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Original Paper

High expansion cycle for marine 2-stroke engine

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Abstract: A diesel engine has the same length as the compression and expansion strokes, but a Miller cycle was introduced that achieved a higher expansion than the compression cycle by operating the changing intake valve. The Miller cycle is a low-compression and high-expansion cycle, so it can achieve high efficiency and NOx reduction effect. However, high-efficiency turbochargers must be adopted simultaneously owing to the reduction of the exhaust energy entering the turbines. This study theoretically reviewed the possibility of applying high expansion cycles and 2-stage T/C to 2-stroke marine diesel engines and considered their effects. The highest pressure ratio that can be achieved in 2-stage T/C system can be obtained when the pressure distribution between the low-and high-pressure stages is the same. A low compression – high expansion cycle was applied to reduce the maximum combustion temperature by more than 200 K, and the 2-stage T/C was simultaneously applied to increase the scavenge air pressure by approximately 4 bar and Pmax by approximately 30 bar. The theoretical review confirmed that the application of the 2-stage T/C and the effect of low compression – high expansion, which is the Miller cycle, can be achieved for a marine 2-stroke engine.

Keywords: High expansion cycle, 2-stage turbocharger, Marine 2-stroke engine, Miller cycle, Atkinson cycle

Nomenclature

π	pressure ratio
η	Efficiency
T	Temperature
m	mass flow
C	heat ratio
k	specific heat ratio
β	cut off ratio
ε_e	expansion ratio
ε_c	compression ratio

Subscript

a	air
c	compressor of T/C
g	gas
p	pressure
t	turbine of T/C
v	volume
LP	low pressure stage
HP	high pressure stage
v	volume
LP	low pressure stage
HP	high pressure stage
$cool$	cooler
$ref.$	cooling water

1. Introduction

The Atkinson cycle and the Miller cycle have something in common in that they are high expansion cycles that lengthen the expansion stroke compared to the compression stroke. However, these two cycles differ in mechanical methods implemented as real engine cycles. The Atkinson cycle, devised by Atkinson in 1882, is more thermally efficient because it has a larger expansion ratio than diesel cycles with the same compression ratio. However, as the structural complexity increases to structurally increase the expansion, durability problems arise. The Miller cycle was devised to compensate for this problem. The Miller cycle implements a high expansion cycle with a long expansion period compared to the compression period by adjusting the opening and closing periods of the intake valve during the suction process of the intake air from the atmosphere. The Miller cycle, which is

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a low-compression and high-expansion cycle, has a longer expansion ratio than the compression ratio and has a more effective expansion work, which improves performance. Additionally, a low compression pressure from a low compression ratio can reduce the combustion temperature to improve the nitrogen oxides. This technology is mainly applied to four-stroke engines.

Gonca *et al.* mentioned that the application of Miller cycles through simulation and experimentation to a four-stroke organization resulted in theoretical improvements in specific fuel consumption (SFC) and NO, with an improvement of approximately 2.5% power loss **[1]**. In other studies, Gonca *et al.* reported simultaneous reductions in CO_2 and reductions in NO, HC, and CO in combination with a four-stroke diesel engine and a Miller cycle applied with turbo charging **[2]**. Lee *et al.* reported the effectiveness of applying Miller cycles with high efficiency T/C to a marine two-stroke diesel engine **[3][4]**.

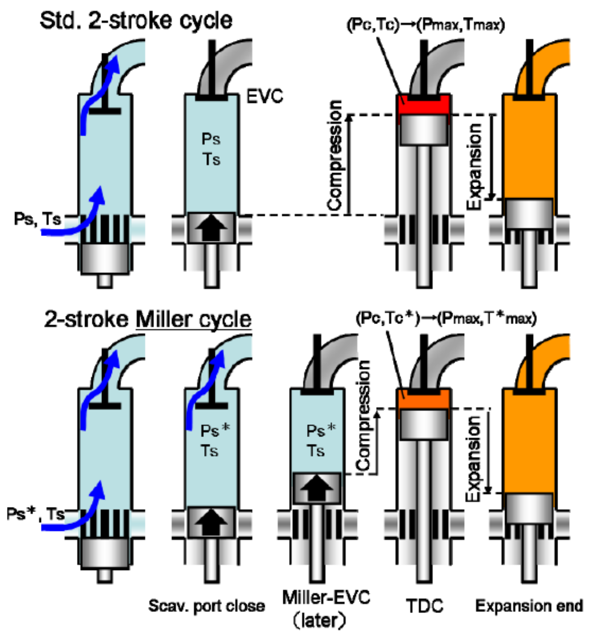

Figure 1: Miller cycle for two-stroke marine engine**[1]**

The system has scavenging ports, the start of compression, and the end of expansion is determined by the closing and opening of the exhaust valve. The low-pressure and high-expansion cycles of a marine two-stroke engine can be implemented by adjusting the closing timing of the exhaust valve. **Figure 1** presents a conceptual diagram of the Miller cycle of a two-stroke engine using the uniflow-scavenging method. Because the two-stroke engine of the uniflow scavenging system has an exhaust valve on the top of the cylinder cover, the start of compression occurs when the exhaust valve is closed. In the figure, for diesel cycles, compression can be achieved from the time the piston passes through the port. In the case of the Miller cycle, the piston passes through the port, and the exhaust valve closes during the upward stroke to initiate compression. To realize high expansion, reducing the compression ratio by delaying the closing of the exhaust valve will reduce the maximum explosive pressure because the effective compression period will be shortened. To compensate for this, a Miller cycle must be applied simultaneously with a high-boost turbocharger.

The marine 2-stroke engine is designed to be longer than the on-shore diesel engine to increase the propulsion efficiency, and the exhaust gas energy is lower than that of the 4-stroke engine, as the intake through the port pushes out the exhaust during the desired period. This means that the energy flowing into the turbocharger was low. Therefore, a high-efficiency turbocharger is required to apply a high-expansion cycle to two-stroke engines. The T/C system is a common application in diesel engines, and several research papers have been published to reduce emissions through T/C **[4]-[7]**. In this study, we theoretically examine the feasibility and effectiveness of the high expansion cycle by applying a two-stage turbocharger to marine 2-stroke engines.

2. 2-Stage Turbo Charging

2.1 Factors influencing the turbo charging system

For the high efficiency of the two-stage turbo charge system, several factors affect the engine performance, as follows:

- Charge Air Pressure

  The charge air pressure must be supplied at a certain level to reach the cylinder power and maximum pressure required by the design. To determine the boost air pressure, the timing of closing the exhaust valve must be considered. When the charge air pressure increases and the exhaust valve closing timing is delayed, the compression pressure is maintained, and the compression temperature decreases owing to the decrease in the effective compression ratio, which is effective in reducing NOx.

- Efficiency of turbo charging system

  The turbo-charge efficiency indicates whether the energy transferred from the expansion energy of the exhaust gas of the engine is converted into the inlet pressure of the engine. High efficiency can be obtained by achieving a high charging pressure with the same exhaust gas energy.

- Turbine flow characteristics

  The compressibility of exhaust gases causes resistance to the flow of exhaust gases through the turbine, which leads to a loss of flow. The flow characteristics of axial turbines in low-speed engines do not significantly affect the turbine efficiency.

- Waste gate rate

  The matching point of the turbo-charging system can be moved to a partial load through the bypass of compressed air or exhaust under a high load. The pressure above that required by the engine in a two-stage turbo-charging system can be controlled by a bypass at the high-or low-pressure stage.

- Intercooling temperature

  Intermediate cooling in a two-stage turbo charging system is a critical factor that can increase the efficiency of the entire turbo-charging system. Lower cooling temperatures can increase the efficiency of the supercharger. The limiting factor for intermediate cooling is condensate, which affects the temperature of the coolant and high-pressure compressor. The temperature at which moisture condenses in compressed air should be considered.

- Partition of the compressor overall pressure ratio

  The distribution of the pressure ratio is related to the optimization of the supercooling system and is affected by the temperature of the intermediate cooling.

2.2 Theoretical application of two-stage turbo charging to marine 2-stroke diesel engine

Figure 2 shows a schematic diagram of the two-stage turbo-charging system and diesel cycle of the 2-stroke engine. The two-stage turbo charging configuration consists of turbines and compressors at the low-and high-pressure ends, and between the compressor and the engine for each high-and low-pressure cooler. Unlike 4-stroke diesel engines that use pulse pressure turbochargers to generate speed energy from the pulsation of exhaust gas energy supplied to the turbocharger, marine 2-stroke engine usually uses constant pressure turbochargers through exhaust gas receivers. The blowdown energy emitted from the cylinder was converted to a constant pressure in the exhaust gas receiver.

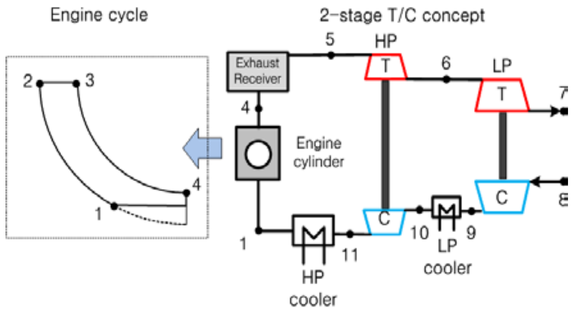

Figure 2: Schematic concept of two-stage T/C system for 2-stroke diesel cycle**[8]**

In this study, we omit the calculation of the wastegate, which should be considered in the turbo-charging system because it aims to implement a high expansion cycle with the maximum pressure ratio achieved through the two stages of the exhaust gas energy emitted by the 2-stroke cycle engine.

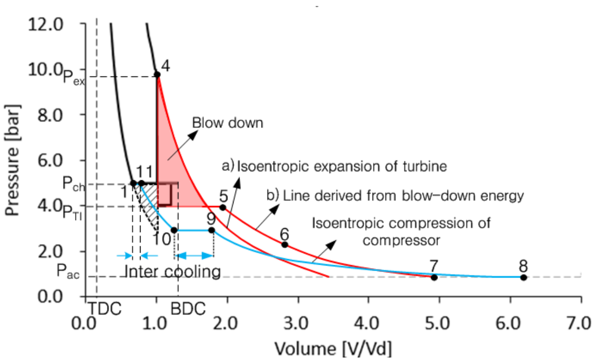

Figure 3: P-V diagram of two-stage T/C system for 2-stroke diesel engine

Figure 3 shows the exhaust process of the 2-stroke cycle and the compression process of the two-stage turbo-charging system. The start of the exhaust process in an ideal cycle begins at point (4) when the exhaust valve is opened. The exhaust gas energy from point (4) is released into the atmosphere and provides working energy to the turbines. Line (a) represents a line for the isentropic expansion of the turbine from point (4) to atmospheric pressure.

The blow-down energy increases the enthalpy of the exhaust gas, which then expands along line (b) by the turbine.

$$T_{ti,a} = T_{ex} \cdot \left[\frac{P_{ti}}{P_{ex}}\right]^{\frac{k-1}{k}} \tag{1}$$

$$T_{ti,b} = T_{ex}\left[\frac{1}{k} + \frac{k-1}{k}\frac{P_{ti}}{P_{ex}}\right] \tag{2}$$

Line (a) can be calculated as in **Equation (1)**, and line (b), in which the exhaust gas is collected in the manifold and injected into the turbine, can be derived as in **Equation (2)**.

The power of the high-pressure turbine and low-pressure turbine obtained from the expansion of the exhaust gas is used to power the low-pressure and high-pressure compressors. Atmospheric air at point (8) is compressed in a low-pressure compressor and then passes through the intermediate cooler at point (9), resulting in a higher air density. In the high-pressure compressor, one more compressed air, point (11), is supplied through the high-pressure cooler to point (1), the starting point of the cycle.

The pressure ratio of the low-pressure and high-pressure compressors through the two-stage turbo charge can be calculated using the **Equations (3)** and **(4)**.

$$\pi_{c\,LP} = \left[\frac{\eta_{c\,LP} X_t A_{tLP}(1-A_{t\,HP})}{X_c} + 1\right]^{\frac{k_a}{k_a-1}} \tag{3}$$

$$\pi_{c\,HP} = \left[\frac{\eta_{c\,HP} X_t A_{t\,HP}}{X_c\left\{(1+B_{c\,LP})(1-\eta_{cool})+\eta_{cool}\frac{T_{ref.}}{T_a}\right\}} + 1\right]^{\frac{k_g}{k_g-1}} \tag{4}$$

The substitution characters used in the above formula are as follows.

$$X_t = m_g C_{pg} T_5, \quad A_{t\,LP} = \eta_{t\,LP}\left(1-\pi_{t\,LP}^{\frac{1-k_g}{k_g}}\right),$$

$$A_{t\,HP} = \eta_{t\,HP}\left(1-\pi_{t\,HP}^{\frac{1-k_g}{k_g}}\right)$$

2.3 Application of two stage turbo charging to the 2-stroke ideal cycle

Figure 4 shows the calculation process for a high-expansion cycle with a two-stage turbo-charging system. This is the process of calculating the state of each point that can derive the maximum pressure ratio through a two-stage system by inducing the exhaust gas energy, point (4) in **Figures 2** and **3**, which can be obtained from a single-stage turbo-charging system.

To calculate the state of each point in the cycle, the P1 and T1 states at the start of compression are assumed. This induces point (4) states, which are the pressure, temperature, and mass flow at the start of the exhaust process from the cycle. The state of each point in the compression and expansion processes can be induced by the ideal gas state equation, and the heat generated by combustion is replaced by the cut-off ratio (β) of the diesel cycle.

The turbine inlet state is calculated by considering the blow-down energy, as shown in **Equation (2)**. Then, we calculate the maximum pressure ratio that can be obtained through a 1-stage turbo-charging system.

The new exhaust gas outlet conditions (P4`, T4`, and m_g`) are calculated through an ideal diesel cycle under the condition of the maximum pressure ratio obtained from the single-stage turbo-charging system. The turbine inlet conditions are calculated using the same procedure as above and apply the two-stage turbo-charging system to determine the maximum pressure ratio that can be obtained. In the two-stage compression system, the pressures at the low and high stages are calculated using **Equations (3)** and **(4)**.

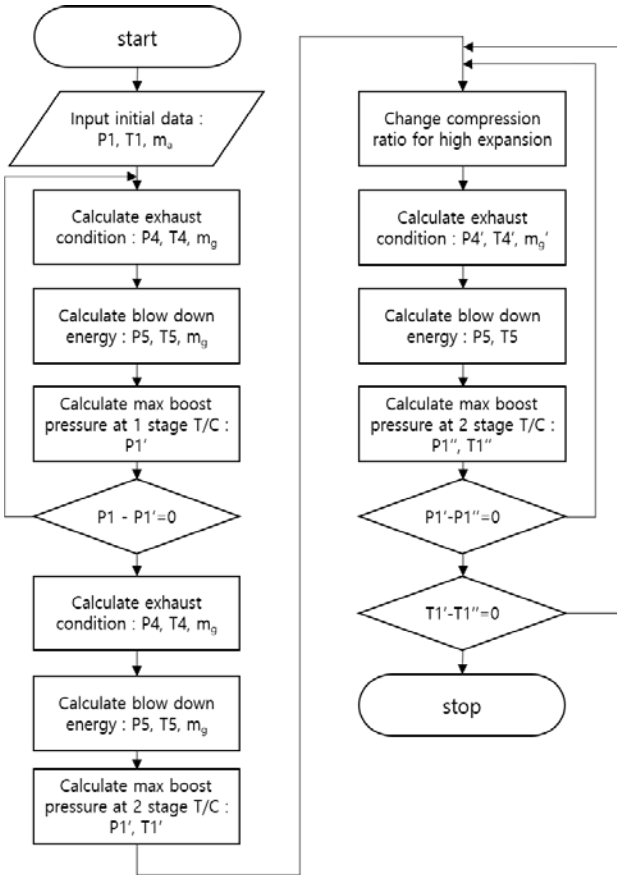

Figure 4: Flowchart for calculation of high expansion cycle adopted two-stage T/C

The high expansion cycle, miller, is calculated by finding point 1 in **Figure 3**, where the maximum pressure of the two-stage turbo-charging system matches the pressure on the compression line of the diesel cycle achieved through single-stage turbocharging. The compression ratio obtained by calculating the starting point (P4', T4', m_g') of the exhaust process of the Miller cycle and repeated calculations until the engine inlet pressure and temperature are equal is the maximum compression ratio achieved by applying a two-stage turbo-charging system.

Figure 5 shows the maximum pressure ratio that can be achieved through a two-stage turbo-charging system according to the cut-off ratio (β) of the ideal Miller cycle. The cut-off ratio indicates the degree of calories introduced into the cycle, and the larger the cut-off ratio, the higher the calories. As the cut-off ratio increases, the maximum pressure ratio that can be achieved by the two-stage turbo charge increases.

The maximum pressure of the two-stage turbo-charging system is achieved near the point where the pressure ratio of the low-pressure and high-pressure sides is equal to 1. The higher the cut-off ratio, the higher the pressure on the low-pressure side, the higher the maximum pressure of the two-stage turbocharger.

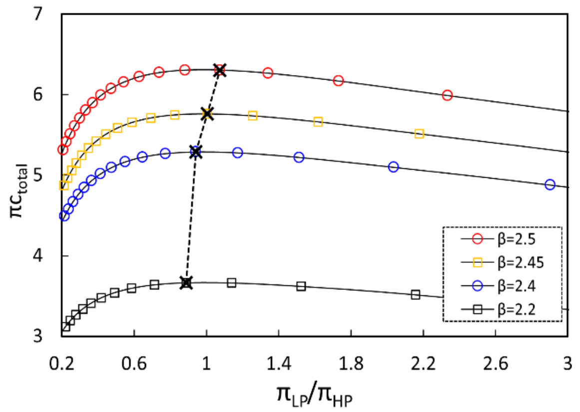

Figure 5: Achievable maximum pressure ratio by cut-off ratio according to two-stage T/C in ideal miller cycle

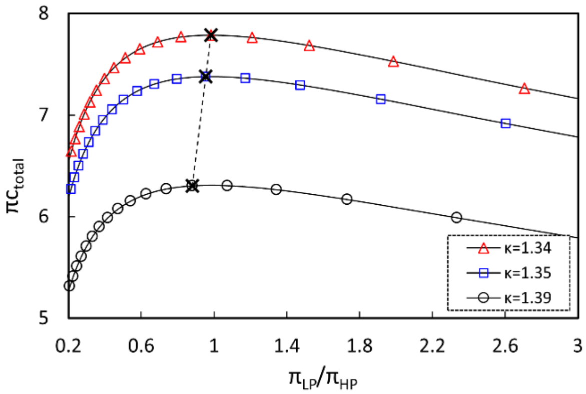

Figure 6: Achievable maximum pressure by a specific heat ratio according to two-stage T/C in ideal Miller cycle

Figure 6 shows the maximum pressure ratio that can be achieved according to the specific heat ratio (k) when applying a two-stage turbocharger in an ideal Miller cycle. The heat ratio of air was 1.4. To determine the pressure ratio for turbo charging from various calories in the simulation, the specific heat ratios were assumed to be 1.34, 1.35, and 1.39.

The smaller the specific heat ratio, the greater is the pressure that can be achieved. A smaller specific heat ratio ($k = C_p/C_v$) indicates that the specific heat at constant volume (C_v) is relatively larger than the specific heat at constant pressure (C_p), which is used to increase the internal energy during combustion. Therefore, an elevated combustion chamber temperature increases the exhaust gas temperature. This increases the turbine inlet temperature and, as a result, increases the amount of heat entering the turbine, thus increasing the turbine power. This can increase the maximum pressure because it increases the power of the compressor.

3. Results

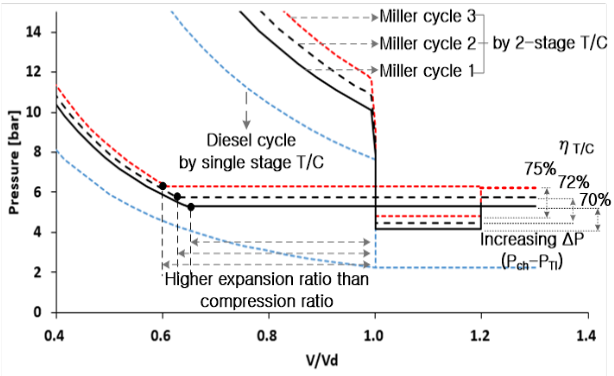

Figure 7: P-V diagram of the scavenge and exhaust part of high expansion cycle adopting two-stage T/C

Figures 7 and **8** show the low compression – high expansion of the Miller cycle applied with a two-stage turbo charge of different T/C efficiencies ($\eta_{T/C}$) from the diesel cycle, which achieved a scavenge air pressure of 2 bar by applying a single-stage turbo charge. **Figure 7** shows an enlarged area with low pressure to help understand when compression occurs and where exhaust heat escapes. The ideal cycle of the blue dotted line is a diesel cycle that can be achieved by a single-stage turbo charge. In the figure, Miller cycles 1, 2, and 3 represent the Miller cycles achieved by applying a two-stage overcharge, as shown in **Figure 4**. The greater the difference between Pch and Pti in the Miller cycle, the more efficient is the turbocharger. As the efficiency of the turbocharger increases from the exhaust conditions of the diesel cycle, the achievable compression ratio decreases. This means that the achievable Miller angle increases. A decrease in the compression ratio decreases T_{max}, resulting in a decrease in NO_x.

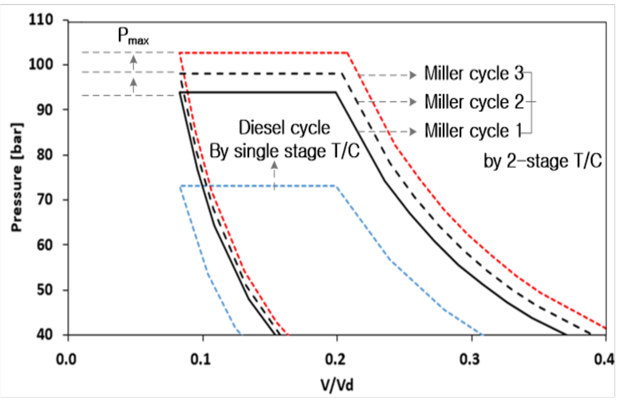

Figure 8: P-V diagram of the maximum pressure part of high expansion cycle adopting two-stage T/C

Figure 8 shows an enlarged display of high pressure in the cases shown in **Figure 7**, indicating that P_{max} also increases as the supercharger efficiency increases. The high expansion cycle, miller cycle through high-efficiency turbo-charging maintains or increases the output owing to an increase in P_{max} compared to a diesel cycle, resulting in a decrease in T_{max}, resulting in two effects: improved power and reduced NO_x.

Table 1: Comparison between diesel and miller cycles

Item	β	ε_e	ε_c	$\varepsilon_e/\varepsilon_c$	η_e	P_{ch}	P_{TI}	P_{max}	T_{com}	T_{max}
Unit					%	bar	bar	bar	K	K
Diesel	2.2	14.4	14.4	1.0	65	2.2	1.9	73.2	807	1938
Miller 1	2.2	14.4	10.2	1.4	70	5.3	4.1	93.8	691	1692
Miller 2	2.4	14.4	8.9	1.6	72	5.7	4.4	98.0	697	1707
Miller 3	2.6	14.4	7.5	1.9	75	6.3	4.8	102.6	705	1726

Table 1 summarizes the results of **Figures 7** and **8**. The figures for pressure and temperature for the diesel and Miller cycles are compared. Note that diesel cycles with one-stage T/C have a charging pressure of 2.2 bar and Miller cycles 1, 2, and 3 with two-stage T/C are higher at 5.3, 5.7, and 6.3 bar, respectively. Pmax can also be approximately 30 bar higher when a two-stage T/C is applied, which increases the output. In contrast, Tmax shows that the diesel cycle is the highest at 1938 K, and the Miller cycle is lower than that of the diesel cycle. This is the result of reduced compression ratios in the Miller cycle, resulting in a lower compression pressure. A comparison with the Miller cycle shows that P_{max} and T_{max} increase as the T/C efficiency increases. Furthermore, the higher the T/C efficiency, the greater the rate of expansion ($\varepsilon_e/\varepsilon_c$). This means that the efficiency of the T/C is extremely important to achieve good performance and reduction in combustion temperature, which are the effects of the high-expansion Miller cycle.

4. Discussion

To increase engine performance, the heat generated by combustion must be sufficiently used in the cycle to increase the efficiency of the cycle. Because the engine cycle is connected to the turbine cycle of the T/C, the optimal performance is determined by matching the engine cycle with the T/C cycle. The marine low-speed two-stroke diesel engine has not yet introduced a two-stage turbo-charging system. However, improving global warming and marine air pollution requires continuous performance improvement of engines, which are mostly used by merchant ships. Therefore, in this study, we examine the possibility of improving performance and reducing nitrogen oxides through theoretical studies of high-expansion cycles with two-stage T/C applied to 2-stroke marine diesel engines.

This study examined the possibility of performance improvement by simultaneously applying high expansion cycles and two-stage T/C to improve the performance of diesel engines. It was confirmed that the same calorific input into the cycle increased the Pmax and improved the performance. However, the feasibility of two-stage turbo charge and high-expansion cycles under various loads is challenging in this work because the waste gates of low-pressure and high-pressure turbines required for two-stage system configurations are not considered. Therefore, we believe that further research is needed on system configurations that can realize optimal performance through the application of waste gates.

The calculation of nitrogen oxides can be calculated using mechanisms in the chemical reactions of fuel and air molecules composed of oxygen and nitrogen. The Zeldovich mechanism, prompt mechanism, N_2O mechanism, and NNH mechanism were used in the calculation of nitrogen oxides **[9]**. Because the reaction of the Zeldovich mechanism to NO is determined by the combustion temperature and reaction rate at chemical non-equilibrium states, this study did not reflect the direct calculation of NOx and considered the possibility of reducing NOx by calculating the combustion temperature at each state. However, this study calculates only the highest combustion temperatures according to different specific heat and cut-off ratios of the diesel cycle; therefore, direct connections are difficult because the chemical reaction mechanisms for equilibrium and non-equilibrium are not considered at all. It is reported in various literature that NO_x is produced by strongly reacting at temperatures above 1600℃ **[10][11]**, so it is possible to reduce NO_x by reducing combustion temperature.

5. Conclusion

In this study, we derive the results of a review of the theoretical application of a two-stage turbocharged Miller cycle while simultaneously implementing a high expansion cycle by reducing the compression ratio from the ideal cycle of a marine two-stroke diesel engine to increase the relative expansion ratio. The result is the theoretical maximum pressure and distribution of pressure ratios that can be reached through a two-stage turbo-charged system compared to the input calories, and the effect that can be achieved when applied to the high expansion theory cycle. In summary, the results are as follows:

The maximum pressure that can be achieved in a 2-stage turbo-charging system increases as the cut-off ratio increases, which means that the size of the calories put into the cycle increases.

1) The maximum pressure in 2-stage turbo-charging system can be achieved when the pressure ratio of the low-and high-pressure compressors is the same.
2) A high expansion cycle can be realized by reducing the compression ratio in the diesel cycle, which can reduce the highest temperature of the combustion chamber, which is expected to reduce NOx.
3) Because the scavenging pressure can be further increased through the application of the 2-stage T/C, even if the compression ratio is lowered in the high expansion cycle, the maximum combustion pressure can be further increased.
4) High efficiency of T/C is important to achieve the high expansion cycle.

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Author Contributions

Conceptualization, J. W. Lee; Methodology, H. M. Baek; Software, H. M. Baek; Formal Analysis, J. W. Lee; Investigation, J. U. Lee; Resources, J. W. Lee; Data Curation H. M. Baek; Writing-Original Draft Preparation, H. M. Baek; Writing-Review & Editing, J. W. Lee; Visualization, J. U. Lee; Supervision, J. W. Lee; Project Administration, J. W. Lee; Funding Acquisition, J. W. Lee.

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