Evaluation analysis of scavenging process of two-stroke marine diesel engine by experiment and simulation

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Abstract
The performance of the two-stroke engine depends on the scavenging process, which affects air-fuel mixture and combustion. This paper presents results from experiments in which the timing of exhaust valve opening and closing was varied in two-stroke marine diesel engine during shop testing, and reveals the effect on performance and emission, as well as the corresponding concept analysis. A simulation model was then developed based on AVL BOOST and experimental results, its validity was confirmed by actual measurements. This simulation enables a more thorough investigation into engine performance features and the scavenging process. In particular it provides a detailed examination of how changes in EVO and EVC timing impact the scavenging process and ultimately engine performance. These results are summarized, and based on these results, optimal EVO and EVC settings are suggested for balancing the scavenging process, and also the engine performance and NOx emissions.

Key words: Two-stroke marine diesel engine, Scavenging process, Exhaust valve opening and closing, Fuel consumption, NOx emissions

1. Introduction
Two-stroke diesel engines are widely used for propulsion of commercial vessels, especially large bulk carriers, container ships, and tankers. Due to the increasingly strict environmental protection regulations of the IMO (International Maritime Organization), more attention has been given to the research and development of two-stroke marine diesel engine in order to achieve high efficiency of combustion, low fuel consumption, and importantly, ultralow NOx emissions.

Since the 1980s, most two-stroke marine diesel engines have used port-valve systems in scavenging process. This scavenging system is known as uniflow type, and it has been described at length, including testing and calculations, by Gordon P. Blair in his book entitled Design and Simulation of Two-Stroke Engine. This paper focuses on the uniflow scavenging two-stroke marine diesel engine. Compared to the four-stroke engine, the scavenging process of two-stroke engine is far less optimized. The four-stroke diesel engine possesses separate intake air and exhaust gas, and therefore has enough time to perfect the scavenging process. In contrast, the intake air and exhaust gas of the two-stroke diesel engine participate in compression and expansion of engine respectively, and between these working processes, exhausted air is scavenged with the fresh intake cylinder charge, and thus the time available to optimize this mixed working/scavenging process is very limited.

Despite the limited time available for the scavenging process to occur, optimal scavenging is critical to the performance of two-stroke engine as it controls the quality and quantity of charged air and the formation of the air-fuel mixture, which in turn impacts the combustion process, and ultimately engine performance and emission. Thus, an organized scavenging process that and yields as much fresh air as possible is essential for achieving optimal cylinder power, fuel consumption and emission. S. Scott Goldsborough optimized the scavenging system by simulating the two-stroke cycle and found that a stratified scavenging scheme employing a uniflow geometry and a stable, low temperature/pressure charge would produce optimal engine efficiency and emission profiles. However, the target of this simulation was the free piston engine, not a marine diesel engine. Hickory Zachariah Foudray developed a physical model to characterize scavenging and proposed that nitrogen oxide production is an indicator of scavenging efficiency. He showed that in a direct-injection two-stroke engine, scavenging efficiency is largely a function of delivery ratio and...
air/fuel ratio. Several other studies have used models to understand flow dynamics and the mechanics of the scavenging process. Kristian Mark Ingvorsen used a dynamic model and PIV Measurements to research turbulent swirling flow in a two-stroke uniflow-scavenged engine. Gerhard Regner studied the scavenging process in the opposed piston two-stroke diesel engine in Commercial Vehicle Applications. Finally, Jiang Guodong simulated unsteady flow in the two-stroke uniflow-scavenging process and used the ICE-ALE numerical method to model viscous fluid dynamics. Using this model, he produced detailed descriptions of time dependent in-cylinder flow fields and exhaust residual distributions. However, despite extensive research into the dynamics of the scavenging process, far less research is devoted to experiment based evaluation of the scavenging processes in the two-stroke marine diesel engine.

High costs, practical complications and limited application have caused research on the two-stroke marine diesel engine to lag behind automotive engines. Moreover, the size of the two-stroke marine engine, as well as the buildings and planning required to operate full-scale test facilities makes experiment-based research difficult.

In this paper, experiments are performed to evaluate the timing of Exhaust Valve Open (EVO) and Exhaust Valve Close (EVC) in a small-bore two-stroke marine diesel engine in order to gain more insight into the scavenging process and its key parameters. Then based on these measurements, simulate the scavenging process of the two-stroke marine diesel, and use the simulation to evaluate the scavenging process, and how modifications to the process improve engine performance and emission.

2. Experiment and Result Analysis

2.1 Background and Information

The research presented in this paper focuses on a typical, commercial small-bore two-stroke marine diesel engine with seven cylinders, a common rail fuel-injection system, and a port-valve uniflow scavenging system. Compressed and cooled air from the intake valve enters the combustion chamber through 27 scavenging air ports located in the lower side of the cylinder liner. Then exhaust is emitted from the combustion chamber into the main exhaust gas pipe via an exhaust gas valve located in the middle of the cylinder cover. This exhaust gas valve exists in every cylinder cover and it is driven by a hydraulic oil system and controlled electronically by a system integrated with the main engine control system so that it can be opened or closed as required. Each cylinder has two fuel injectors, located symmetrically on the periphery of the cylinder head.

The basic specification of the engine, the geometry section of scavenging system, and the sensor location are shown in Fig. 1.
In-cylinder pressure was measured using a Kistler piezoelectric pressure transducer coupled to a Kistler charge amplifier at 0.5° Crank Angle (CA). The sensor was installed in the indicator valve (located on cylinder cover) which connected with the combustion chamber (see Fig. 1 (b) Sensor Location for Cylinder Pressure and Fig. 1(c)). The scavenging air pressure was measured using a Kistler piezoelectric pressure transducer located in the bottom and underside of the cylinder block (see Fig. 1 (b) Sensor Location for Scavenging Air Pressure and Fig. 1(d)).

The temperatures before and after the turbines were measured by high-temperature thermocouple sensors with cable from Kongsberg Maritime. These temperature sensors possess a wide dynamic range, spanning from 0°C to 800°C. The locations of the temperature sensors are illustrated in Fig. 1(e). The temperature sensor located ‘before’ the turbine was placed at the end of the main exhaust pipe, connected to an expansion piece next to the turbine. The temperature sensor located ‘after’ the turbine was placed downstream of turbine and was about 0.3m to the turbine’s outlet.

NOx emission was measured by a specialized exhaust analyzer system from America named CAI600. The sample probe, sample position, and testing program used, were all in accordance with <<Marine Diesel Engine Nitrogenoxide Emission Control Technology>> and other related IMO Regulation.

During the experiment, the timing of EVO and EVC were varied and in-cylinder pressure, scavenging air pressure, temperature before turbine and after turbine, NOx emissions, etc., were measured, recorded and analyzed. In the following experimental analysis and investigation, 75% load of the target engine was considered as the operative and investigated load.

### 2.2 EVO Varied Experiment and Analysis

For the first set of experiments, the timing of Exhaust Valve Open (EVO) was varied while the engine ran at 75% load and the effective output power and engine speed were kept constant. The EVO parameter was set to 115°CA ATDC (Crank Angle) (EVO115) and 117°CA ATDC (EVO117) and the performance data were measured and recorded according to the following reference working conditions listed in Table 1.

<table>
<thead>
<tr>
<th>Table 1 Reference Working Condition during EVO Varied Experiment</th>
</tr>
</thead>
<tbody>
<tr>
<td>Common Rail Pressure(MPa)</td>
</tr>
<tr>
<td>---------------------------</td>
</tr>
<tr>
<td>EVO115</td>
</tr>
<tr>
<td>EVO117</td>
</tr>
</tbody>
</table>

Fig. 2 shows the comparison of measured parameters at EVO115 and EVO117. When EVO was varied from 115°CA to 117°CA, temperature before and after turbine, compression pressure, maximum combustion pressure and Brake Specific Fuel Consumption (BSFC) remained nearly constant, while NOx emissions increased slightly.

As shown in Fig. 3, the in-cylinder pressure at 115°CA and 117°CA were nearly coincident. When the cylinder pressure changes are magnified between 0.3 MPa and 0.5 MPa (Fig., 3b), it is observed that cylinder pressure decreases dramatically during the transition from Intake air Port Open (IPO) (138°CA) to Intake air Port Close (IPC) (222°CA), especially at the beginning of IPO.
At this point, the scavenging air pressure (Scav. Pre. Fig. 3b) was greater than cylinder pressure. After IPO, cylinder pressure and scavenging pressure appear to balance and then at IPC cylinder pressure decreases slightly again before increasing back above scavenging pressure.

The changes in cylinder pressure described above are tied to the air exhaust process occurring during the period from EVO to IPO. During this period, exhausted air is freely emitted. As exhaust is emitted, scavenged air flows into the cylinder, increasing the differential between scavenging air pressure and cylinder pressure at the start of IPO. Following IPO, the two pressures equilibrate until IPC when cylinder pressure dips slightly and then rises back above scavenging air pressure.

2.3 EVC Varied Experiment and Analysis

In the next set of experiments, the timing of EVC was varied while the engine was running at 75% load and the output power and engine speed were held constant. Performance data were according to the reference working condition listed in Table 2.

<table>
<thead>
<tr>
<th>EVC</th>
<th>Common Rail Pressure(MPa)</th>
<th>Start of Injection (°CA)</th>
<th>Injection Duration (°CA)</th>
<th>EVO (°CA)</th>
<th>EVC (°CA)</th>
<th>Scav.Air Pre. (MPa)</th>
<th>Injection Strategy</th>
</tr>
</thead>
<tbody>
<tr>
<td>EVC255</td>
<td>70</td>
<td>-4.65</td>
<td>21.60</td>
<td>117</td>
<td>255</td>
<td>0.369</td>
<td>Standard</td>
</tr>
<tr>
<td>EVC260</td>
<td>70</td>
<td>-4.65</td>
<td>21.60</td>
<td>117</td>
<td>260</td>
<td>0.370</td>
<td>Standard</td>
</tr>
<tr>
<td>EVC265</td>
<td>70</td>
<td>-4.65</td>
<td>21.60</td>
<td>117</td>
<td>265</td>
<td>0.368</td>
<td>Standard</td>
</tr>
</tbody>
</table>

Fig. 4 presents the comparisons between parameters measured when EVC occurred at different time. When EVC
was changed from 255°CA to 265°CA, the temperature before and after turbine were slightly elevated, particularly the before turbine temperature (Fig. 4a). In contrast, compression pressure and maximum combustion pressure exhibited a significant decrease (Fig. 4b). NOx emissions also decreased substantially while BSFC increased (Fig. 4c).

Fig. 5 shows the comparisons between in-cylinder pressure when EVC was varied from 255°CA to 265°CA. Cylinder pressure decreased normally, but then showed a more rapid and dramatic increase during combustion at EVC 255 compared to EVC265 (Fig. 5a). This more dramatic increase in pressure upon combustion also produced more engine noise and vibration.

For a closer examination of in-cylinder pressure during the transition from IPO to IPC, the cylinder pressure changes between 0.3 MPa and 0.5 MPa were magnified. Similar to the process illustrated in Fig. 3, cylinder pressure dramatically decreased at the beginning of IPO, then balanced with the scavenging air pressure and then dipped slightly after IPC before rising back above scavenging pressure. At EVC265, cylinder pressure equilibrated quickly with scavenging pressure, and reached the referential scavenging pressure line resulting in blowback of scavenged air. Thus, the scavenging process was cleaner and more efficient at EVC255 and EVC260, as shown in Fig. 5(b).

2.4. Discussion

Based on these experiments it can be concluded that, for a small-bore two-stroke marine diesel engine, varying EVO leads to a moderate impact on engine performance while varying EVC leads to a much greater impact. The timing of EVO influences scavenging air blowback, while EVC timing impacts compression. The work volume of the engine in relation to crank angle during operation can be described by the following equation:

\[
V = \frac{\pi D^2}{4} \left[ \frac{S}{\varepsilon - 1} + \frac{S}{2} \left( 1 + \frac{1}{\lambda} \right) - \cos \left( \frac{\pi \phi}{180} \right) - \frac{1}{\lambda} \sqrt{1 - \lambda \cdot \sin^2 \left( \frac{\pi \phi}{180} \right)} \right]
\]

(1)

Where \( V \) is the work volume, starting from crank angle \( \phi = 0 \), \( D \) is the cylinder bore,
$S$ is the piston stroke, $\lambda$ is the ratio between connect rod and crank web,
\[ \varepsilon \] is the compression ratio.

The effective compression ratio is defined as the ratio between work volume when the exhaust gas valve is closed and clearance volume and it is described by the following formula:
\[ \varepsilon = \frac{V}{V_c} \]  (2)

Where $V_c$ is engine clearance volume when the piston is at TDC.

When the EVC timing is changed from 255°CA to 265°CA, the engine’s effective work volume $V$ is reduced, and the compression ratio decreases accordingly. Furthermore, the delay in EVC (from 255°CA to 265°CA) leads to an increased possibility that exhausted air will reverse flow from the main exhaust pipe and thus a greater proportion of exhausted air will mix with working charge in the cylinder. Consequently, the cylinder temperature will increase due to heating by exhausted air, the cylinder pressure at the end of compression will be lower, combustion will be slow, and the maximum combustion pressure and maximum combustion temperature will decrease, resulting in reduced NOx emission and increased BSFC. As predicted, this conceptual analysis correlates well with our experimental results outlined above.

Additionally both our results and our conceptual analysis point to a potentially beneficial impact of delaying EVC on engine performance. When EVC is delayed, the compression process is reduced meaning that the power consumed during compression is reduced and the thermal efficiency of the engine is increased. Moreover, when EVC is delayed and exhausted air reverses into the cylinder, it creates a kind of internal Exhaust Gas Recirculation (EGR) in the two-stroke marine diesel engine that represents one potential approach to low NOx emissions.

3. Further Investigation by Simulation
3.1. Simulation Model Setup and Validation

In order to conduct a more thorough investigation of the internal scavenging process, an engine thermal cycle and gas exchange model was set up based on AVL BOOST, and quasi-dimensional and 2-zone combustion. A heat release model called AVL MCC was adopted, which considered the development of premixed combustion and diffusion combustion. The NOx emissions model was based on the Zeldovich mechanism. The soot formation model was based on Schubiger and the heat transfer model was based on Woschni 1990.

During simulation, EVO117 and EVC260 were considered as reference condition and input condition respectively (listed in Table 2), the deviations of performance parameters between simulation and measurement are shown in Fig. 6.

![Fig. 6 Deviations between Simulation and Measurement](image)

(a) Performance Parameter (b) Cylinder Pressure History

The results demonstrate that there was limited deviation in performance parameters between simulation and measurement (all within 1%). Also the calculated cylinder pressure during compression and combustion conformed almost exactly to that observed in the experiments above, with only a slight deviation around IPO, IPC and EVC due to fluctuations upon opening and closing of the intake port and exhaust valve. The agreement between measurement and simulation indicates that the simulation can be used for the further investigation and analysis.

3.2. Further Scavenging Process Investigation
3.2.1. EVO Varied Simulation Investigation

The strategies for varying EVO are listed in Table 3. Performance parameters and emission data were calculated and are shown in Fig. 7.

| EVO113 | 70  | -4.65 | 21.60 | 113  | 260  | 0.370 | Standard |
| EVO115 | 70  | -4.65 | 21.60 | 115  | 260  | 0.370 | Standard |
| EVO117 | 70  | -4.65 | 21.60 | 117  | 260  | 0.370 | Standard |
| EVO119 | 70  | -4.65 | 21.60 | 119  | 260  | 0.370 | Standard |

The simulation results closely followed the experimental results reported in section 2.2. As shown in Fig. 7, combustion pressure, effective output power, NO\textsubscript{x} emissions and volumetric efficiency increased whereas BFSC decreased when EVO was changed from EVO113 to EVO119. Soot decreased as EVO decreased, reaching the lowest value at EVO117.

As shown in Fig. 8(a), at EVO119, scavenged air was blown back at the beginning of IPO. As EVO decreased, blow back decreased until it eventually disappeared. Fig. 8(b) shows that at EVO113 and EVO115, exhaust gas was reversed into the cylinder at the start of IPO and IPC because the exhaust gas pressure was greater than cylinder pressure at these points. As EVO increased, exhaust gas reversal diminished.

Scavenging air pressure showed substantial fluctuation due to the geometry of the scavenging port and movement of the piston, and this fluctuation is a major difference between two-stroke and four stroke diesel engines.

The results demonstrate that when EVO occurs late, e.g. EVO119, there is blow back of scavenged air. However, when EVO occurs early, e.g. EVO113, reduces expansion work while exhaust gas reverses and mixes with freshly scavenged air. This reversal causes an increase in CO\textsubscript{2} content, a decrease in maximum combustion temperature, cylinder pressure and NO\textsubscript{x}, and an increase BSFC and soot. As exhaust gas reversal increases, it approaches the levels needed to
achieve internal exhaust gas recirculation (IEGR).

The findings presented in this paper suggest that, setting EVO to EVO117 is the better strategy for balancing blow back of scavenged air and exhaust gas reversal, and also a good solution for balancing BSFC and NO\textsubscript{x} emissions.

### 3.2.2. Various EVC Investigation

The strategies for varying EVC are listed in Table 4.

<table>
<thead>
<tr>
<th></th>
<th>Common Rail Pressure (MPa)</th>
<th>Start of Injection (°CA)</th>
<th>Injection Duration (°CA)</th>
<th>EVO (°CA)</th>
<th>EVC (°CA)</th>
<th>Scav. Air Prec. (MPa)</th>
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<td>-4.65</td>
<td>21.60</td>
<td>117</td>
<td>265</td>
<td>0.370</td>
<td>Standard</td>
</tr>
<tr>
<td>EVC270</td>
<td>70</td>
<td>-4.65</td>
<td>21.60</td>
<td>117</td>
<td>270</td>
<td>0.370</td>
<td>Standard</td>
</tr>
</tbody>
</table>

Performance parameters and emission data were calculated from EVC255 to EVC270 and results are shown in Fig. 9.

![Fig. 9 Simulation Parameters](image)

Again, the simulation correlated very well with experiments carried out in section 2.3. As shown in Fig. 9, when EVC varied from EVC255 to EVC270, combustion pressure, effective output power, NO\textsubscript{x} emissions and volumetric efficiency decreased while BFSC and soot increased.

Fig. 10(b) shows that slight exhaust gas reversal occurred at the beginning of IPO whereas far more reversal occurred at the beginning of IPC. Earlier EVC led to greater exhaust gas reversal at the beginning of IPO while later EVC led to greater exhaust gas reversal at the beginning of IPC.

![Fig. 10 Scavenging Process during EVC Various](image)

Substantial fluctuation was observed in exhaust gas pressure due to different exhaust types from freely exhausting.
to force exhausting, and was the balance between cylinder pressure and exhausting air pressure.

The results show that for the two-stroke marine diesel engine with a port-valve scavenging system, there will be reversal of exhausted air no matter how EVC is varied. This reversal results from the engine design and geometry. It is possible that exhaust reversal could reach a point at which an optimal mixture of exhausted air and fresh scavenging air is achieved, thereby decreasing the combustible factor, maximum combustion temperature, cylinder pressure and NOx, but increasing BSFC. One potential approach is to achieve internal exhaust gas recirculation (IEGR) by varying EVC.

For the research at this stage, setting EVC to EVC260 appears the best strategy for balancing exhaust gas reversal, and a good solution for balancing BSFC and NOx emissions.

4. Conclusion
The scavenging process of the two-stroke marine diesel engine was evaluated in this paper, and the effects of EVO and EVC timing on engine performance and emission were studied by experiment and simulation. The following conclusions were reached:

a) Varying the timing of EVO and EVC presents a strategy for achieving low NOx emission.

b) The findings from these studies suggest that setting EVO to EVO117 and EVC to EVC260 is the best strategy for achieving an optimal balance between BSFC and NOx emissions.

c) The results suggest that by varying EVO and EVC, it is possible to achieve IERG in a two-stroke marine diesel engine. Further research should be performed to confirm this possibility and potentialities.

Reference