Ships are an integral part of modern commercial transport, leisure travel, and military system. A diesel engine was used for the first time for the propulsion of a ship sometime in the 1910s and has been the choice for propulsion and power generation, ever since. Since the first model used in ship propulsion, the diesel engine has come a long way with several technological advances. A diesel engine has a particularly high thermal efficiency. Added to it, the higher energy density of the diesel fuel compared to gasoline fuel makes it inherently, the most efficient internal combustion engine. The modern diesel engine also has a very unique ability to work with a variety of fuels like diesel, heavy fuel oil, biodiesel, vegetable oils, and several other crude oil distillates which is very important considering the shortage of petroleum fuels that we face today.

In spite of being highly efficient and popular and in spite of all the technological advances, the issue of exhaust gas emissions has plagued a diesel engine. This issue has gained a lot of importance since 1990s when IMO, EU, and the EPA came up with the Tier I exhaust gas emission norms for the existing engine in order to reduce the NOx and SOx. Harsher Tier II and Tier III norms were later announced for newer engines. Diesel fuels commonly used in marine engines are a form of residual fuel, also know as Dregs or Heavy Fuel Oil and are essentially the by products of crude oil distillation process used to produce lighter petroleum fuels like marine distillate fuel and gasoline. They are cheaper than marine distillate fuels but are also high in nitrogen, sulfur and ash content. This greatly increases the NOx and SOx in the exhaust gas emission. Ship owners are trapped between the need to use residual fuels, due to cost of the large volume of fuel consumed, in order to keep the operation of their ships to a competitive level on one hand and on the other hand the need to satisfy the stringent pollution norms as established by the pollution control agencies worldwide.

Newer marine diesel engines are being designed to meet the Tier II and Tier III norms wherever applicable but the existing diesel engine owners are still operating their engines with the danger of not meeting the applicable pollution norms worldwide. Here we make an effort to look at some of the measure that the existing marine diesel engine owners can take to reduce emissions and achieve at least levels prescribed in Tier I. Proper maintenance and upkeep of the engine components can be effectively used to reduce the exhaust gas emission. We introduced a pilot program on diesel engine performance monitoring in North America about two years ago and it has yielded quite satisfying results for several shipping companies and more and more ship owners are looking at the option of implementing this program on their ships.
1.0 Combustion:

The Combustion process is one of the most critical aspects of a diesel engine. It is a highly complex process characterized by unsteady and heterogeneous kind of physical and chemical reactions. It greatly affects the critical aspects of a diesel engine operation like the fuel consumption, the exhaust gas emission, the life of engine components, and the overall optimum performance of the engine. Although the experts very well understand the basic concept of the combustion process, a comprehensive and accurate mathematical model is very difficult to propose. An accurate mathematical model of the complete combustion process will involve models of compressible viscous air motion, fuel lines, fuel pumps, fuel injectors, fuel spray penetration, fuel atomization and evaporation, air and fuel mixing, combustion kinetics, and premixed and diffusion burning process. Even attempting to propose a mathematical model of the combustion process involving all the factors mentioned above, would be extremely lengthy and time consuming.

Over the years, several attempts have been made to simplify the mathematical model of the combustion process. One of the most successful ones has been the heat release or fuel burning rate model. The rate of heat release can be defined as the rate of release of chemical energy from the fuel during combustion. Using high-speed photography and heat release analysis, it was proposed in the model that there are four different stages of the combustion process marked by rapid change in rate of heat release. The four stages are ignition delay, pre-mixed burning, diffusion controlled burning, and combustion tail phase.

1.0 Ignition Delay Phase: Ignition delay period is the time interval from the point of beginning of injection to the point of ignition of fuel. Ignition will not start until the injected fuel is completely vaporized, the air-to-fuel ratio is close to stoichiometric value, and the temperature is sufficiently high for self-ignition.

2.0 Pre-Mixed Burning Phase: As the ignition of fuel begins, the heat transfer to the adjacent area increases rapidly, thereby resulting in rapid combustion of majority part of the injected fuel. This phase is the true characteristic of a diesel engine combustion process. This phase is much more prominent in direct-injection type engines.

3.0 Diffusion Controlled Burning Phase: The rapid combustion of fuel in the pre-mixed burning phase is followed by a lower heat release diffusion controlled burning phase. The pre-mixed burning phase changes the air-to-fuel ratio and hence the diffusion controlled burning phase is mainly governed by the amount of air available for combustion (air entrainment) and the air-fuel mixing process controlled by swirl and turbulence. The diffusion controlled burning phase may also include the burning of fuel deposited on the wall, which is drawn off by subsequent air movement.

4.0 Combustion Tail Phase: Following the diffusion controlled burning phase the heat release will continues at a lower rate and eventually approach zero somewhere in the expansion stroke. Due to the falling pressure and temperature in the expansion stroke, the chemical reaction slows down in the combustion tail phase, resulting in lower heat release.
Several others have also proposed the heat release model but neither one of them has been accurate enough to be applicable to all the cases. A semi-empirical approach has been the most widely accepted approaches amongst them. The approach proposes that two parts relating to pre mixed and diffusion burning phases mainly characterize the heat release or fuel burning. The total heat release or fuel burning is proportional to sum of these two individual phases.

\[ FB \propto f_1(t) + f_2(t) \]

Where, \( f_1(t) \) = pre mixed fuel burning rate
\( f_2(t) \) = diffusion burning rate

Introducing proportionality constant gives us

\[ FB = \beta \cdot f_1(t) + (1 - \beta) \cdot f_2(t) \]

The proportionality constant \( \beta \) is considered to be controlled largely by the ignition delay, since the fuel injected during this period governs the pre mixed burning phase, and also by the overall cylinder equivalence ratio. Thus the proportionality constant is given by

\[ \beta = \frac{1 - a \cdot P^b}{ID^c} \]

Where, \( ID \) is ignition delay and \( a, b, \) & \( c \) are suitable constants to fit engine cylinder pressures.

The approach further proposed that the pre mixed burning rate distribution and diffusion burning rate distribution is given by

\[ f_1(t) = 1 - \left(1 - t^{k_1}\right)k_2 \]
\[ f_2(t) = 1 - \exp(-k_3t^{k_4}) \]

Where, \( k_1, k_2, k_3, \) & \( k_4 \) are shape factors determined as a function of the engine operating condition.

Ignition delay is the time interval between the start of injection and the start of combustion. The start of injection is taken from the point the injector needle starts to lift. The start of combustion, though not precisely defined, is taken from the change in slope of the heat release rate versus time curve, which occurs at ignition. Ignition delay has been generally described by the equation

\[ ID = A \cdot p^{-n} \cdot \exp\left(\frac{E}{R \cdot T}\right) \]

Where, \( ID \) = ignition delay in ms
\( E \) = apparent activation energy for the fuel auto ignition process
\( R \) = universal gas constant
\( P \) = gas pressure in atm
\( T \) = gas temperature in Kelvin
\( A \) & \( n \) are constants dependent on the fuel, injection, and air flow characteristics

2.0 **Understanding Stoichiometri:**

A definite amount of oxygen is required for the complete combustion of a 1 kg of fuel, which can be calculated from the chemical composition of the fuel. The amount of oxygen contained in the cylinder during the compression stroke determines
the amount of fuel capable of being burned per power stroke.

Atmospheric air consists of approximately 79% by volume of nitrogen and approximately 21% by volume of oxygen. However, nitrogen does not play any role in the combustion process. If the theoretical amount of oxygen required for combustion of 1 kg of fuel is taken as being $O_{\text{min}} = \frac{8}{3} c + 8 h - o + s \, \text{m}^3$ kg $O_2$ per kg fuel oil, then

$$L_{\text{min}} = \frac{1}{21} \cdot \left(\frac{8}{3} c + 8 h - o + s\right) \cdot \frac{1}{\text{1.314}} \, \text{m}^3 \, \text{air per kg fuel oil},$$

as the volume mentioned earlier is given at 1 bar and 20° C, or

$$L_{\text{min}} = \frac{8}{23} \left(\frac{8}{3} c + 8 h - o + s\right) \, \text{kg/kg oil}.$$

In reality, however, more air than the theoretical necessary minimum amount is required, since this theoretical amount would just be sufficient. Despite all the efforts made, great difficulties are experienced in trying to achieve complete combustion when only the theoretically necessary amount of air is available, the reason being that this requires the atomized fuel and combustion air to be completely and homogenously mixed, and in such a way that each molecule of fuel is surrounded be the number of $O_2$ molecules necessary for the complete combustion of the fuel molecule.

In a diesel engine the mixing of the fuel and air takes place shortly before and while combustion is taking place and hence such an ideal uniformity of distribution is very difficult to achieve. Incompletely combusted fuel particles either are too late in reacting with the oxygen molecules or do not react with oxygen at all and leave the engine without releasing their total heat of combustion. The incompletely combusted particles leave the engine either as soot or unburned gas in the form carbon monoxide.

Complete combustion of fuel requires the fine atomization or vaporization of the fuel, effective swirling of the air/fuel mixture, and finally the availability of a greater amount of air than the stoichiometric value. Attempts to achieve these conditions are made in an effort to make up for the unavoidable incompleteness in the formation of the mixture. Consequently, in order to be sure that combustion is complete, one must not only
introduce $L_{\text{min}} \, m^3$ of air (at 20° C and 1 bar) per kg fuel into the cylinders, but $\lambda \cdot L_{\text{min}} \, m^3$ (at 20° C and 1 bar) air per kg of fuel, where $\lambda$ is termed as the excess air coefficient, and is a value greater than 1. Depending on the required speed of combustion, and the quality and the homogeneity of the air/fuel mixture, excess air coefficient values are found to be approximately 1.1 with Otto engines and approximately 2 – 4 with diesel engines.

The air excess coefficient can be determined after the amount of $CO_2$, $O_2$, CO and $N_2$ in the exhaust gases has been found by means of the Orsats apparatus. If the percentage volume of these gases is taken to be $co_2$, $o_2$, $co$ and $n_2$ respectively, the amount of air fed to the engine will be approximately $\frac{1}{0.79}$ times greater than the amount of nitrogen to be found in the exhaust gas, as the nitrogen in air is approximately 79 percent by volume and does not play any part in the combustion process. Assuming that the remaining 21 percent of the atmospheric air is oxygen, the excess air to be fed to the engine will then be $\frac{1}{0.21}$ times the content of oxygen in the exhaust gas. The carbon monoxide in the exhaust gas should have been burned to carbon dioxide in accordance with the reaction scheme

$$2CO + O_2 \rightarrow 2CO_2$$

or

$$CO + \frac{1}{2}O_2 \rightarrow CO_2$$

which means that not enough air has been used in the combustion process. The amount of air corresponding hereto is $\frac{0.5}{0.21}$ times the amount of CO existing in the exhaust gas.

Since the excess air coefficient $\lambda$ is the ratio between the amount of air fed to the engine and the theoretically necessary amount, we get:

$$\lambda = \frac{\text{amount of air introduced}}{\text{theoretical amount necessary}} = \frac{\text{amount of air introduced}}{(\text{air introduced} - \text{excess air})}$$

$$\lambda = \frac{1}{0.79} \cdot \frac{n_2}{n_2 - \frac{1}{0.21} \cdot o_2 + \frac{0.5}{0.21} \cdot co}$$

$$= \frac{n_2}{n_2 - \frac{79}{21} (o_2 - 0.5 \cdot co)}$$

If the weight of the air introduced is $L$ kg per kg of fuel oil, the weight of the exhaust gas will be $(L+1)$ kg per kg of fuel oil.

The amount of air to be introduced to achieve complete combustion of 1 kg fuel is therefore $V_L = \lambda \cdot L_{\text{min}} \, m^3$ (at 20° C and 1 bar) per kg fuel oil. With good approximation one can assume that the fuel/air mixture in a diesel engine has the same volume $V_b$ as the air, i.e.

$$V_L = \lambda \cdot L_{\text{min}} = V_b \, m^3$$

at 20° C and 1 bar mixture per kg fuel.

The amount of fuel contained in 1 $m^3$ of fuel/air mixture is then $\frac{1}{V_b}$ kg per $m^3$ (at 20° C and 1 bar).

At complete combustion, 1 kg of fuel develops $h_i$ kJ, which means that in the combustion of 1 $m^3$ of fuel/air mixture $\frac{h_i}{V_b}$ kJ per $m^3$ (at 20° C and 1 bar) is developed.

### 3.0 Exhaust Gas Emission:

As the whole world becomes more and more environmentally conscious and as the word ‘Green’ takes on a whole new meaning, the diesel engine exhaust gas emission has become a topic of great attention for the environmental agencies, the classification societies, the engine manufacturers, and ultimately the diesel engine operators. The diesel engine combustion process almost always leaves byproducts of oxides of nitrogen, unburned hydrocarbons, carbon monoxide, particulate matter, and smoke.
The oxides of nitrogen (NOx), is a group of toxic gases formed by the reaction of nitrogen and oxygen. At extremely high temperatures of combustion, these two gases react to form nitrogen dioxide NO₂ and nitric oxide NO. These gases are major source of ground level ozone (smog) and are also a significant source of acid rains and soot formation. At highly elevated temperatures encountered during combustion, mainly in pre-mixed burning phase, oxygen molecule dissociates into atomic oxygen and reacts with nitrogen to form nitric oxide. When the combustion temperatures are not sufficiently high for atomic dissociation of oxygen then the oxygen molecules react with nitrogen to form nitrogen dioxide. This may happen during the diffusion controlled burning phase of combustion.

There will always be a certain amount of fuel left unburned or partially burned. This shows up as unburned hydrocarbon in the exhaust gasses. These hydrocarbons are toxic in nature; having adverse effects on our health and in some cases are known to cause cancer. In most cases, this amount is negligible and some part of it comes from unburned lubricating oil. Unburned hydrocarbons are formed due to several reasons like insufficient oxygen for complete combustion in some parts of the cylinder and insufficient temperature for ignition of fuel in some other parts of the cylinder. It can also be caused from poor atomization of fuel towards the end of injection or fuel leaking from the sack of the injector after the fuel injector needle is closed. The amount of unburned or partially burned hydrocarbons can be minimized by proper maintenance of fuel injectors and cylinder conditions.

Carbon monoxide is formed as an intermediate product of hydrocarbons fuel combustion. It is mainly formed due to the lack of adequate oxygen to form carbon dioxide or due to insufficiently high temperatures. Carbon monoxide may eventually oxidize into carbon dioxide in the presence of sufficient oxygen and higher temperatures. Just as in case of nitrogen, reaction of carbon with oxygen depends heavily on availability of adequate oxygen and high temperatures for complete oxidation. In absence of either of them, carbon oxidation is incomplete, leaving behind carbon monoxide. As compared to a gasoline engine though, a diesel engine produces less carbon dioxide and carbon monoxide.

The smoke emitted from a diesel engine comes mainly from unburned fuel oil and lubricating oil. The smoke referred to as blue-white smoke is primarily seen in cold starting or idling condition and this usually disappears as the engine warms up and takes on load. Improper air/fuel ratio can in some cases; cause the engine to smoke even in warm conditions and under load but that smoke is black in color. The receding values of temperature and pressure in the combustion tail phase prevents the remaining unburned fuel from burning completely and this gives rise to majority of the exhaust gas emissions including smoke, soot, and particulate matter.

If all of the above pollutants from the exhaust gas emission are studied carefully, it can be noted that most of them are formed due to improper air/fuel ratio and temperature/pressure conditions that are not conducive for complete combustion of the fuel. With the advancement in technology, the modern diesel engine is designed to minimize the exhaust gas emissions. Over time though, the engine components start to wear and the engine becomes untuned and imbalanced. This leads to disturbance in the air/fuel ratio. From our experience in serving the diesel engine community for over 30 years we have found that the problem of pollution from exhaust gas emissions does not come mainly from bad engine design or bad fuel quality but it comes from the inability to maintain the engine or the auxiliary equipment up to the manufacturers specification and in some cases it comes from badly designed propulsion systems.
4.0 Achieving Higher Efficiency:

A diesel engine is a very reliable and efficient power plant. No other internal combustion engine extracts so much energy from the fuel as the diesel engine does. With all the technological advances made by the engine designers, the diesel engine has quite a different image today than it did a few decades ago. Today, the diesel engine has emerged as a highly reliable, powerful and fuel-efficient engine.

Achieve high fuel efficiency requires converting the chemical energy of the fuel which is measured in calorific value, to usable power with minimum power losses. For optimum results, the duration of combustion must be kept as short as possible. Shorter combustion duration ensures that all the energy is released from the fuel immediately as the piston passes the top dead center (TDC) position. This results in much higher combustion pressure for a short duration of time and the released energy can be converted more efficiently into tractive power. The modern diesel engine has undergone a thorough redesign in order to work with these high pressures and temperatures.

In order to achieve a rapid and yet complete combustion, the fuel must be injected in the shortest time possible. To achieve shortest fuel injection interval, the fuel injection system must operate at extremely high injection pressure. Such high injection pressure is required to achieve fine atomization and sufficient penetration of the injected fuel into the highly compressed air in the combustion space. Aided by the air swirl, this results in intensive contact between the injected fuel mist and the available combustion air in the combustion space. Following a brief ignition delay period, the fuel will ignite and most of the fuel will burn in a very short period of time. If majority of the fuel injected does not ignite in a short period of time, some portion of energy released is wasted and might also induce higher thermal stresses in the cylinder liner.

5.0 Engine Performance Monitoring:

The theoretical aspect of diesel engine combustion process and the exhaust gas emission, though extremely important, is generally outside of the scope of knowledge of the engine room crew. One major aspect of pollution control that has been overlooked so far is comprehensive training of the engine room crew in proper operation and maintenance of the engine and its auxiliary equipment. Certain parameters influencing the combustion process are designed into the engine and hence are very difficult to change. Other parameters related to the actual functioning and upkeep of the fuel system, cylinder condition, and the turbocharger are in the hands of the engine room crew. These parameters are the primary focus of our paper. Even without a thorough understanding of the theoretical aspects, there are some practical procedures that the engine room crew can follow effectively to achieve optimum performance of the engine and lesser emissions. It all begins with the primary requirement that the engine room crew should have an interest in proper operation, maintenance, and upkeep of their equipment as per the equipment manufacturers guidelines, which comes from proper education and training. A well-trained crew is more likely to take interest in the maintenance and upkeep of their engine and other equipments.

Secondly, most ship owners and ship management companies are moving from time based maintenance program to condition based maintenance program with longer time between overhaul. This can be achieved by implementing a diesel engine performance monitoring program. This program primarily involves regular inspection and monitoring of the critical aspects of diesel engine like the cylinder liner and piston rings inspection, cylinder pressure and combustion characteristic monitoring, fuel injection pressure monitoring, etc. This program can be split into two parts for the sake of simplicity. The first part is visual inspection of cylinder liner and piston rings for keeping a check
on the amount of wear. Excessive wear in cylinder liners and piston rings are directly linked with engine performance and pollution. A cylinder liner is prone to wear and crack formation over time, the latter of which is hard to detect or predict and this can result in catastrophic failures. Piston rings are also prone to wear over time, which eventually results in collapsing of rings, stuck rings or broken rings. By regularly monitoring the wear limits on cylinder liner and piston rings, unexpected overhauls and delays in operation can be avoided.

**Turbocharging**

First we want to illustrate the simplicity of keeping the turbocharger performance under control, as this is a very important part of engine performance and emission control. In addition, we will provide examples how proper inspection and performance evaluation is an absolute necessity.

With today’s modern engines only the constant pressure system is used. With this system, each cylinder is connected to a single exhaust manifold, whose volume will be sufficiently large to damp down the unsteady flow entering from each cylinder in turn. Only one turbocharger needs to be used, with a single entry, but in many cases more than one is used in order to insure that a reasonable booster pressure can be maintained, even in the event of one turbocharger failure.

The constant pressure system is relatively simple. Apart from its simplicity is the fact that by definition, the conditions of the turbine entry are steady over time. All losses in the turbine that result from unsteady flow are absent. A single entry turbine is used, eliminating end of sector losses that exist with a multi-entry turbocharger (pulse system). In addition, a smaller turbocharger has difficulties in maintaining close tolerances and small clearances.

Consequently, large turbochargers have lower leakage losses and thus higher efficiency. All of this will result in a high average turbine efficiency with the constant pressure system, even though the turbine inlet pressure is not absolutely constant since not all the pulse energy is lost. Some of it is recovered in the form of a higher inlet temperature at the turbine.

With uni-flow two-stroke engines in which some control of the amount of energy supplied to the turbine is achieved by an early opening of the exhaust valve. However, this becomes a problem with an increase in engine ratings due to loss of piston work, which outweighs the additional work that the turbocharger is able to provide.

Consequently, there is a crossover point where the use of constant pressure turbocharging becomes more advantageous than the pulse system. Primarily, the application in which the constant pressure system is used is highly rated engines that are required to run at or near their rated power output for most of their service life.

The above is a very simplified explanation in regards to the use of the constant pressure system, but it is our own experience that most vessel owners and their operating crew do not understand the importance of turbocharger performance, and the problems caused if proper operating parameters are not maintained. Since the two-stroke engine is neither self-aspiring nor self-exhausting, it relies on a positive pressure drop between the inlet and exhaust manifold in order to run at all.

The scavenging process is the only real key to a successful relationship between the inlet manifold and exhaust manifold pressures. It is a function of the overall turbocharger efficiency, the turbine inlet temperature and to a lesser extent, the air-to-fuel ratio. Of course it is the turbine area that principally decides the exhaust manifold pressure and the air-to-fuel ratio and overall turbocharger efficiency that controls the relationship between the inlet and exhaust manifold pressure.

The performance of a turbocharger is very dependent on the gas angles at the entry to the
impeller, diffuser, and turbine rotor. The blade angles are set to match the gas angles, but a correct match can only be obtained when the mass flow is correct for a specific rotor speed. Away from this design point, the gas angles will no longer match the blade angles and an incidence loss will occur due to separation and subsequent mixing of high and low velocity fluid. These losses will increase with increasing incidence angles; consequently, turbochargers are not well suited for operation of a wide mass flow range. Due to this it follows that it is only possible to match the turbocharger correctly to a specific point in operating range of the engine.

Based on the above, it must be understood that since the turbocharger can only be matched to a specific point in the operating range of the engine, then the required booster pressure achieved is directly related to the mean effective pressure of the engine at that point. Apart from this, when incidence losses in the turbocharger become an issue (away from the layout point), including drop in exhaust gas energy and temperature to the turbine, then the turbocharger can no longer supply the correct charge air pressure to the engine, as is required for a specific mean effective pressure that exists at any partial load condition.

The larger the compressor pressure ratio, the larger this deficit becomes at partial load condition. It is also important to understand that turbocharger speed and therefore charge air pressure are not directly related to each other, since the operating point is determined by the equilibrium between compressor power input and turbine power output. It can only be stated qualitatively that the turbocharger speed increases with the rate of gas flow through the engine and the inlet temperature to the turbine, which relates to the load of the engine.

We have calculated the turbocharger efficiency at 100% engine load condition from the test stand and sea trial of a MAN B&W 6S60MC engine. In regards to the overall turbocharger efficiency it is the same at both trials, 66%, which should be considered to be a reasonable overall turbocharger efficiency. The changes we see from both trials are the compressor and turbine efficiencies. On the sea trial the compressor efficiency increased and turbine efficiency decreased. Based on the formulas used we can see the following relationship:

$$\eta_{tot} = 0.9055 \frac{T_1(R_1^{0.286} - 1)}{T_2(1 - R_2^{0.265})}$$

Based on above formula we can see that by lowering the compressor inlet temperature ($T_1$) we will lower the overall turbocharger efficiency and by lowering the turbine inlet temperature we will increase the overall turbocharger efficiency.

Since the overall turbocharger efficiency was the same at both trials, we have evaluated the parameters that changed the compressor and turbine efficiencies. Based on the compressor efficiency formula we can see the following:

$$\eta_{comp} = \frac{361440 \cdot T_1(R_1^{0.286} - 1)}{\mu(u)^2}$$

$R_1$ has increased on the sea trial due to a higher ambient pressure, higher booster pressure, and a slight pressure drop over the air cooler, it had increased from 0.4102 on the test stand to 0.4163 on the sea trial, and due to the lower turbocharger rpm on the sea trial, the factor $u^2 = (\pi \cdot 0.6588 \cdot 13000)^2$ is lower, it has decreased from 791487925.5 to 723191281.4. The changes in these parameters increase the compressor efficiency on the sea trial. In regards to the turbine efficiency we can see that $R_2$ has changed from 0.2628 on the test stand to 0.2572 on the sea trial. This is mainly due to the increase in ambient pressure and pressure after the turbine and a decrease in exhaust manifold pressure. But in order to simplify the situation in regards to turbine efficiency we can use the following relationship:
As stated earlier, turbocharger speed and booster pressure are not directly related to each other. We can clearly see that in the results from both trials as discussed. At 100% engine load on the sea trial the turbocharger rpm was 13000 rpm with a booster of 2.41 bars. On the test stand at the same load, the turbocharger rpm was 13600 rpm with a booster pressure of 2.32 bars. The changes in turbocharger performance mainly relate to the ambient conditions that existed during these two trials.

This is the kind of reasoning that needs to be conducted between the technical staff and the engine room staff in order to ensure that turbocharger performance is kept within their design and layout parameters.

**Charge Air Cooling**

In general one can state that the principal reason for turbocharging is to increase the power output of an engine without increasing its size. This is achieved by raising the inlet manifold pressure, by that increasing the air mass, allowing more fuel to be burnt. But since it is nearly impossible to compress air without a temperature rise and since the objective is to raise the air density, this temperature rise during compression will partly offset the benefit of increasing the pressure since

\[
\rho = \frac{P}{R \cdot T}
\]

Due to this the objective must therefore be to obtain a pressure rise with a minimum temperature rise. This implies isentropic compression, in which case the temperature rise will be given by the following equation:

\[
\Delta T_s = (T_2 - T_1)_s = T_1 \left[ \frac{P_2}{P_1} \right]^{\frac{\gamma - 1}{\gamma}} - 1
\]

But unfortunately due to the inefficiencies in the compressor, the actual temperature rise must consider the isentropic efficiency of the compressor, so consequently we can rewrite the equation as follows:

\[
\Delta T'_s = T_2 - T_1 = \frac{(T_2 - T_1)_S}{\eta_c} \left[ \frac{P_2}{P_1} \right]^{\frac{\gamma - 1}{\gamma}} - 1
\]

As can be seen from the above equation, the higher the isentropic efficiency of the compressor, the closer the temperature rise approaches the isentropic temperature rise. Several very important points emerge from the above. First, the benefit obtained by raising the inlet manifold pressure is reduced substantially due to the temperature rise in the compressor, which itself is dependent on the compressor efficiency. Secondly, the advantage of high compressor efficiency in order to hold the air temperature to lower levels is relatively small, but worthwhile. Thirdly, in absolute terms, the benefit that can be obtained by cooling the compressed air back to nearly ambient conditions is substantial and it increases with increased pressure ratios.

**6.0 Parameters that have a direct effect on fuel consumption and the combustion process**

**6.1 Pmax Pressure**

The Pmax pressure in relation to the mean indicated pressure is a very important parameter in regards to the specific fuel consumption. A larger Pmax/MIP ratio results in better combustion process efficiency. Consequently, one should always make sure to operate the engine with the largest ratio as possible (ultimate Pmax pressure) at each load profile of the engine. As an example, by increasing the Pmax by 1 (one) bar we can reduce the specific fuel oil consumption by 2-3g/BHPH depending on engine type.
6.2 **Heat release starting point**

The beginning of the heat release will start where the fuel has completely evaporated. The air to fuel ratio is close to stoichiometric value and the temperature is sufficiently high for self-ignition. When the heat release starts, heat transfer to the adjacent area of the fuel spray will increase very rapidly, and the combustion will rapidly cover a major part of the fuel spray. It is this rapid combustion (following the ignition delay) that is characteristic of the diesel combustion process.

Based on the above it becomes clear that it is the duration of the heat release and its distribution in the combustion chamber that will affect the combustion process efficiency. As an example, by increasing the heat release by an additional 5° crank angle, the fuel consumption will increase by approx. 2g/BHPh.

6.3 **Tangential turning effort**

For this particular parameter the Pmax and expansion pressure play a major role, as explained later in this report. Incorrectly adjusted fuel pump timing will increase the fuel consumption quite substantially as explained in this paper. By ensuring proper function and operation of the engine, emissions will be very low due to optimized fuel consumption.

Following are three cases we have encountered in the past few years.

---

**Fig. 2 Alpha Pressure Curve**

**Fig. 3 Torque on the Crankpin Per Crank Angle**

**Fig. 4 Calculation of Tangential Turning Effort**
CASE STUDIES

CASE STUDY #1: Poor Engine Maintenance and Performance Control.

In this case we are dealing with an MAN B&W 6S60MC engine rated at 9,428kW at 92rpm. The major problem was that the engine could not start at all after a shutdown in cold weather. What led up to this problem was very poor maintenance procedures and performance evaluation. When the engine failed it had just over 26,000 initial operating hours. This very popular engine is commonly used on bulk carriers and tankers, with over 1,000 engines in service.

Since the engine is an S-engine (Super Long Stroke) certain design parameters must be considered; the most important parameter is the amount of liner surface that is exposed to the combustion flame front. This has a major influence on cylinder oil lifetime and lubrication ability. The larger the exposed liner length, the poorer the cylinder condition will be and there will be higher resulting liner wear. The exposed liner length is a function of:

- Piston design. A high top-land will provide better protection of the liner running surface than a relative lower top-land design.
- Engine stroke and connecting rod ratio. A long stroke engine with a low connecting rod ratio exposes a larger cylinder area to the combustion than does a short stroke engine having a high connecting rod ratio.
- Actual duration of heat release, measured in degrees of crank angle. A short combustion duration is preferable.

In addition, the combustion exposure time has an important function in regards to the deterioration of the lubrication oil film on the liner surface. Most chemical reactions are time related, and the degree of oil break down is proportional to the time in which the cylinder oil is exposed to the combustion. The combustion exposure time is a function of:

- Engine revolutions. Lower engine rpm gives a higher exposure time.
- Duration of heat release as previously mentioned.

Other design parameters play an important role but not in regards to the engine performance itself. As an example: the position of the oil quills to the upper piston ring in the TDC position - including the horizontal distance between the oil quills in order to ensure proper oil distribution in the liner.

In the past one has always judged the cylinder conditions relative to specific wear of cylinder liners and piston rungs. Low wear rates was the primary indicator of good liner condition. From a reliability standpoint, the condition of the piston rings and their ring grooves is more relevant, as this condition apparently is harder to predict – and could eventually put engine reliability at high risk.

If the combustion process in the cylinder is examined and the heat release is measured, it is then well known that the combustion duration in crankshaft degrees is load dependent, with the largest duration at the highest load condition. A very long combustion duration means that the piston has moved far down in the cylinder liner before combustion stops. In some cases it may be so far that too much liner surface area is exposed to the combustion flame front, consequently, the cylinder lubrication is burned away resulting in a severely damaged oil adhesive wear.

It must also be understood that the lack of proper sealing between the piston rings and cylinder liner means an increase in thermal load of the cylinder liner. Experience shows an almost exclusive correlation exists between poor piston ring condition and thermal cracks in the liner running surface in the top piston ring zone of the liner. For the reasons stated, the safety margin of perfect cylinder condition is very narrow, and can only be widened by practical
means, such as improvement of overhaul procedures and their frequencies and strict engine performance control.

In order to start an engine performance program, certain design parameters need to be known; even beyond regular operating data taken on the test stand and the original sea trial. One such parameter is the compression ratio of the engine and its function to determine the cylinder condition. Another important parameter is the piston position in relation to the crankshaft angle, since the position of the piston in the cylinder liner for any crank angle depends upon the crank position and the connecting rod-to-crank ratio.

This is an important parameter when judging the cylinder liner wear profile, if abnormal wear starts to occur. In addition, we can determine the liner area that is exposed to the flame front if any abnormality occurs with the combustion cycle (measured heat release diagram). We can find the compression ratio using the following expression:

\[ \varepsilon = \sqrt[\frac{P_2}{P_1}] \]

Furthermore, we can find the piston position for any crank angle by the following equation of motion:

**Piston Displacement**

\[ S = r \left( 1 - \cos x \right) + \frac{r}{4L} \left( 1 - \cos 2x \right) \]

First Derivative

**Piston Velocity**

\[ v = \frac{d}{dt} x \times \frac{d}{dx} x \]

\[ \frac{dx}{dt} = r \sin x + \frac{r}{2L} \left( \sin 2x \right) \]

\[ \frac{dx}{dt} = 2\pi N \text{ (rad per min)} \]

Constant for velocity

\[ \frac{2\pi}{60} = 0.1047 \]

\[ V = 0.1047Nr \left( \sin x + \frac{r}{2L} \left( \sin 2x \right) \right) \]

Second Derivative

**Piston Acceleration**

\[ a = \frac{d}{dt} v \times \frac{d}{dx} v \]

\[ \frac{dv}{dt} = \frac{2\pi N}{60} \left( \cos x + \frac{r}{L} \left( \cos 2x \right) \right) \]

\[ \frac{dv}{dt} = \frac{2\pi N}{60} \text{ (rad per sec)} \]

Constant for acceleration

\[ \frac{2\pi}{60} \times \frac{2\pi}{60} = 0.01096 \]

\[ a = 0.01096N^2r \left( \cos x + \frac{r}{L} \left( \cos 2x \right) \right) \]
Based on the sea trial data for the engine in question, which commenced December 26th, 1997, we find the following results in regard to the ratio:

\[
\text{Ratio} = \frac{P_{\text{comp}}}{P_{\text{scav}}}
\]

\[P_{\text{comp}} = 120 \text{ bar} \quad P_{\text{scav}} = 2.32 \text{ bar} \quad \text{Ambient Pressure} = 1.006 \text{ bar}\]

\[P_{\text{comp,abs}} = 120 + 1.006 = 121.006 \text{ bar}\]

\[P_{\text{scav,abs}} = 2.32 + 1.006 = 3.326 \text{ bar}\]

\[\text{Ratio} = \frac{121.006}{3.326} = 36.38\]

This ratio is achieved with a new engine and becomes very important when judging the cylinder condition as the engine continues its service life. Now we have reached the last engine performance report taken by the engine crew November 10th, 2004 and the following is the result:

**Cylinder Condition:**

Compression pressure ratio on November 10th, 2004

<table>
<thead>
<tr>
<th>Cylinder #1</th>
<th>27.4</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cylinder #2</td>
<td>29.2</td>
</tr>
<tr>
<td>Cylinder #3</td>
<td>28.8</td>
</tr>
<tr>
<td>Cylinder #4</td>
<td>28.1</td>
</tr>
<tr>
<td>Cylinder #5</td>
<td>28.5</td>
</tr>
<tr>
<td>Cylinder #6</td>
<td>29.2</td>
</tr>
<tr>
<td><strong>Average</strong></td>
<td><strong>28.5</strong></td>
</tr>
</tbody>
</table>

**Booster Pressure** = 1.86  
**Atmospheric Pressure** = 1.02 bar

\[P_{\text{comp}} - \text{Cylinder #1} = 27.44(1.86+1.02) = 79.027 - 1.02 = 78 \text{ bar}\]

\[P_{\text{comp}} - \text{Cylinder #2} = 29.17(1.86+1.02) = 84 - 1.02 = 82.99 \text{ bar}\]

\[P_{\text{comp}} - \text{Cylinder #3} = 28.8(1.86+1.02) = 82.94 - 1.02 = 81.92 \text{ bar}\]

\[P_{\text{comp}} - \text{Cylinder #4} = 28.13(1.86+1.02) = 81.01 - 1.02 = 79.99 \text{ bar}\]

\[P_{\text{comp}} - \text{Cylinder #5} = 28.48(1.86+1.02) = 82.02 - 1.02 = 81 \text{ bar}\]

\[P_{\text{comp}} - \text{Cylinder #6} = 29.17(1.86+1.02) = 84 - 1.02 = 82.99 \text{ bar}\]

**Expected** \(P_{\text{comp}}\) **value with good cylinder liner and piston ring condition:**

\[P_{\text{comp}} = 36(1.86+1.02) = 103.68 - 1.02 = 102.66 \text{ bar}\]

**Average** \(P_{\text{comp}}\) **value** = 81 bar
As can be seen from these results the condition of all cylinders has deteriorated over time. The sad part of this is the fact that with monitoring and proper inspections, this could have been avoided.

What effect will this have on the engine in question? The following two parameters calculated and presented in the following graphs will play a major role in emission control.

![Fig. 6 Loss of Compression Air Pressure Over Time](image)

Loss of Compression Air Pressure Over Time

![Fig. 7 Loss of Compression Air Temperature Over Time](image)

Loss of Compression Air Temperature Over Time

Due to the loss of compression ratio, the consequence is lower compression pressure and consequent $P_{\text{max}}$ pressure. This will have a negative effect on the crankpin turning effort (tangential force) based on the following expression:

$$T = (P + P_0) \sin \Theta \left(1 + \frac{\lambda^2 \cos \Theta}{\sqrt{1 - \lambda^2 \sin^2 \Theta}}\right)$$

- $P$ = gas force
- $P_0$ = reciprocating mass force
- $\lambda$ = connecting rod ratio $R/L$

As seen, there are two contributions to the tangential effort, a force from the cylinder gas force and an inertia force from the reciprocating mass.
Based on the above and last taken engine performance readings, we see the following:

<table>
<thead>
<tr>
<th>Cyl</th>
<th>#1</th>
<th>#2</th>
<th>#3</th>
<th>#4</th>
<th>#5</th>
<th>#6</th>
</tr>
</thead>
<tbody>
<tr>
<td>P\text{max}</td>
<td>123</td>
<td>132</td>
<td>131</td>
<td>130</td>
<td>123</td>
<td>130</td>
</tr>
<tr>
<td>P\text{comp}</td>
<td>78</td>
<td>83</td>
<td>82</td>
<td>80</td>
<td>81</td>
<td>83</td>
</tr>
<tr>
<td>P\text{max} - P\text{comp}</td>
<td>45</td>
<td>49</td>
<td>49</td>
<td>50</td>
<td>42</td>
<td>47</td>
</tr>
</tbody>
</table>

Engine rpm = 86  
T/c speed = 12,200  
Scav. air pressure = 1.86 bar  
\( P_{\text{atm}} = 1.02 \text{ bar} \)

Based on these readings we can clearly see that the existing cylinder condition on all cylinders are in very poor condition, this is based on the fact that the achieved compression pressure reached is only 81 bar average, with good cylinder conditions, it should have been 103 bar average, based on the following:

\[
P_{\text{comp}} = 36.38(1.86 + 1.02) = 104.8 - 1.02 = 103.8 \text{ bar}
\]

Furthermore, an increase in the pressure difference between \( P_{\text{max}} - P_{\text{comp}} \) have occurred, this pressure difference should be below 35 bar, but the measurement shows an average of 47 bar pressure difference. This unacceptably high pressure rise will expose the piston rings, particularly the top piston ring, to very high pressure shock resulting in ring collapse and subsequent ring sticking and broken piston rings, allowing the combustion gas to bypass the rings by that increasing the thermal load of the liners.

This large increase in the pressure difference is very often the result of inexpedient re-adjustment of the \( P_{\text{max}} \) pressure with the VIT timing when lower values have occurred. Particularly with a reduction in compression pressure due to certain abnormal parameters that have occurred, as in this case with very poor cylinder conditions.

By adjusting the \( P_{\text{max}} \) pressure (increase pressure), this can only be achieved by advancing the injection timing (port closing point), when this occurs it will reduce the crankpin turning effort (tangential force) since the \( P_{\text{max}} \) peak pressure will occur closer to the TDC position. At the TDC position the turning effort is zero. In order for the engine to maintain its set speed with a reduced crankpin turning effort the engine will ask for more fuel since the governor only senses speed it will admit more fuel in order to maintain the set engine speed. This can clearly be seen in the plotted graph where the fuel rack position has been plotted in regards to the test stand results (nominal propeller curve), sea trial data with the actual propeller curve (lighter propeller) and at the last performance readings taken November 10th, 2004:

\[\text{Fig. 10 Fuel Pump Index Sheet}\]

It can clearly be seen that at 78% engine load, the engine is asking for nearly a 100% fueling, compared with the test stand results, but a lot more when we compare it to the sea trial results. This will definitely offset the air-to-fuel ratio in the cylinders and increase the thermal load of the liners in particular.

If we now consider the ratio between the absolute compression pressure and absolute scavenging pressure, with the bad cylinder condition as explained earlier in this paper, we find the ratio
for cylinder #1 to be 27.4 and cylinder #6 to be 29.2. During a piston ring inspection we took pictures of the existing piston ring conditions, and we have chosen the cylinder with the worst and best existing ratios. As can be seen for cylinder #1 the two top rings are broken and the other two rings have completely collapsed. For cylinder #6 we found the top ring intact, but signs that ring collapse has occurred in #2 and #3. As can be seen from these results, it follows that our evaluation is accurate enough that if these parameters would have been monitored and plotted over time, and inspected with reasonable frequency, these problems could have been avoided.

CASE STUDY #2: Engines Properly Maintained With Performance Analysis and Regular Inspections.

Here is an example of a fleet of ships with a MAN B&W 7K80MC-C engine. Here the engines were well taken care of, by introducing engine performance monitoring program with regular inspection and monitoring of the critical aspects of the diesel engine. As can be seen from the graphs the graphs below, there is hardly any loss of compression pressure and compression temperature over time. This show that there is hardly any wear in the cylinder liner and piston rings. Routine visual inspection of the cylinder liner and piston rings from the scavenge air ports confirmed this fact. The following pictures show the excellent condition of piston rings as found in one of our inspections.
Fig. 13 Minimal Loss of Compression Air Pressure Over Time

Fig. 14 Minimal Loss of Compression Air Temperature Over Time

Fig. 15 Excellent Piston Ring Condition After 17,428 Service Hours

Fig. 16 Excellent Piston Ring Condition After 17,428 Service Hours
The second part of the diesel engine performance monitoring program is cylinder pressure and firing characteristic monitoring, and fuel injection pressure monitoring. Gone are the days when the engine room crew used draw cards and firing pressure gauges to measure cylinder pressures. Today, we use modern electronic pressure gauges that can give us much accurate information about compression pressure, firing pressure, the crank angle at which maximum pressure occurs in the cylinder, the expansion pressure, etc and can automatically calculate mean indicated pressures and indicated power of each cylinder. This also gives us an insight into the fuel injection timing and the combustion characteristics of each cylinder.

Cylinder pressure measurement performed with the electronic pressure gauges gives us a lot of useful information about the cylinder condition and the overall health of the engine. These measurements can be overlaid on their respective values calculated from the engine manufacturer’s data and this can reveal how much the engine has deviated from its theoretical performance. The vessel owners can then set a limit for the deviation. Following are some of the example of how easy to understand, these graphs can be for a ship owner and the engine room crew. Graphical representation also invokes interest in the engine room crew who are normally scared off by numbers and equations. These graphs have proved excellent tools in educating and training the engine room crew. The information gathered by these electronic instruments helps us determine the characteristic ratios of the engine like the ratio between the firing pressure and the mean indicated pressure, the ratio between the compression pressure and the scavenging air pressure, and also the ratio between the firing pressure and the compression pressures. These ratios can be effectively used to evaluate the performance of the engine and they also point out towards the problems, if any. As compared to the draw cards and firing pressure gauges, these electronic pressure gauges also accurately measures the combustion timings in terms of crank angle like the time of ignition and the time when maximum pressure occurs. This information is necessary to locate any problems in the fuel injection system and the VIT system and can be used to fine-tune the fuel injectors and VIT. The complexity of a modern diesel engine not only warrants regular monitoring of the combustion chamber but also of the systems like the fuel injection system, the VIT, and the turbocharger.

The Alpha pressure curve clearly shows where the firing occurs in each cylinder. The area enclosed by the curve up to the dotted green peak (A2) is pure compression and the area enclosed by the curve above the dotted green line (A1) is the heat released from the fuel. The ratio between A1/A2 is a characteristic of a diesel engine and it can be used to plot a trend over time, which gives us an idea of the condition of the combustion chamber and the overall engine. Equally important is the ratio between the compression pressure and the scavenging air pressure.

![Alpha Pressure Curve](image1.png)
The torque on the crank pin also plays a significant role in maintaining proper air/fuel ratio inside the cylinders. As an example, Fig. 19 shows how the engine has been loosing torque per crank angle as indicated by the measured curve. The loss in torque transferred occurred because the crew had advanced the VIT. If there is a loss in torque transferred, the vessel speed tends to go down. It will cause the governor to demand for more fuel, thereby changing the air/fuel ratio in the combustion chamber. These changes in the air/fuel ratio over time leads to higher thermal stresses in the liner, higher wear in the liner and piston rings, and higher exhaust gas emissions. The torque transferred was later corrected by correcting the VIT and it is represented by the red dotted line.

CASE STUDY #3: Engine and Propeller Mismatch.

The required power-to-speed characteristics of a propulsion engine are governed by the performance of the propeller, and will therefore depend on whether a fixed or variable pitch propeller is used. In this case a fixed pitch propeller is used. The characteristics of the fixed pitch propeller are such that the power requirement increases with the cube of speed, and the torque (BMEP) increases with speed squared. It happens that the output characteristics of a turbocharged diesel engine are ideal for this application. Consequently, the turbocharger matching becomes a case of optimism rather than a compromise.

In this case we are dealing with a 9-cylinder inline engine, equipped with a three-pulse turbocharging system. The three-pulse system is by far the best system for transient response at large load changes. The propulsion system was installed in a series of harbor tugboats and heavy black smoke was emitted when the vessel was operating in bollard pull condition assisting larger vessels in port.
The engine is rated at 1,485 kW at 1,000 rpm. Based on this, we calculated the nominal propeller curve for the installation in question (blue line in the following diagram). We then used our engine monitoring system on a sea trial in bollard pull condition and found that the engines were operating at the torque limit (red line in the following diagram). Operating at the torque limit is also where the turbocharger can no longer supply adequate air to the engine, and the consequence is very black smoke released through the exhaust system. How can such a mismatch occur in today’s technical world even more disturbing is how many people overlooked this problem in the design phase of this project.

![Engine Power based on Nominal Propeller Curve](image1)

**Summary and Recommendations:**

During engine operation, several basic parameters need to be checked and evaluated at regular intervals. The purpose is to follow alterations in:

- The combustion conditions
- The general cylinder condition
- The general engine condition

- in order to discover any operational disturbances.

This enables the necessary precautions to be taken at an early stage, to prevent the further development of trouble. Careful monitoring will also make it possible to predict any decrease in performance and deterioration of the condition of the engine.

Furthermore, checking the essential parameters will ensure optimal mechanical condition of the engine components, and optimum overall plant efficiency.

On all plants today, temperatures and pressures are continuously monitored to ensure adequate flows and to avoid excessive temperatures. Signals are fed into a safety system which protects
the engine by giving commands for “Slow down” and “Shut down” of the engine.

In the absence of a permanently installed cylinder pressure monitoring device, the cylinder pressures shall be taken every two weeks in order to check the timing and injection and the tightness of the piston rings pack.

When evaluating the performance and condition of the engine, one has to take into account the influences caused by each individual parameter. Also, in order to be able to compare the results on a time line, the values are all converted to ISO standard ambient conditions. This also makes it easier to predict the scheduling of planned overhauls.

**Emissions:**

Great strides have been made by the designers of diesel engines over the last decade to reduce the harmful emissions from the exhaust gas. Many of these reductions have been made without seriously jeopardizing the fuel oil consumption of the engines. Upcoming legislation around the world will intensify the pressure for overall reductions SOx (acid rain).

As an example, California’s CARB (California Air Resources Board) has released new regulations imposing emission limits and requirements for auxiliary diesel engines on ocean going ships in “Regulated California Waters.” These provisions, which became effective on January 1, 2007 apply to US and foreign ocean going ships in California waters and out to 24 miles offshore. Covered vessels must utilize either marine gas oil or marine diesel oil at or below 0.5% sulfur content. By 2010, the maximum allowable sulfur content is reduced to 0.1%.

As the saying goes, what is “today” in California will be “tomorrow” everywhere else and it is therefore safe to assume that emission restrictions will become more onerous in the coming years.

More and more of the cruise ships arriving in the US west coast ports are having receptacles and transformers installed to enable the vessel to be “cold-ironed,” i.e. accept shore-based power while in port.

IMO (International Maritime Organization) issued regulations for the Prevention of Pollution from ships, which were adopted in 1979 Protocol to MARPOL 73/78 and which are included in Annex VI of the Convention. The Protocol entered into force on May 19, 2005. The rules on NOx emission are specified in Conference Resolution 2, the NOx Technical Code.

Annex VI also defines the requirements for a “Unified Technical File” for the main engine in question. This document which specifies procedure, mainly based on performance measurements, by which the operator can verify compliance with the “IMO NOx Technical Code” to the Flag State Authority (or their representative) when the engine is later checked in service.

As a final statement with the cases presented in this paper and other cases studied in the past two decades gives us a very good indication that most emission problems occur due to poorly maintained engines, and when an engine is well maintained, it reduces the emissions quite significantly.

**References:**


Turbocharging the Internal Combustion Engine by N. Watson and M. S. Janota (1982)
Diesel Motor Ships Engines and Machinery by *Christen Knak* (1979)

Vibration Control In Ships
*Veritec Marine Technology Consultants* (1985)