Prediction of Marine Diesel Engine Combustion Characteristic in Transient Condition By Using Seiliger Process Cycle

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Abstract. Modern diesel engine cycle can be predicted by applying variety of simulation methods. These models are used for optimization of engine parameters, investigation of control strategies or evaluation phenomena that are difficult to measure. In practice, using engine cycle simulation methods, the number of expensive experiments studies could be reduced and very useful result could be obtained for prediction of engine design parameters in a short period of time. Furthermore, these models are also used to test different solutions (technologies) or control strategies which can help to reduce fuel consumption, increase power output, reduce emissions to meet emission requirement. In diesel engine modeling process, in cylinder pressure measurement is the fundamental method to investigate the combustion process and an effective way to get insight into the combustion phenomena. The main objective of the paper is focusing on the transient modeling of marine two stroke diesel engine and calculating (prediction) the parameter of the cylinder process model by using the Seiliger cycle process. The research approach is carried out using data from a MAN-10L90MC diesel engine and mathematical model of the test engine is calculated by using Microsoft Visual 6.0 VC++ programming.

1. Introduction
Today Diesel Engine has been used in different forms and different standards, starting from normal household car to heavy truck, electricity production plant to marine sea going vessels. The most attractive feature of the compression ignition diesel engine is its excellent fuel efficiency, which can surpass 40% in vehicular application and even 50% in large two stroke units of marine propulsion. Due to the above reason, Diesel engine has been used in marine industry as Main engine for many years. The marine industry has always favored using diesel engines for its benefits mentioned.
However, the regulation is still revolving around the benefits of the players in the industry and is lagging behind other industries. Recently strong regulation has entered into force in attempt to reduce the emission of NOx and SOx by IMO (International Maritime Organization). Especially, emission from transient load conditions are not yet on the table of discussion yet but it is obvious to predict that it is the matter of time in a relatively short term that more strict legislation will come into force to parallelize with other industries. In the other hand, IMO’s day by day very strict rules and regulations leads the requirement in improvement of marine diesel engine in very important condition. So it is same as to do experiment in according to comply with MARPOL and to do more in combustion process of Diesel engine to reduce NO\textsubscript{x}, SO\textsubscript{x} gases and to know better about combustion inside the cylinder. Therefore to overcome this kind of difficulties is by using high technology of modern era high performance computer and it associated programs. There are a lot of advantages by using high performance computers such as saving a lot of money and time, can observe diesel engine’s phenomenon in very comfortable way, and get more accuracy and accurate data close to real life Engine. By this way, we can achieve the new modern engines which are complying with IMO’s newest rules and regulations.

1.1. Literature review
A number of thermodynamic models of marine diesel engines have been developed for various applications due to the uniqueness of marine diesel engines and their operation and the computer programs for marine applications must be specifically designed, requiring that each application requires a different model. Miedema and Lu (2002) modeled the diesel engine based on the Seiliger (thermodynamic) process. The result of the model shows that the diesel engine behaves like a second order system when operating governor area and more like a first order system in the constant torque (overload) area [1]. Ritzen (2003) introduced a four state mean value model, to make the model usable and simple it was simulated with a fixed step size solver. Chesse (2004) presents a marine diesel engine simulation code designed for real time performance. The various equations are derived from the laws of thermodynamic, thus guaranteeing qualitatively accurate predictions and allowing for easy use with any type of engine. The code is based on the “filling and emptying” method with various simplifications to achieve real time performance. Benvenuto (2005) has applied simulation techniques to marine propulsion systems obtaining interesting results. The transient operation of the main components of marine propulsion plants and their different elements such as engine, shaft lines, clutches, gears, bearings and elastic couplings, have been studied and simulated according to the operational requirements of the ship. Theotokatos (2007) carried out the mapping of the performance and emission parameters of a merchant vessel propulsion system over the ship operating envelope by using a cycle mean value engine model. Yu Ding (2011) investigated the 4 stroke engine combustion process and heat release measurement with Seiliger process and Vibe heat release model [2].

2. Modeling the in-cylinder process with Seiliger model

2.1. Model description and measurement description
The Seiliger process is a simple method to model the in cylinder process of diesel engine. It was use to explore the future of diesel engine in earlier years. By given the trapped pressure and temperature it is capable of catching the main features of the cylinder process, i.e. heat input, indicated work output,
maximum pressure and temperature. The Seiliger parameters are applied in a mean value simulation model as the combustion parameters to give a global description of the combustion process. The marine diesel engine in cylinder process can be parameterized by finite stages in different kinds of Seiliger process, providing a simple and reliable method for mean value modeling of the cylinder process. Often the 5 points Seiliger process with two combustion stage is used but in order to model combustion adequately is deemed necessary here [3].

2.2 Six point Seiliger cycle

Figure 1 shows the six points Seiliger process model. The stages can be described as follows;
1-2: polytropic compression, 2-3: isochoric combustion, 3-4: isobaric combustion, 4-5: isothermal combustion and expansion, 5-6: polytropic expansion indicating a net heat input caused by late combustion during expansion (advanced seiliger cycle). Stage (6-1) is assumed that the exhaust gas is directly replaced by intake air. In the actual simulation model, the Seiliger process breaks off in point 6, followed by the gas exchange model. [3].

![Figure 1. Six Point Seiliger cycle](image)

2.3 Parameterization of the in-cylinder process

The Seiliger process can be described by a finite number of parameters which fully specify the process together with the initial (trapped) condition and the working medium properties. The definition of the Seiliger stages and the Seiliger parameters are given in Table 1.

| Seiliger State | Work $W [J]$ |
|---------------|--------------|
| 1-2 | $W_{12} = \frac{m_{12} R_{12}}{n_{comp}} * (T_2 - T_1)$ |
| 2-3 | There is no work done between these two points. |
| 3-4 | $W_{34} = m_{34} R_{34} * (T_4 - T_3)$ |
| 4-5 | $W_{45} = m_{45} R_{45} T_4 \log \left( \frac{V_5}{V_4} \right)$ |
Once all the Seiliger parameters are obtained, the pressures, temperatures and work done in the various stages of the Seiliger cycle can be calculated according to following Table 1 and Table 2. The polytropic exponents $n_{comp}$ and $n_{exp}$ play an important role in determining the stage 1-2 and 5-6, so they have to be chosen according to experience. [4]

### Table 2 Seiliger process definition and parameters

| Seiliger Stage | Seiliger Definition | Parameter Definition | Parameter Definition | Seiliger parameter |
|----------------|---------------------|----------------------|----------------------|-------------------|
| 1-2            | $P_2 = P_1 r_c^{n_{exp}}$ | $V_1 = V_2$ | $T_2 = T_1 r_c^{n_{exp} - 1}$ | $n_{comp}, r_c$ |
| 2-3            | $P_2 = P_1 a$ | $V_2 = V_3$ | $T_2 = T_3 a$ | $a$ |
| 3-4            | $P_3 = P_2 b$ | $V_3 = V_4$ | $T_3 = T_4 b$ | $b$ |
| 4-5            | $P_4 = P_3 c$ | $V_4 = V_5$ | $T_4 = T_5 c$ | $c$ |
| 5-6            | $P_5 = P_4 r_c^{n_{exp}}$ | $V_5 = V_6$ | $T_5 = T_6 r_c^{n_{exp} - 1}$ | $n_{exp}, r_c$ |

#### 3. Experimental investigation of the Seiliger combustion parameters

In this section, the investigation of the Seiliger combustion parameters is carried out which are based on the marine diesel engine measurement. These parameters can be calculated based on the test data of MAN 10L90MC Engine. The main specifications of the MAN 10L90MC Engine are shown in following Table 3.

### Table 3 Main specification of marine diesel engine MAN10L90MC

| Parameter                        | Unit | Value |
|----------------------------------|------|-------|
| Nominal Engine speed             | rpm  | 84.0  |
| Maximum effective power          | KW   | 43430 |
| Cylinder Number                  |      | 10    |
| Bore                             | m    | 0.9   |
| Stroke                           | m    | 2.9   |
| Compression ratio                |      | 16.8  |
| Connecting rod length            | m    | 3.5   |
| Exhaust valve open               | deg  | 114   |

#### 3.1 Engine Combustion parameters simulation

As described above Marine Diesel engine’s combustion is a complicated process and it have to consider several facts and factors from different point of view. To simulate the dynamic characteristics of the diesel engine, several engine parameters must be considered and used. Diesel
engines operate with an air excess ratio and it is an important parameter for combustion, engine power, specific fuel consumption, and emissions. It indicates the ratio of amount of scavenge air to amount of actual air for combustion [5-6].

\[ \infty = \frac{m_{in}}{m_f \lambda \phi_s} \]  

\( \alpha \) = excess air ratio, \( \phi_s \) = coefficient of scavenging, \( m_{in} \) = air mass flow, \( \lambda = 14.3 \) for complete combustion of 1 kg of fuel, \( m_f = \) fuel mass flow

From above equation, the air fuel ratio Lambda is obtained.

\[ \lambda = \frac{m_{in}}{m_f \infty \phi_s} \]  

Calculation of fuel mass flow,

\[ m_f = \frac{m_N \text{cyl} n_e}{60 N_{st}} \]  

\( N_{cyl} = \) number of cylinder, \( n_e = \) engine rpm, \( N_{st} = \) number of stroke, 1 for 2 stroke engine and 2 for 4 stroke engine, \( m = \) mass of fuel

The ignition delay of fuel spray is important from the viewpoint of preparing the fuel before injecting into the engine as well as selecting optimum injection.[7].

\[ \tau_f = c_1 P_g^{-c_2} \exp \left( \frac{E_a}{R_{mol} T_g} \right) \phi^{-c_3} \]  

\( \phi = \) Fuel air equivalence ratio, \( P_g = \) integrated mean gas pressure, \( T_g = \) integrated mean gas temperature, \( R_{mol} = \) the universal gas constant, \( c_1 = \) Arrehenius exponent (17.68ms), \( c_2, c_3 = \) constant depending on the fuel and injection characteristic (1.7, 3.402), \( E_a = B/CN +25 \), \( B = \) a constant (618,840-1310,000), \( CN = \) cetane number (45……50)

Specific heat at constant volume is noted as \( c_v \) and specific heat at constant pressure is noted as \( c_p \) as well. It can be defined as follows, In addition to the above-cited quantities; a few other characteristic quantities are used [8-9].

\[ \text{specific heat ratio} = \frac{c_p}{c_v} \]  

In addition to the above-cited quantities, a few other characteristic quantities are used in modeling of diesel engine which are as follows. The average piston speed is a characteristic speed for internal combustion engines which are consider in calculation of heat coefficient.

\[ \text{piston speed} = \frac{2Ln_e}{60} \]  

The compression ratio is the total cylinder volume in reference to the compression volume.

\[ r = 1 + \frac{V_d}{V_c} \]
3.2 Seiliger process calculation and combustion parameter calculation

In any given case, the trapping volume, \( V \), is a constant. This is also true for the gas constant, \( R \), for gas at the prevailing gas composition at the trapping point. The gas constant for exhaust gas, \( R_{ex} \), is almost identical to the value for air, \( R_a \). Because the cylinder gas composition is usually mostly air, the treatment of \( R \) as being equal to \( R_a \) invokes little errors. For any one trapping process, over a wide variety of scavenging behavior, the value of trapping temperature, \( T_{tr} \), would rarely change by 5%. Therefore, it is the value of trapping pressure, \( P_{tr} \), that is significant variable in consideration of trapping mass. The net effect of the cylinder scavenging process is to fill the cylinder with a mass of air, \( m_a \), with in a total mass of charge, \( m_{tr} \), at the trapping point. [10].

Basically, two thermodynamic laws are used in cycle analysis, the ideal gas law and the polytropic compression law. The mass of gas in a combustion chamber remains essentially constant during the compression process. If the temperature also remained constant, Equation (9) indicates that the pressure would vary inversely with the volume. In an engine, the temperature increases considerably during the compression stroke; therefore, the pressure increases more than in an isothermal process. The initial pressure and temperature of the process represent (\( P_1 \) and \( T_1 \)) a scavenge air pressure and temperature of the engine inlet which can be referred from engine manual. The volume at trapped condition is \( V_1 \) can be calculated by using the crank angle (\( \alpha \)), assuming a constant engine speed to calculate the crank angle in following equation (16) [11];

\[
V(\infty) = A_h L_s \left[ \frac{1}{r - 1} + \frac{1}{2} (1 + \cos \infty) + \frac{1}{\lambda} \left( \sqrt{1 - \lambda^2} \cdot \sin^2 \infty \right) \right]
\] (8)

After obtaining the all Seiliger parameters and pressures, temperatures of the respective states, the work done produced for any given engine cycle and the effective pressure of the simulated diesel engine can be calculated as follows.

\[
W_i = \int_0^{360} P_{cyl}(\infty) dV(\infty)
\] (9)

\[
P_e = \eta_m \cdot P_i(\infty)
\] (10)

Finally Power of the marine diesel engine is simply calculated for N rpm and n cylinder number.

\[
\text{Engine Power} = P_e L A N n
\] (11)

4. Result of the simulation

The result compared with the bench test data shows that the model can reflect the actual system’s steady-state and dynamic characteristics correctly. Following figures show the comparison of actual result of the test engine and simulation result of the Seiliger process. Also the important characteristics in combustion of marine diesel engine such as maximum pressure, power, and specific fuel consumption and compression pressure are shown in Table (4) and (5) with errors compare with the actual ship trial test results. From which we can said that the errors are less than 5% and which are within the acceptable region. It is clear that the diesel engine combustion process is extremely complex and some important consequences can be note down. A higher engine compression ratio can be used in the diesel engine which improving its fuel conversion efficiency and improving the efficiency of diesel engines at higher loads. The delay period between the start of injection and start of
combustion must be kept short. By varying the amount of fuel injected per cycle with the engine airflow essentially unchanged which cause lower pumping work and improving the engine part-load mechanical efficiency. Advancing injection timing or increasing injection pressure improves combustion efficiency raises combustion temperature which leads to higher NOx formation. Also a rise in inlet temperature results in a rise of output temperature as we expected. But inlet temperature has a large effect on lambda which values increases when decreasing inlet temperature. As conclude, the paper simulated working process of marine diesel engine by combining diesel engine and Seiliger process model which can reflect not only some average parameters but also important parameters in real time with high performance and less error.

Figure 2. Power Comparison

Figure 3. Pmax or Combustion pressure Comparison

Figure 4. Pcomp Compression pressure Comparison
Figure 5. Simulation Result of Combustion Temperature and Compression Temperature

Figure 6. Simulation Result of Mean Indicated Pressure

Table 4. Simulation and Test Bed Result Error

| Load % | RPM | Power (KW) | Pmax (MPA) |
|--------|-----|------------|------------|
|        | Test | Simulation | Error%     | Test | Simulation | Error%     |
| 25     | 51.8 | 10976      | 10763.62   | 1.934912 | 8.81       | 6.169      | 4.9460708 |
| 50     | 65.2 | 21786      | 20935.95   | 3.901822 | 10.76      | 8.483      | 3.7116912 |
| 70.4   | 72.8 | 30413      | 29896.8    | 1.697300 | 11.84      | 11.139     | 3.5223048 |
| 75.4   | 74.6 | 32493      | 32304.33   | 0.580532 | 13.92      | 11.56      | 2.3648648 |
| 90.8   | 79   | 39122      | 38485.67   | 1.626544 | 13.86      | 13.88      | 0.2514367 |
| 100.8  | 84.6 | 43432      | 43043.77   | 0.89387  | 13.918     | 13.92      | 0.4329004 |

Table 5. Simulation and Test Bed Result Error

| Load % | RPM | Pcomp | SFOC |
|--------|-----|-------|------|
|        | Test | Simulation | Error% | Test | Simulation | Error% |
| 25     | 51.8 | 7.2    | 4.425 | 4.838709 | 189.5 | 193.23 | 1.968337 |
| 50     | 65.2 | 9.47   | 7.08  | 1.666666 | 180   | 187.313 | 4.062777 |
5. Conclusion

The simulation is program is now link with other simulator program and provide as teaching assistant tool. The more research have to be done in simulating modern diesel engine with late injection timing, high pressure injection equipment, and even common rail system, with the simulation models which are based on the Seligler model as basic principle. Finally the combustion model is generally at the center of interest in the marine diesel engine model literature and it is the most complex phenomena occurring during the engine cycle. As conclude, the developed simulated mathematical model of the marine diesel engine is good compromise for students who are engaged in the new generation of the marine diesel engine development.

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Acknowledgments
I would like to acknowledge with gratitude, support and kind of my supervisor Prof. Sun Jian Bo and a number of my friends and colleagues in encouraging me to start the work, preserve with it and finally to publish it. Finally I would like to show my grateful feeling to China Scholarship Council (CSC) who awarded the scholarship to a huge number of foreign students and Dalian Maritime University who give me enormous knowledge in my marine engineering professional.