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Engine Performance based on Condition Monitoring Using Ship Shore Data Connections

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Contents

Abstract

The present paper focus is on diesel engines performance monitoring as these engines are the shipping working horses. The deficient operation of a diesel engine may be detected and corrected to minimise emissions, consumption and its operating cost and maximise engine availability.

This paper establishes a methodology of engine systems efficiency, and how they may be monitored via ship shore data link using BOEM-S (cloud-based software). The paper concentrates on thermal loading of diesel engines, control of components and on some details on the subject of combustion control. Finally, we will illustrate some of the results by examples.

Keywords: Diesel engine performance monitoring; emissions; operating costs; cloud-based software.

Introduction

In the shipping industry, maintenance underwent some evolution, from curative to programmed or scheduled maintenance which is often performed too early or too late, resulting either in high costs due to unnecessary replacements or functional failures, affecting ship performance and availability. Worse is a failure of components like a steam turbine, or a gearbox, that cannot be easily inspected is often disassembled for inspection on schedule, therefore generating the risk of introducing faults due to the inspection or re-assembly, resulting into failures. Infant mortality is therefore introduced into the components in mid-life due to the scheduled inspections.

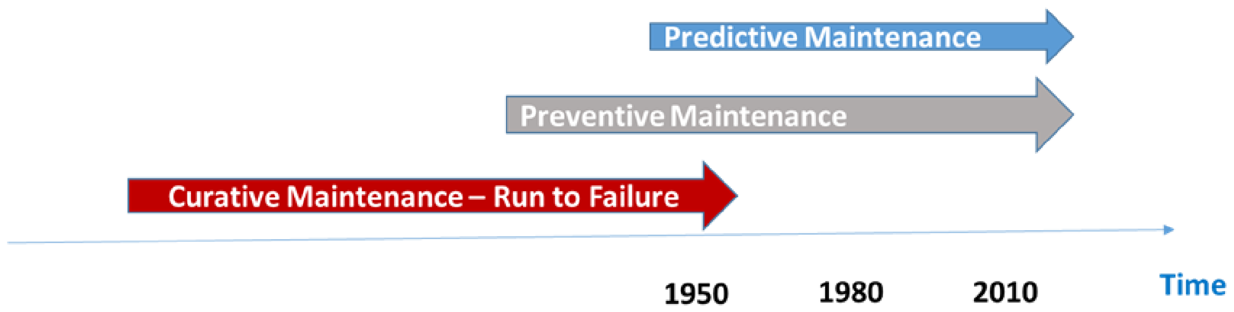

Figure 1 - Evolution of maintenance techniques.

Shipping and ships are becoming more sophisticated and more demanding. Power, fuel consumption, emissions and plant availability are operational variables, of utmost importance for the ship owners and operators of today. As a result, prediction of performance is fundamental to take the appropriate corrective actions on time, before costly failures, and off optimised operation takes place.

Along with those demands, ships are generating more and more data, becoming virtually impossible for a human to effectively monitor, digest and conclude about plant performance and adequate corrective actions. In fact, that's all about the operation at its highest efficiency as possible. The implementation of a condition based monitoring system can lead to more accurate and timely maintenance tasks, with all its benefits, like improved safety and availability of ship systems of lower costs of maintenance.

On the other hand, there is a clear tendency today that ship-owners and operators are more interested in complete measurements of machinery plant. One of the main reasons for this is not so much the fuel oil price but the efficiency and the environment stricter rules. This results in more interrelated machinery systems designed for utilization of waste energy and operating in more demanding conditions. Control of consumption and capacity is, therefore, more important.

Typically indicated power, pressures and temperatures on the main engine are measured only during trial trips. In combination with our research work, we have made measurements also on critical auxiliary systems in the engine room. Those measurements have included electrical consumption, pressures temperatures and flows, etc.

Many of the results indicate clearly deviations between design figures and results really achieved in service. From the point of view of energy consumption and reliability of plants, this is important, and we believe on more complete measurements also on auxiliary systems need to be made, as many deviations encountered are due to as-built deviations, but some may be intentional.

Heat exchangers capacity and their circulation pumps may be one such case, where higher capacity pumps are used to counterbalance heat exchangers of reduced heat transfer areas, therefore, generating excessive fuel consumption and emissions during all ship life, as these small heat transfer areas are compensated by larger flow rates.

The focus of the paper will be the diesel engine. The diesel engine will be analysed in some of their most important subsystems, namely, Air system, and its components, and fuel system and its components. The paper will concentrate on thermal loading of deficient diesel engine subsystems and its consequences, finalising with some examples.

Diesel engines are the shipping industry working horses, therefore performance monitoring is of utmost interest as a toll for availability, reliability, operational cost and environmental impact of their operation. The deficient operation of a diesel engine may be detected and corrected as to minimise its consumption but also its operating cost. The present paper establishes a methodology of engine systems efficiency or performance monitoring, and how they may be monitored via ship shore data link.

The use of BOEM-S as a Ship Performance and Condition Control Tool

To have a condition based ship performance tool, it is required to have a perfect knowledge of what, when and how to measure, but also how to filter and interpret automatically the data collected, as ship operation modes and number of data points may be enormous a costly, and operational environment may vary suddenly from fair to rough weather.

Ship performance is intimately related to ship systems performance, therefore it is of utmost importance to know the systems that affect the ship performance aspects under study.

From the point of view of fuel consumption, main engines performance has a major impact. Most of the Main Engine deficiencies show up as increased specific fuel oil consumption and abnormal exhaust gases temperatures, indicating a defective operation of air and or fuel systems comprising turbochargers and respective air coolers, filters but also injection systems, comprising fuel injection pumps, injector nozzles, etc.

For a monitoring system economically and technically viable, the key performance variables to monitor the condition of such systems, their acquisition frequency, and filtering must be properly defined and set.

As an example, let us observe a methodology of Air System Condition Assessment.

Air System. Flow Condition Monitoring.

The main engine is normally operating under very variable conditions. The draft, the increased resistance of the hull, climatic conditions, wind and sea, influence the readings. It is, therefore, necessary to develop parameters which give a definite criterion for the engine condition assessment, and which are independent of all external conditions.

Figure 2 represents a simplified schematic of a diesel engine air system. When the air has passed the compressor and the pressure has increased to from Pa to P1, the air enters the air cooler and air receiver, cylinder, turbine and exhaust boiler and finally it is expelled to the air. The pressure P1 is the driving pressure that causes the air to flow through the respective canals and restrictions.

∆pf	Pressure drop filter	Pc	Compressor discharge pressure
∆pc	Pressure drop air cooler	Pg	Exhaust gases pressure before turbine
Pa	Atmospheric pressure	Tg	Exhaust gases temperature before turbine
Ta	Atmospheric temperature	Te	Exhaust gases temperature after turbine
Ps	Scavenging pressure	Pe	Exhaust gases temperature after turbine
Ts	Scavenging Temperature		

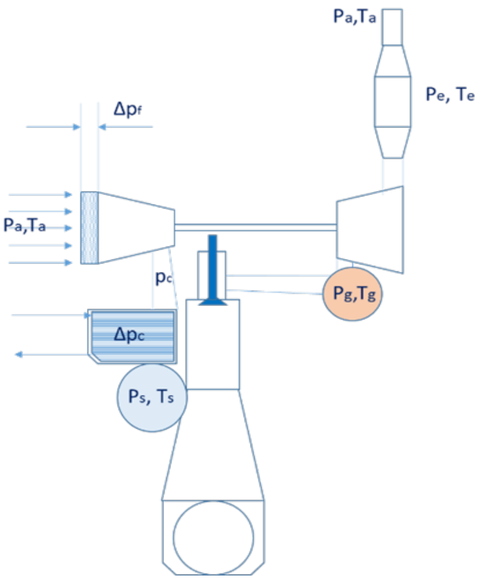

Figure 2 - Schematic of some control variables for condition monitoring.

Assuming, for example, the air quantity is the same and the pressure P1 increases, the explanation must be an extra restriction and or an increase fouling in the air system. The mass of air in the cylinders when the compression begins can be made proportional to Ps/Ts, by assuming a constant scavenging efficiency. If we further assume that the fuel injected per stroke is constant, that means same mean pressure, the combustion excess air is also proportional to the same quotient.

Here we clearly can see the importance of minimising the scavenging temperatures and increasing the pressure as this gives the maximum excess air and hereby the lowest process temperatures during combustion. One of the main function, if not the main function of air flow through the engine is the cooling of the combustion chamber components.

Control of the amount of air is important, direct measurement of airflow on board is for practical reasons impossible due to the complicated piping and required measuring equipment. Therefore, it is of importance to use an alternative method to measure indirectly an air mass flow rate. A factor “A”, which is proportional to the true air mass flow rate may be used with advantage.

The “A” factor, may be calculated using the engine test bed measurements data (reference condition), based on the fuel consumption and exhaust gases temperatures as a function of load.

The basis for this calculation is that the exhaust heat loss of an engine is practically constant when the load is constant, even if the airflow is reduced. The calculation of the air factor, "A" is therefore done very easily and based on simple measurements on board. Using the same notations as in Figure 2, a diesel engine exhaust gas heat loss can be written as:

$$P_{exh}(kW) = \dot{m} \times C_p \times (t_e - t_0) = \alpha \times \dot{m}_f \times H_u$$

where:

$\dot{m}$ = Air flow rate (kg/s);

C_p = Specific heat of exhaust gases (kJ/kgK);

$\dot{m}_f$ = Fuel oil supply to the engine (kg/s);

H_u = Net Calorific Value of the fuel (kJ/kg)

α = Exhaust gas heat loss as part of the heat supplied to the engine by the fuel (kJ)

Assuming that C_p and α are constant and $\dot{m}_f \propto A$ it can be concluded that air factor is given by:

$$A = \frac{\dot{m}_f \times H_u}{(t_e - t_o)}$$

Shows that the percentage of heat loss through the exhaust gas flow remains practically constant even if the reduction in turbocharger efficiency due to fouling or damage occurs. The measurements are valid for a constant pressure supercharged crosshead engine. The abscissa axis 100% corresponds to 100% loading on the engine, and all the calculations are done relative to this.

A closer inspection of Figure 3, shows that the assumption of a constant heat loss through the exhaust gases could be used with a good approximation both regarding the influence from the load and from the turbocharger efficiency.

It seems therefore that the air factor "A" is adequate to monitor the air mass flow based on the constant exhaust losses. By using the relative air factor "A", it is thus not necessary to know exactly the percentage exhaust gas heat loss, as "A" is calculated for the whole load range based on to the reference measurements from the test bed.

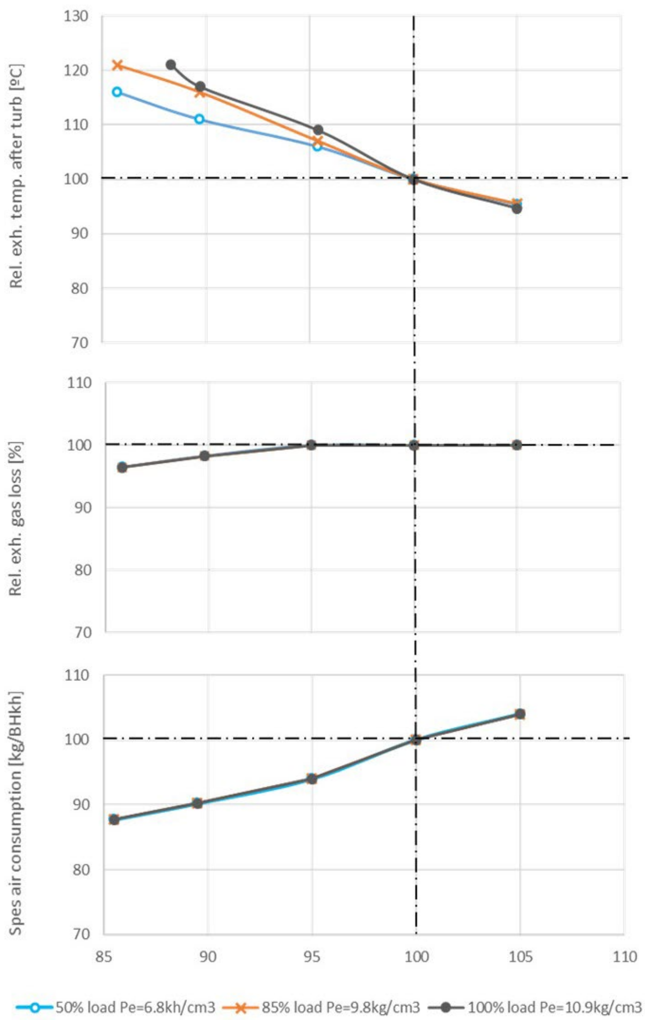

Figure 3 - Correlation between SFOC, Relative exhaust gas temperature after turbocharger, Relative exhaust gas heat losses and specific air consumption, as a function of Relative Turbocharger Efficiency. (on the right)

To conclude the above discussion one can say that the indirect determination of the air flow based on the proposed method gives the following control possibilities:

Pressure Drop

The total pressure drop through the engine and exhaust system is the result of general engine fouling. Compare the reference curve shown in Figure 4.

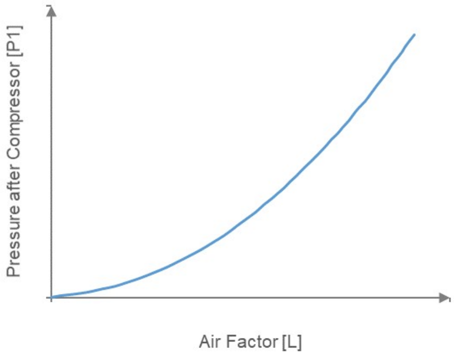

Figure 4 - Compressor discharge pressure as a function of air factor "A" (on the right).

The fouling of the air filter

The added resistance to air flow passage through the filter can be correctly monitored using a very cheap pressure gage as in Figure 5 when one uses a diagram giving the normal correlation between the pressure drop ΔP_1 and the air factor "A".

It should be mentioned here that in practice there are very different opinions regarding the pressure drop limit through the air filter. To discuss this problem, in Figure 6 are illustrated the results from measurements carried out on a medium speed engine under full load.

As per Figure 6, below, it is remarkable that if the normal pressure drop of 150 mmWC increases to 400 mmWC, the air quantity through the engine decreases only about 2%, and the exhaust gas temperatures increase only 30 ºC.

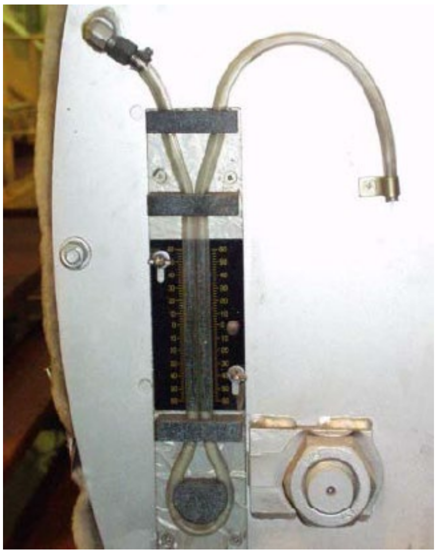

Figure 5 - The aspect of the pressure drop gage across the turbocharger air filter.

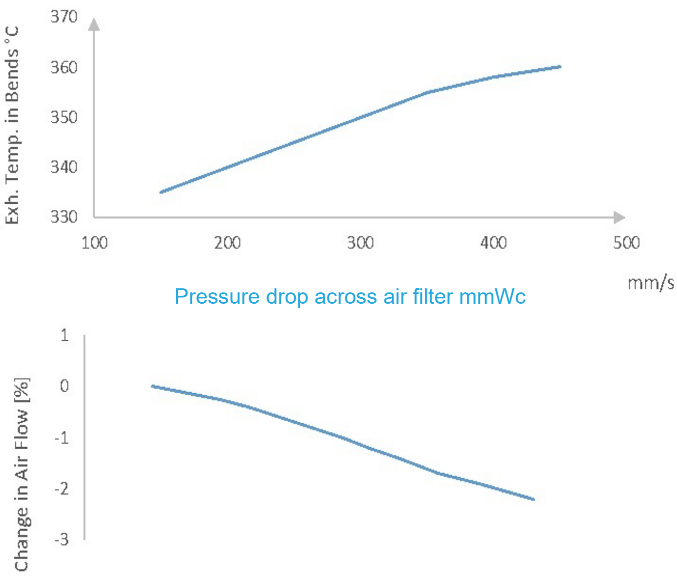

Figure 6 - Air flow rate and Exhaust gas temperature variations as a function of Pressure drop across turbocharger air filter ΔP_1.

Air cooler monitoring

The pressure drop across the air cooler and its influence on air factor “A” must be produced. Pressure drop in relation to the increase on to the air factor “A”. Compared to the total scavenging pressure, an increase of the pressure drop ΔP_2 in the air cooler is not of minor importance.

To estimate the next cleaning operation, it is, therefore, necessary to control the scavenging temperature Ts in relation to the increase of ΔP_2, but also, the identification of the side of the air cooler that is in need of being cleaned, namely air side or water side. Figure 7, below shows the local instrumentation for the monitoring of Pressure drop across air cooler and Scavenging air and exhaust receiver pressure.

Scavenging air and exhaust receiver pressures.

Pressure drop over Scavenge Air Cooler ΔP_{Cooler}

Figure 7 - Gauges for Pressure drop across air cooler and Scavenging air and exhaust receiver pressure measurements.

Exhaust gas piping back pressure

The monitoring of the exhaust piping back pressure, is of utmost importance, as it affects directly the turbocharger speed, and therefore the Scavenging pressure, Specific Fuel Oil Consumption and engine thermal load. To monitor this pressure drop, a plot relating the Pressure drop over the exhaust system should be arranged, with partial measurements for the boiler, silencer and spark arrester as a function of the air factor. The pressure drop is critical because of increase in the turbochargers back pressure and consequently its speed reduction and the corresponding reduction of the scavenging air flow.

Engine Thermal Loading

The mean temperature of the gases inside one engine cylinder varies from engine to engine and with load, but in average and at full load varies between 500 ºC and 800ºC.

At such high average temperatures, the heat removal must be effective, if not, lubricating oil would be burnt quickly, and the components clearances between the friction parts would become zero, and the parts would wear down rapidly, the piston crowns, cylinder heads and valves would be melted. Thus, to keep the parts under permissible temperature levels, heat removal must be carried in a very efficient way. Cooling is achieved in a great extent through the induced air, but some components need to be individually cooled, such as piston crowns, liners and cylinder heads. Forced cooling can be achieved by means of water circulation or through air circulation in an open or closed configuration of the cooling system.

Heat produced by the friction of the piston is transferred through the cylinder liner to the cooling water; heat originated by friction on the bearings is carried away by the lubricating oil. The combustion heat is transferred through the cylinder liner and cylinder head to the circulating water. Only in this way, the temperature of the parts is maintained in fairly acceptable values.

Forms of heat transfer in an engine

In an engine cylinder, it can be found the three different ways of heat transfer, namely Conduction, Convection and Radiation. In studying and calculating the heat transfer in internal combustion engines, the three main laws must be considered, namely: Fourier 's law of conduction; Newton's law of cooling and Stefan-Boltzmann law of radiation.

Heat Transfer Coefficients

The amount of heat rejected per time unit by the combustion gases to the cylinder liner wall, cylinder head and piston crown by contact and radiation can be expressed by the following expression:

$$Q_g = Q_r + Q_c = \left[\alpha_r\left(T_g - T_w\right) + \alpha_c\left(T_g - T_w\right)\right] \times A \quad \frac{kJ}{s}$$

$$Q_g = \alpha_g \times \left(T_g - T_w\right) \times A \quad \frac{kJ}{s}$$

where:

$\alpha_g = \alpha_c + \alpha_r$ = Coefficient of heat transfer from the combustion gases to the combustion chamber walls by contact and radiation;

α_c Coefficient of heat transfer by contact or conduction;

α_r Coefficient of heat transfer by radiation;

T_g, T_w temperature of the gases and of the walls in contact with the combustion gases.

The determination of the coefficients of heat transmission for each mode (conduction, convection and radiation), is more scientific art, and will be not addressed in this paper).

Keeping the practical focus, it is important to monitor what are parameters we have to use as a total guidance for the operation of the engines.

From experience with large supercharged crosshead engines, it is known that characteristic metal temperatures in the cylinder liners are a relevant criterion for the engine components thermal loading according to the above introduction.

For a four-cycle medium speed diesel engine, the problem is quite different because the cylinder liner and cover temperatures normally are not subjected to excessively high heat stresses. However, it is a general experience with medium speed diesel engines than run on heavy fuel oil, that high exhaust temperatures and respective thermal loading can result into exhaust gas valves damage and frequent overhauling of the cylinders.

It is known that when the exhaust valve seat temperature exceeds a certain critical temperature limit a strong high-temperature corrosion attack should be expected dependent on the sodium and vanadium content of the heavy fuel oil in use.

This high-temperature corrosion develops fast and it is due to the decreased eutectic temperature of the alloy material formed with the valve material and its seat together with vanil-vanadates of sodium from the heavy fuel oil. Sodium acts as a paste (flux) for vanadium slag. When unfavourable quantities of vanadium and sodium are present in a fuel they react at combustion temperatures to form (eutectic) compounds with ash melting points within operating temperatures.

In molten form sodium/vanadium ash can corrode alloy steels, and when this condition is allowed to persist unchecked, high-temperature corrosion, overheating, and eventual burning away of exhaust valves, valve faces, and piston crowns is not uncommon. This sodium/vanadium ratio and its relationship to ash melting temperature are shown in Figure 8.

The measurement of valve seat temperatures virtually impossible under normal running conditions, and as previously for the air system we have to develop some indirect methods to determine the heat loading of the exhaust valves. In Figure 9 is shown some basic investigations used to find a thermal load parameter.

$$P_e^a \left(\frac{T_S}{P_S}\right)^b \Delta T_e^c$$

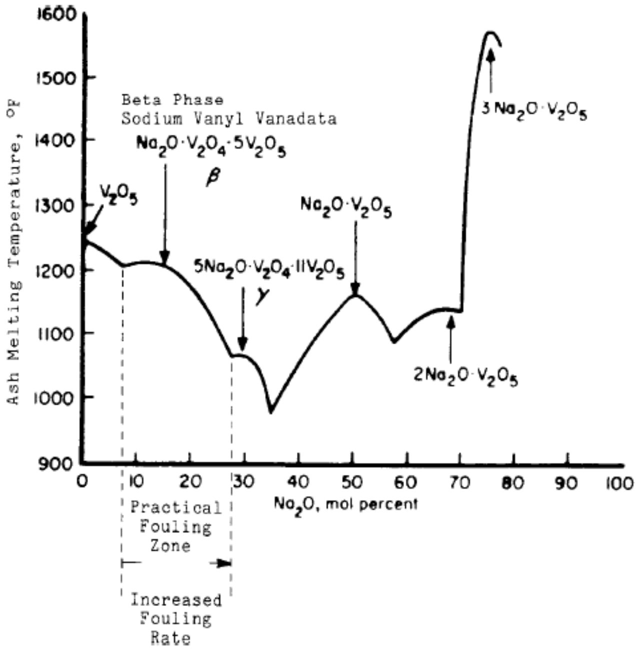

Figure 8 - Effect on eutectic temperature of Na_2O_3, and other salts. (Reference, ABS Notes on HFO-2008)

The diagrams of Figure 9, illustrates some test results where the engine primarily is run clean and under normal conditions serving as a reference. The two upper diagrams show the scavenging pressure and corresponding exhaust temperatures which are measured as a function of the mean effective pressure or engine load.

Tests:

Reduced area of turbine nozzles (nozzle ring)

This causes an increase in the scavenging pressure and we further find a tendency to decrease the exhaust gas temperatures. Reduction in turbine inlet area is very often experienced in practice because of fouling, and in the monitoring diagrams, this will be shown as an increase of the flow resistance for the same air quantity. This can also in some cases cause a higher scavenging pressure.

The decrease in exhaust gas temperatures, in this case, is partially due to the increase in the excess air in the cylinders, due to turbocharger higher speed.

Running with a reduced area of the turbo-compressor nozzle ring (diffuser).

This causes a lower scavenging pressure. The situation is similar to a damaged or heavily fouled compressor. As can be seen in the diagram of Figure 9, this causes an increase in exhaust temperatures after the cylinders.

The tests shown represent on both cases limits very often encountered in practice on medium speed diesel engines. On the last diagram the valve temperature is correlated with the following parameter:

$$P_e^a \times \left(\frac{T_S}{P_S}\right)^b \times \Delta T_e^c$$

This parameter is used as abscissa axis arranging all the measured values on a straight line by a proper choice of the exponents a, b, c.

where:

P_e = mean effective pressure

T_s = Tscav = scavenging air temperature

P_s = Pscav = scavenging air temperature

ΔT_s = difference between the exhaust gas temperature and the engine room air temperature (assuming the engine air intake is from the engine room)

Due to the influence from loading, damages and fouling on ships hull and machinery, these temperatures and pressures will in service vary independently, and thus cause different influences on exhaust valve seat temperatures.

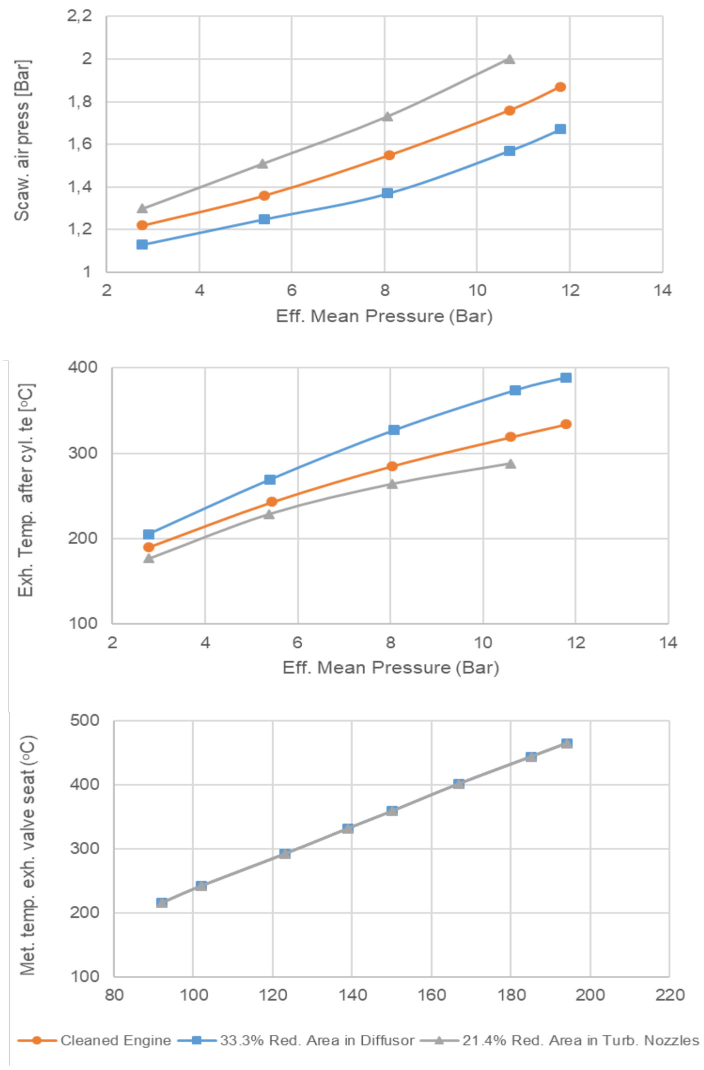

Figure 9 - Test results of Reduced Nozzle Area, Reduced area of the turbo-compressor nozzle ring (diffuser) and Metal temperature of exhaust valve as a function of the parameter $P_e^a \times \left(\frac{T_S}{P_S}\right)^b \times \Delta T_e^c$

No matter the combinations while in operation, the acceptance of this generic parameter will give an indication of the expected valve seat temperature variations.

Normally the exact metal temperature for the engine in question is not known, and Professor Knut Langseth has proposed to use the parameter calculated from 100% loading on testbed as a reference. All philosophy lying behind is that when an engine manufacturer has accepted his design for this reference loading, one must expect that the engine in service could be loaded at least up to this value without damage, **i.e.**, the calculated parameter corresponds to a nominal condition of engine operation.

It will be shown how this parameter can vary about the nominal value, depending on the engine condition and its load.

There are no restrictions to run with values exceeding 100% for this temperature parameter. The most important is to note the value of the parameter when the valve starts to burn. 1% increase in valve loading parameter corresponds to about 5ºC in the metal temperature.

In the combined parameter P_e must be understood as an independent variable or engine load, while the other three variables should be regarded as generic variables related to the engine condition. If the air flow is reduced, the ΔT_e will increase. The variables P_s and T_s are dependent on the engine load. Some engines may have the parameter well above 100%, in particular when using special valve seat materials like Nimonic.

Engine

Fuel pumps

The amount of fuel delivered by the fuel pumps can be expressed by the following expression:

$$V_t = C_1 \times f \times n \times \eta_v \text{ (Litres/hour)}$$

where:

C_1 = constant
f = fuel rack (pump index)
n = revolution per minute
η_v = volumetric efficiency of the pump

One such diagram as illustrated in Figure 10, with the fuel oil consumption in [litres/hour] of fuel oil as a function of the [fuel injection pumps index x the revolutions], it can be used as an excellent reference to monitor the fuel pumps volumetric index.

All measurements in service which correspond to the measured reference from the testbed indicate that the injection pumps have an unaltered volumetric efficiency and therefore no wear.

Another important question is whether all combinations of fuel pump index and revolutions fall about the reference line from the test bed. This is particularly important when the engine in service is driving a Controllable Pitch Propeller. As shown in Figure 10, all measured points from ½ pitch and above are spread within very narrow limits about the reference line, while, however, 0 pitch evidently gives such a small pump stroke that it is impossible to avoid a certain pump bypass.

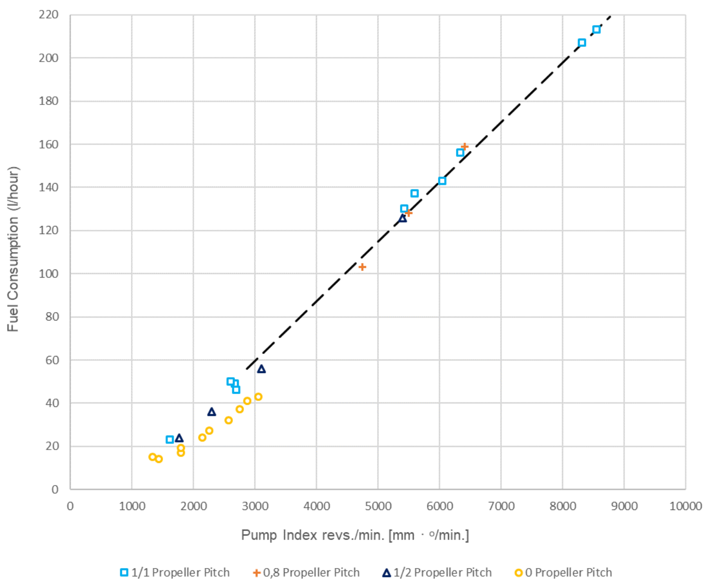

Figure 10 - Fuel delivered as a function of Fuel pump index and engine speed, for a controllable pitch propeller propulsion plant.

It is sufficient for the purpose to state that for all practically used pitch ratios a monitoring diagram like the one in Figure 10 is adequate for monitoring the fuel pumps condition, which may be improved by an accurate fuel oil consumption measurement, for example using a Coriolis type of flow meter with automatic density - temperature corrections, installed at the engine fuel oil “loop”, so that a second mass flow meter may be avoided.

Nowadays, the task of making a fuel oil mass balance may not be easy, unless provision of adequately located flow meters is specified at an early stage of the fuel system design, in particular on board vessel with “uni-fuel” concept, or on board the ones that do the fuel return to the daily tanks.

Indirect Measurement of Power

The fuel pump equation, as shown above, can easily be transformed to the following equation for the mean effective pressure as:

$$P_e = C_2 \times f \times \eta_t \times \eta_v \text{ [kg/cm}^2\text{]}$$

where:

C_2= constant
η_t = engine’s thermal efficiency
f = fuel rack (pump index)
η_v = volumetric efficiency of the pump

Representing graphically the relationship between Fuel Rack (or Fuel pump index) as a function of Mean effective pressure or load, a diagram like the one of Figure 11 may be plotted and serve to monitor fuel pumps conditions, but also hull fouling. Where the pump index is plotted against the mean effective pressure, will give a normal relation between pump index and mean effective pressure from test bed.

In this relation, the relationship between that two variables is incorporated, through the product of the two efficiencies η_t and η_v. This means that if one fuel pump index measurement is performed, it can be first plotted on the graph of Figure 10. If the result here is in accordance with the reference curve, the mean effective pressure P_e, can be determined with a good approximation from the graph of Figure 11, by inserting the same fuel pump index. If by the contrary, the fuel pumps are worn and thus give a service point on graph of Figure 10, which falls below the reference curve, a pump index correction is needed to counteract the fuel pump wear, and in this way the graph of Figure 11 is still useful to determine the mean effective pressure Pe.

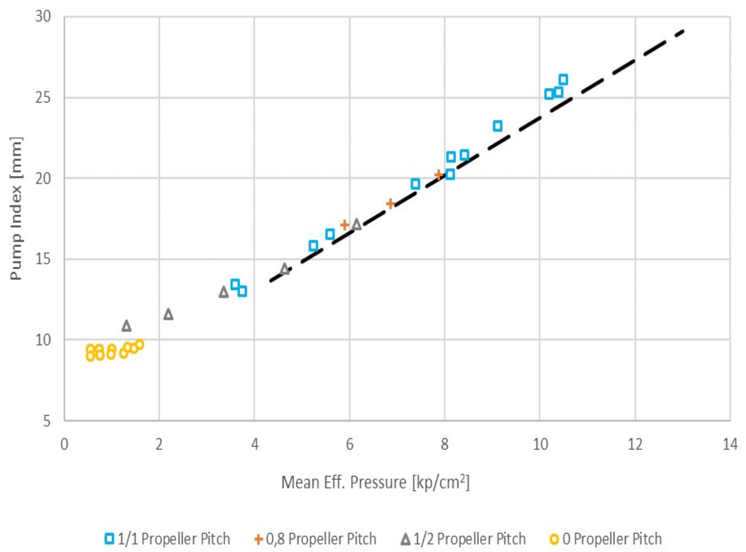

Figure 11 - Average fuel injection pump rack as a function of Mean effective pressure, for different propeller pitches.

The points plotted in Figure 10 and Figure 11 are measurements from the same engine that had its break power [BHP] measured. As it can be seen, in Figure 10, the values fall on top of the line, but on Figure 11, it can be observed that the measured value are all of them slightly above the reference curve by about 5%, indicating that there is a corresponding reduction in the engine thermal efficiency, **i.e.**, there is an increase in the SFOC of about 5%.

One other way, of measuring power is to determine Pe from de indicator diagrams, as:

$$P(kW) = P_e \times L \times A \times N$$

where:

P_e = Mean effective pressure (kPa);
L = Stroke (m);
A =Area of cylinder (m^2)
N = engine speed (rad/s)

Turbocharger

The most practical way is to control the compressor and the turbine separately. Without going into detailed theory, it is possible to evaluate the performance of the compressor side, by plotting its compression ratio against its speed of rotation. As a reference for this characteristic, the engine test bed data should be used, and an in-service data point below the reference line indicates a damage or fouling of the compressor.

The data points plotted in Figure 12, show how an overhaul has given an improved performance of the compressor.

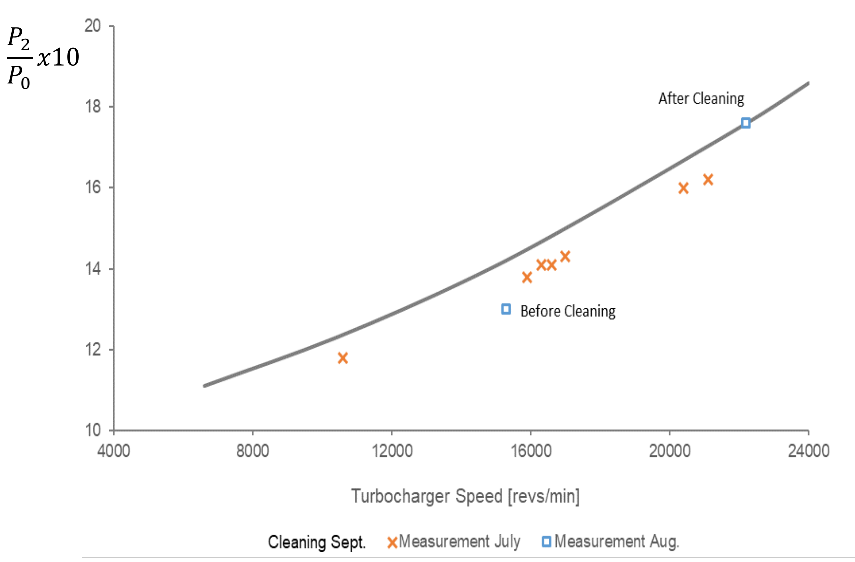

Figure 12 - The relationship between turbocharger speed and compression ratio. The figure illustrates how to monitor the turbocharger efficiency and how it will be noticed the improvement of performance after cleaning.

When this graph is to be prepared, it is usual to correct its speed of rotation, for the expected mean engine room temperature, so that the diagram may be used directly on board.

To perform a similar control to the turbine side, of the turbocharger, a simplified diagram to the one of Figure 12 may be used.

The most accepted and usable method seems to be a diagram where the corrected revolutions of the turbocharger are plotted as a function of the heat supplied to the engine.

As long as the SFOC [g/kWh] does not deviate too much from the testbed results, we can roughly calculate the heat supplied by the exhaust gases to the turbine, which should be proportional to the fuel consumption in kg/second, multiplied by the fuel Net Calorific Value [kJ/kg].

A useful way of power estimate above 40% typically, is through the use of a similar diagram to that of Figure 12, but where engine power is plotted against turbocharger RPM, this in reference to the engine shop trial data.

Combustion Control

Pressure Control

Figure 13 shows a diesel process with compression from the scavenging pressure Ps to the compression pressure Pcomp which differs from the theoretical cycle due to the following deviations:

- No loss to cooling water neither during compression nor during expansion;
- Instantaneous pressure rise (at constant volume) from Pcomp to Pmax, where Q1 is the energy added per cycle to achieve such pressure rise;
- Combustion proceeding at constant pressure equal to Pmax. Energy added per cycle is equal to Q2;
- The combustion gases are thought to be removed instantaneously at BDC, without any flow loss through valves and ports removing from the cylinder an energy Q3.

The energy balance thus becomes theoretically made available for mechanical work becomes:

$$(Q1+Q2)-Q3 = 0$$

If it is assumed that the fuel injection pumps maintain a certain constant index representing and heat equivalent to (Q1+Q2), it is evident that Q3 must be minimised to increase the thermal efficiency.

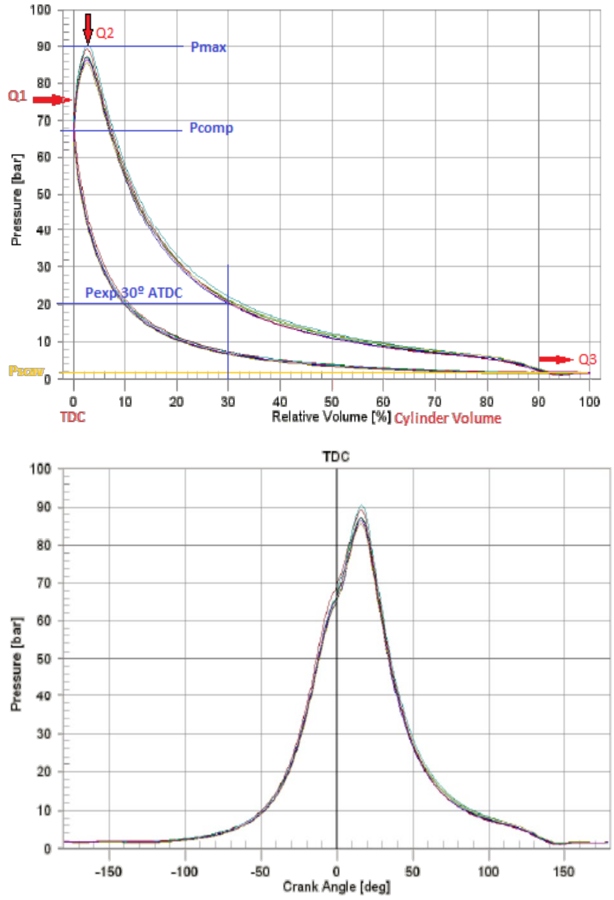

Figure 13 - Illustrations of closed (upper), and open diagram (bottom) diagrams, and their respective contained information.

As the cylinder is closed, the mass of the combustion gases remains constant throughout the cycle, then it is the temperature of the gases that must be minimised. Considering the dotted theoretical process of Figure 13, which is characterised by keeping Pcomp and (Q1+Q2) constant, but reducing Pmax. The new expansion line will necessarily lie above the previous one, and will hence represent the higher exhaust temperature at the start of the exhaust stroke. This will increase Q3 (energy loss through exhaust gases) with a corresponding reduction in available energy for mechanical work, **i.e.**, a reduced process efficiency. Or in other words, a higher fuel consumption to obtain the desired power from the considered cylinder. Figure 13, also makes evident that in reality, it is the pressure increase from Pcomp to Pmax that determines the efficiency of the diesel process. The smaller the pressure rise, the longer the combustion takes place while the piston is moving downwards, and the higher the exhaust gas temperature.

Figure 13, also suggests the measurement of the angular shift from TDC to the crank position where Pmax takes place in reality. It is also important to control the expansion pressure (typically at 30º ATDC) in order to keep control over the combustion speed and possible afterburning.

For engines with a modest pressure rise (up to 3 Bar/CA), it should be expected that a small variation in the RPR from the correct values will result in an important increase in the fuel consumption. Small percentages, result in heavy extra costs annually.

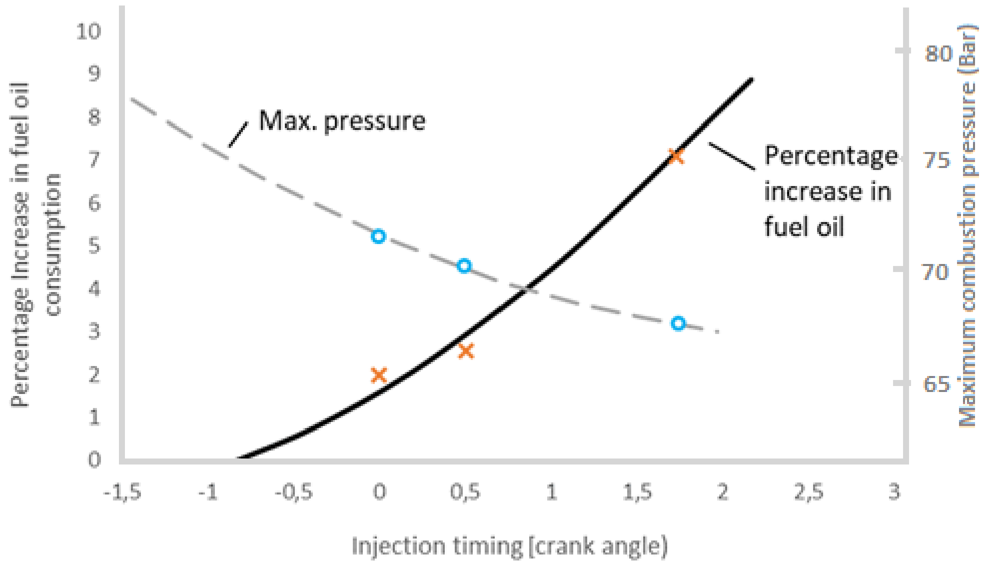

Figure 14 - Influence of injection timing on SFOC.

Examples

A medium speed 4 stroke diesel engine operated with HFO, was monitored and it was possible to identify a consistent decreasing in its turbocharger pressure ratio Ps/Po, these values were logged and plotted on diagram A. The reference diagram was established based on shop trial data, as per Figure 12.

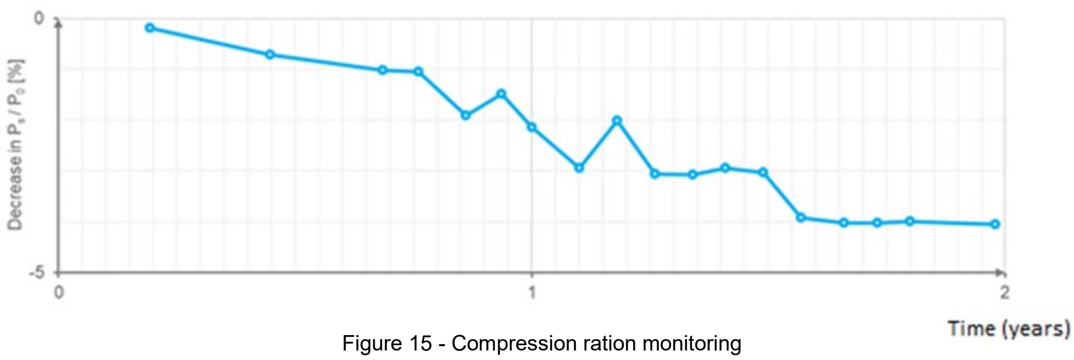

Figure 15 - Compression ration monitoring

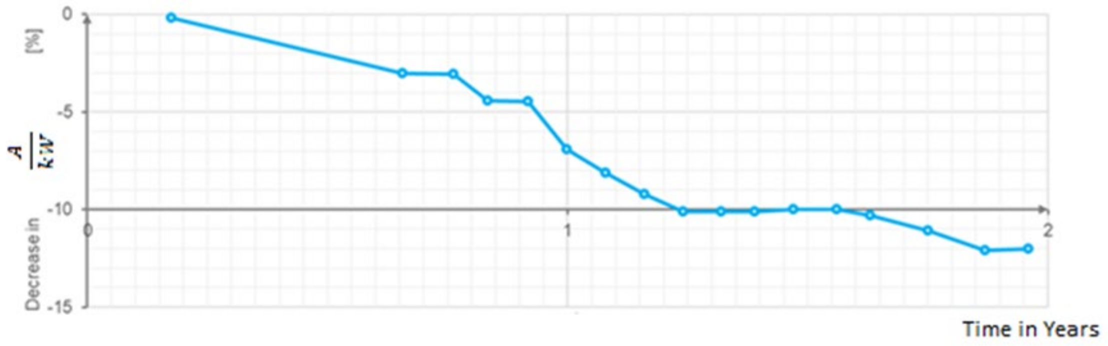

Figure 16 - Decrease of $\frac{A}{kW}$ (air coefficient/brake power) as a function of time.

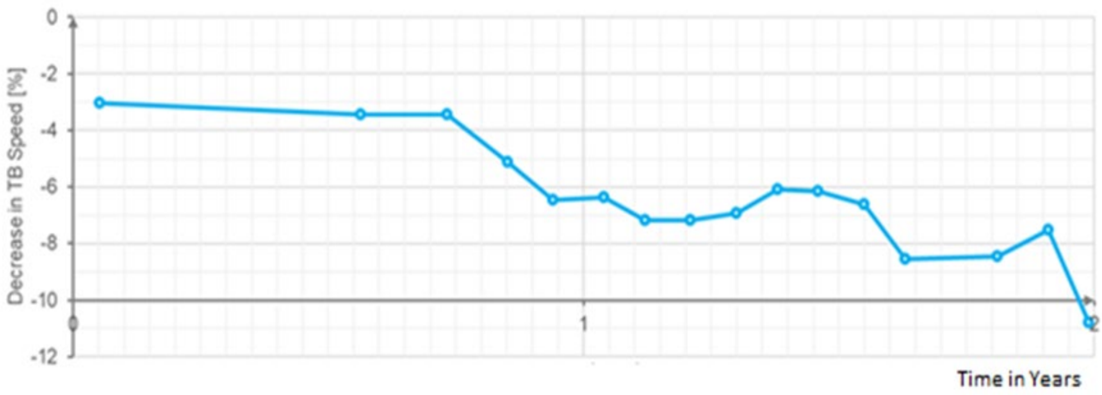

Figure 179 - Reduction of turbocharger speed as a function of time.

In this particular case, a reduction in mass air flow has occurred and caused an increase in the exhaust temperatures and the engine was operated at reduced load, bringing the turbocharger to a non-efficient matching point.

Some improvements were done, but the actual root cause was not tackled and, the engine thermal load was still very high, so a number of engine failures were registered and the accumulated thermal stresses lead to cylinder head cracks as well as piston crown cracks.

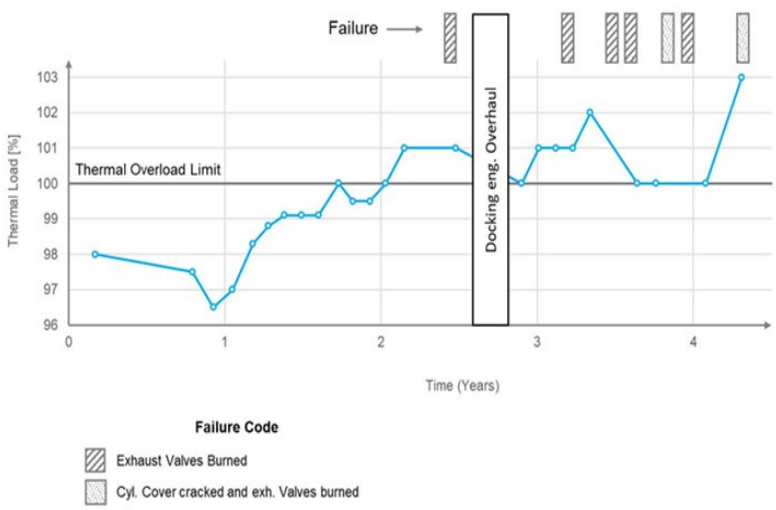

Figure 20 - Monitoring of engine thermal load along 5 years

Finally, the engine was scrapped as the engine block and most of the cylinder components like cylinder heads were found aged.

Conclusions

The correct adoption of these new technologies requires a big change of mind-set in management, comprising the adoption and the integration of new technologies within the business cycle. It involves a change in the culture of an organization and includes the development of trust and belief in a black box, as well as the acceptance of shortcomings such as errors.

The benefits to ship-owners and operators by implementing a condition monitoring strategy are related to cost savings and the predictability of maintenance operations. Cost savings are related to reductions in maintenance needs such as inspections and repairs, decreased costs due to failures and downtimes and unplanned maintenance, lower insurance costs, and savings due to improved equipment performance that may lead to decreased fuel consumption. Predictability, besides reducing the uncertainty around decisions, is also a source of competitive advantage and can be used in many business relations to attract charterers.

The accurate picture obtained of a vessel's current and future status also contributes to a higher awareness of system capabilities. This allows preventive actions to be performed to increase reliability and safety, and to ensure that ship capabilities match operational requirements.

In addition, the increased knowledge and documentation of a ship’s health status and maintenance history will increase the second-hand value of a well-maintained ship.

Moreover, with onshore support possible, part of the operations can be delegated to specialized personnel and systems that can keep track of performance and risks. This also has a positive effect on the crew competences, since on board personnel can be involved in a continuous learning process in close collaboration with the onshore organization.

In this way, not only will the ship be maintained effectively, but the competence and experience of the crew will also be enhanced.

Component manufacturers can also benefit from condition monitoring to improve the quality of components during the lifecycle by analysing operational data and trying to minimize losses and failures in relation to different operation profiles.

Maintenance can be optimized based on condition and suggestions regarding the timing of repairs, substitutions, etc. can be given to the shipowner.

The quantity of data, which may be generated by a vessel, is huge, being only possible to deploy successfully one such system if one can use a Big Data dedicated software like BOEM-S.

BOEM-S is receiving the ship data, correcting it and analysing it, using one expert system, dedicated to ship propulsion and auxiliary machinery, comparing automatically the actual data corrected with corrected data, and identifying the deviations, creating alarms and tendencies.

Bibliography

N.Petrovsky, Marine Internal Combustion Engines MIR Publishers . Moscow

MaK Projecttierung Project M 551 / M552

Incropera Dewitt, Principles of Heat and Mass Transfer Willey Seventh Edition;

MAN B&W Operation Instructions Manual of MC and MCE Engines 2004;

ISO 3046 Parts 1, 2, and 3 2010;

Annexes

Graph		Deviation	A	B	C	D	E	F	G	H	I	K
1	$P_{scav}=f(P_{mi})$	+							■			
		-				■				■	▲	■
2	$P_{comp}=f(P_{scav})$	+						■				
		-	■									
3	$P_{max}=f(P_{mi})$	+			■		■					
		-		■								
4	$P_{exp}=f(P_{mi})$	+		▲			▲					
		-			▲							
5	$\alpha P_{max}=f(P_{mi})$	+		■			▲					
		-			■							

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