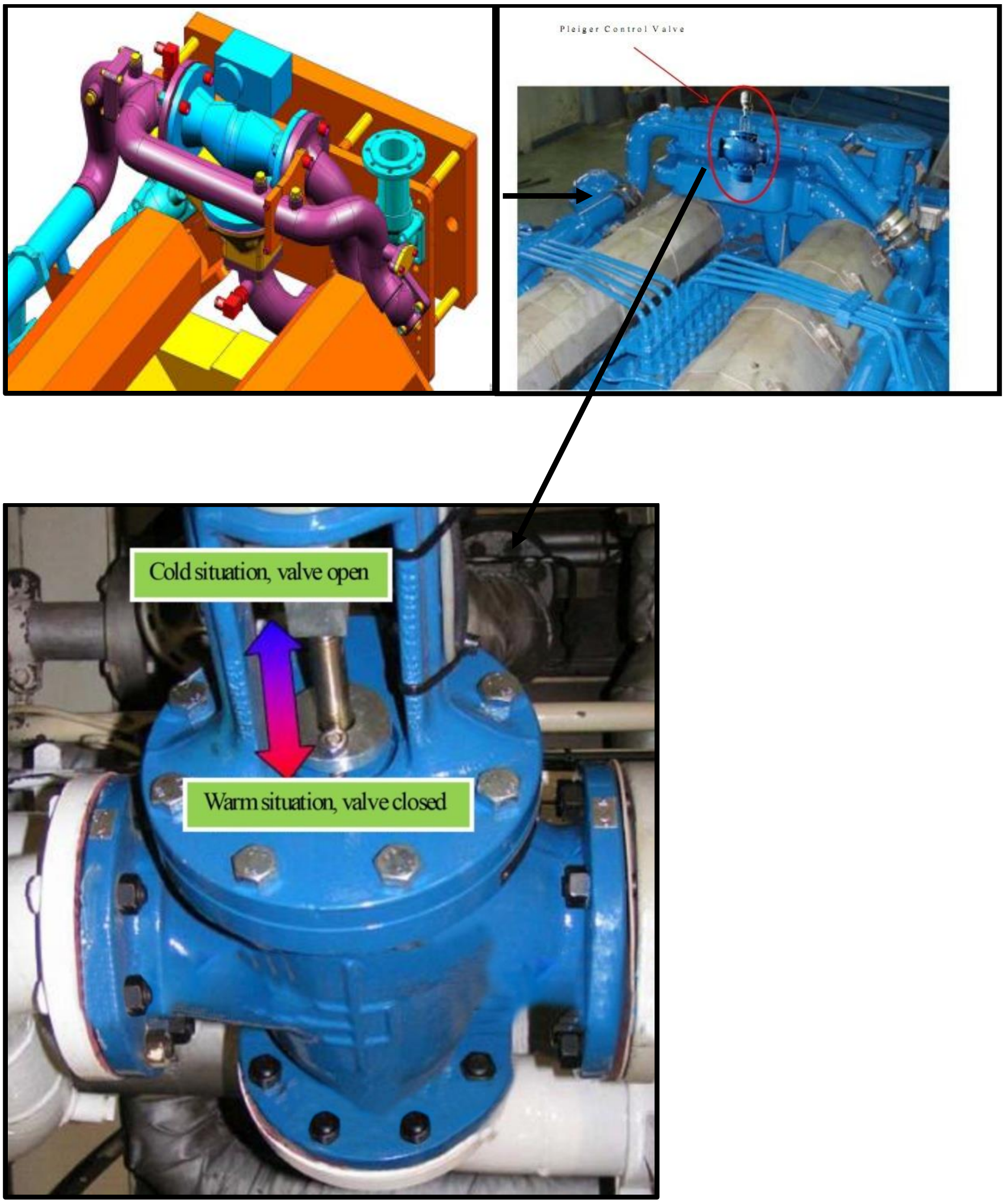


Figure 5.4: way valve with Pleiger controller fitted in the system

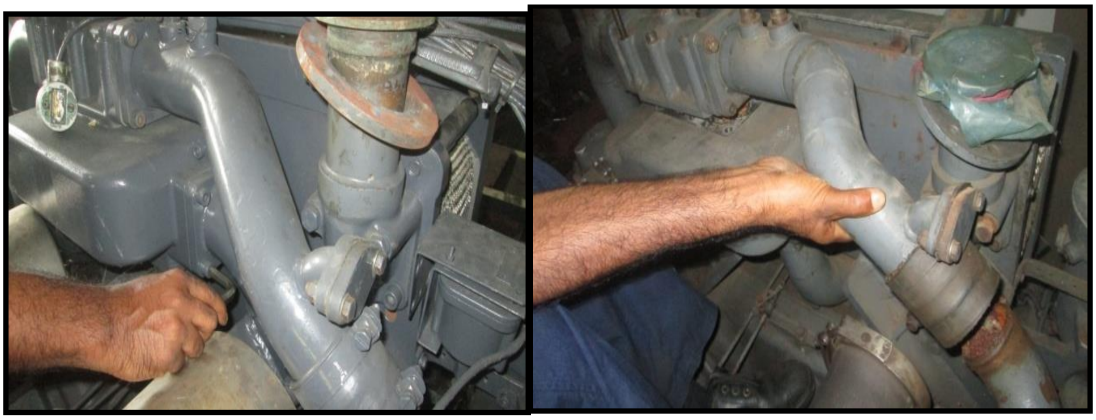


Figure 5.5: Attending modification

This modification was carried out during the first week of October on-board P473 FAC. No major pipe modifications in the system were done; however, a few additional pipe requirements were done with a slight change of the configurations. In the previous system, the cooling water out from the engine was connected to the thermostat from either side as there were 02 Nos. thermostatic elements in the system. However, in the modifications, the cooling water out from both banks of the engine was connected to the three way valve from one side. Pipe diameters to the three-way valve remain the same as 02 Inches. The following drawings of the proposed arrangement were prepared with the assistance of the local agent of the OEM. Basic material requirement was prepared based on the drawing as follows:

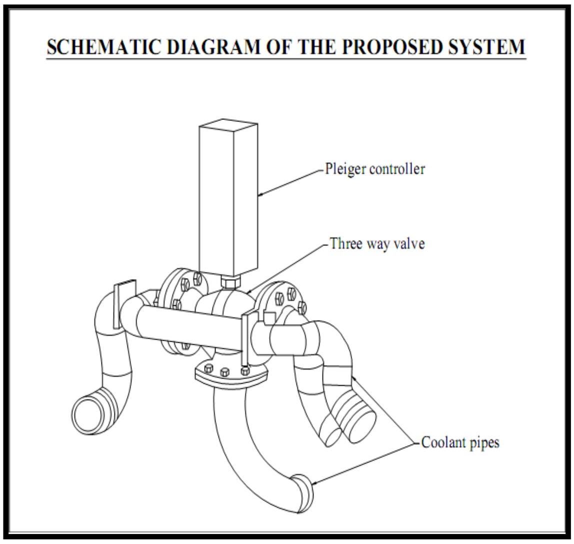


Table5.1:Material Requirement of the proposed system

Description	**Qty**
3-Way Valve with Pleiger Controller	1
Cooling Water Pipe(2 inches size)	6
Suppot	2
Gascket	1
Bracket	1
Gascket	1
Gascket	1
Hexagon screw	06
Hexagon socket screw	30
Hexagon Nut	36
Washer	40
Plug	16
Sealing ring	14
Plug Hex. Head	8
Sealing ring	2
O-ring	3
Flanger	1
Gascket	6
Connection Tube	1
Hose clip	12

The cost of the 3-Way valve with the Pleiger Controller was approximately EURO 14,000 (Approx. Sri Lankan Rs.2.3 Million). As this is for an experimental purpose, a usable valve was obtained from the local agent of the OEM with slight modifications to the flanges. However, this high cost involvement is only for the first valve as it involves design, mould, etc. As per the OEM, the cost will be reduced for the second set to approximately EURO 10,000 (Approx. SL Rs.1.6 Million).The approximate cost for the other materials of the modification was SL Rs.0.4 Million. Hence, the total cost of spares for the modifications was approximatelyRs.2 Million.

After installing on the craft, the cooling water behaviour was checked by installing thermo couples to the cylinder liner. The parameter was recorded by using the data logger. The craft was operated at a steady state condition and load variations. The recorded data is as follows:

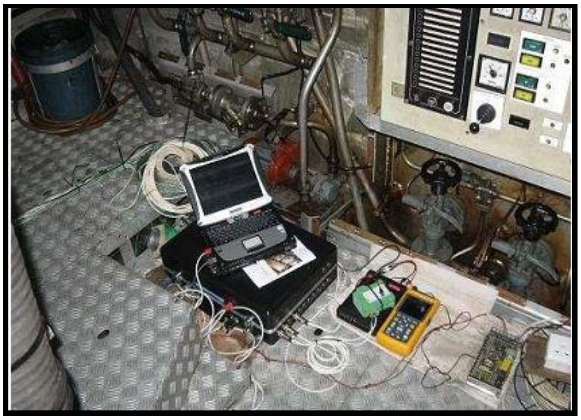


Figure 5.6: Recording final data from Data Logger

Parameters were recorded in several occasions and summary of recorded parameters is as follows:

Running time with 700 rpm : 30 min

Steady state conditions at 700 rpm

HT cooling water out average	: 63 °C
HT cooling water in	: 56 °C
Liner thermocouple 1	: 77 °C
Liner thermocouple 2	: 67 °C
Liner thermocouple 3	: 70 °C

Conditions during acceleration from 700 – 1800 rpm:

Acceleration time 700 – 1800 rpm	: 20 seconds
HT cooling water out average	: 80 °C
HT cooling water in	: 72 °C
Liner thermocouple 1	: 126 °C
Liner thermocouple 2	: 99 °C
Liner thermocouple 3	: 102 °C

Steady state temperatures @ 1800 rpm:

HT cooling water out average	: 81 °C
HT cooling water in	: 72 °C
Liner thermocouple 1	: 128 °C
Liner thermocouple 2	: 95 °C
Liner thermocouple 3	: 102 °C

The recorded data and also the graph below indicated that a constant cooling water temperature remained even with fast power increase or decrease of the engine. When increasing the load from 700 rpm to 1800 rpm within 20 seconds, the cooling water achieved an elevated temperature. This indicates that the thermostat is anticipating at power in or decrease power. Further, the difference between HT cooling water in and out of the engine fell well below 12°C, which is the specified requirement. The comparison of parameters before and after the modification, and during transient load changes is as follows:

Table 5.2 Comparison of Data

Parameter (average)	Before the modification (°C), when increasing 700-1800 rpm, duration 33 seconds	After the modification (°C), when increasing 700-1800 rpm, duration 20 seconds
HT cooling water out	64	**80**
HT cooling water in	56	**72**
Liner thermocouple 1	110	**126**
Liner thermocouple 2	79	**99**
Liner thermocouple 3	80	**102**

The graphical representation of the behaviour of the cooling system, which was obtained from the Data Logger is as follows:

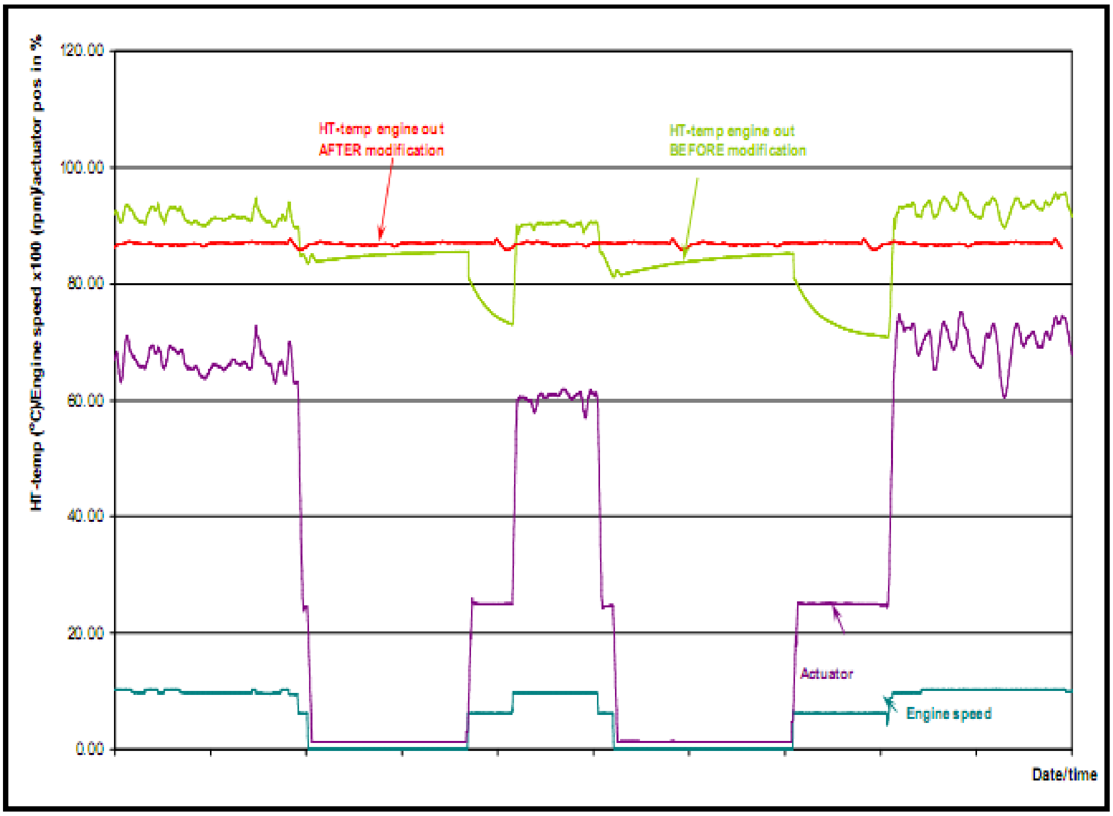


Figure 5.7: Coolant temperature behaviour from Data Logger

The above Graph revealed that the coolant water temperature remain almost constant after the modification to the thermostat. There is no deviation of the rest of the parameters of the engine after the modification. A detail of sea trial report is attached as an Appendix to this report. After the modification, the engine has operated approximately 400 running hours at various rpm ranges and load variations without any problems. The above comparison revealed the following and these are prime requirements of a high speed diesel engine:

a. During transient load changes, the cooling water temperature compensates very quickly and also the cylinder liners achieved the required temperature to maintain sufficient clearance between cylinder liner and piston
b. The difference between HT cooling water in and HT cooling water out is within the specified limit which is also a prime factor to maintain the required clearance.
c. The maximum HT cooling water temperature achieved approximately 80 °C and it is in the required limit, which is also a sensitive factor.

Hence, the modification can satisfactorily overcome the research gap. However, in order to evaluate the complete results, the engine needs to operate at least 2000 operating hours under various load variations.

6. FUTURE WORK

This research identified the following other possible reasons for the cylinder liner seizure of the Deutz TBD 620 engines operating in the SLN.

a. Surface Texture of Cylinder Liner

b. Low Load running of engines

As per the OEM, they have addressed these problems, for which however theSLN needs to pay attention by studying in-depth in liaison with the OEM, or this can be continued as another research as well. In addition, the SLN needs to educate all the operators of the craft regarding the way they can contribute to overcome this problem by operating the craft at optimum level as far as possible.

The Metallurgical composition is also a very crucial factor which can contribute to cylinder liner failures. This research is based on the facts indicated by the OEM regarding the Metallurgical aspects. However, further studies can be continued on this.

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