

# New Concept for Overcoming the Trade-Off between Thermal Efficiency, Each Loss and Exhaust Emissions in a Heavy Duty Diesel Engine

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## ABSTRACT

To overcome the trade-offs of thermal efficiency with energy loss and exhaust emissions typical of conventional diesel engines, a new diffusion-combustion-based concept with multiple fuel injectors has been developed. This engine employs neither low temperature combustion nor homogeneous charge compression ignition combustion. One injector was mounted vertically at the cylinder center like in a conventional direct injection diesel engine, and two additional injectors were slant-mounted at the piston cavity circumference. The sprays from the side injectors were directed along the swirl direction to prevent both spray interference and spray impingement on the cavity wall, while improving air utilization near the center of the cavity. Results obtained with a heavy-duty single cylinder engine equipped with multiple injectors indicated that it was possible to achieve the desired heat release rate profile by independent control of injection timing and duration (fuel injection pressure was kept in constant) for each fuel injector. Furthermore, smoke emissions were reduced by improved in-cylinder air utilization, which was possible through a different air-fuel mixture formation process than that found in conventional single-injector diesel engines. Results showed reduced friction loss, heat loss and NO<sub>x</sub> (nitrogen oxides) emissions, while maintaining indicated thermal efficiency by suppressing the peak cylinder pressure, bulk average temperature, and spray flame impingement to the cavity wall. Additionally, a simultaneous reduction in smoke and NO<sub>x</sub> emissions was achieved, without any deterioration in CO (carbon monoxide) and THC (total hydrocarbon) emissions, even compared with conventional diesel combustion. "CONVERGE" three-dimensional numerical simulation results also suggested that rapid homogenization of local equivalence ratio by improved mixture formation could result in the simultaneous reduction of smoke and NO<sub>x</sub> emissions, even with EGR.

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## INTRODUCTION

Many measures for thermal efficiency improvement have been studied and proposed over the years. A larger proportion of constant volume combustion by an increased fuel injection rate, advanced injection timing, HCCI (homogeneous charge compression ignition) combustion [1], PCI (premixed compression ignition) combustion [2] etc., have been investigated, since the thermal efficiency of the Otto cycle is ideally higher than other reciprocating engine cycles [3]. An increase in compression ratio or specific heat ratio also contributes toward increasing theoretical cycle efficiency. Furthermore, low temperature combustion by means of higher boost and higher EGR rate conditions have attracted attention in recent years because of the associated theoretical thermal efficiency improvement and heat loss reduction [4]. Nevertheless, drastic improvement in brake thermal efficiency has not been achieved yet, because simultaneous reduction in individual energy losses is difficult utilizing conventional measures, i.e. heat loss, pumping loss, exhaust loss and mechanical friction, and brake thermal efficiency [5].

However, very large two-stroke marine engines have already achieved brake thermal efficiencies exceeding 50%, in spite of engine size effects. In comparison with commercial diesel vehicles engines, marine engines operate at relatively lower fuel injection pressures of about 90MPa, and lower effective compression and expansion ratios of 14.0:1 and 17.0:1 respectively [6], which suppress peak cylinder pressure and averaged cylinder temperature. The resulting near-constant pressure (theoretical Diesel, not Otto) combustion cycle exhibits very low mechanical loss and heat loss. That is, finding the new balance point between cycle efficiency and individual losses is more important than simply raising the theoretical efficiency.

Direct control of the heat release rate profile in a current automotive diesel engine is generally difficult due to hardware constraints, e.g. limited fuel injection rate, only one injector, even if both the fuel injection pressure and timing can be precisely controlled. On the other hand, large two-stroke marine diesel engines usually have two or more injectors per cylinder, for which injection can be phased to control the heat release rate profile and thus obtain nearly constant pressure combustion (Diesel cycle.) Furthermore, heat loss can be decreased more than in conventional diesel engine by preventing spray impingement to the cavity walls by directing fuel sprays in the

tangential direction, since the injectors are mounted at circumference of the cavity. As a result, these marine engines achieve very high brake thermal efficiency with minimum heat and friction losses due to lower averaged cylinder pressure and temperature.

Several studies dealt with diesel engine equipped with multiple injectors, e.g. PREDIC [7] and MULDIC [8] concepts were proposed to potentially control PCCI combustion. These concepts were able to control PCCI combustion by optimizing the spray distribution in the cylinder during ignition delay, however, smoke emissions increased when the opposed sprays collided with each other. Other results showed that NOx and smoke emissions were simultaneously reduced by avoiding interference between pilot and main sprays injected by different injectors [9].

For a new combustion concept by means of multiple fuel injectors, two major advantages over conventional diesel combustion were presumed. Firstly, both combustion phase and cumulative heat release rate profile can be controlled because of the additional degrees of freedom in injection rate shaping, without any interference between temporally overlapped sprays from different injectors, resulting in independent air entrainment and ignition timing. Secondly, in-cylinder air utilization can be improved with properly allotted air for each injector by temporal and spatial control of the sprays. In this study, the optimum heat release rate profile for increasing brake thermal efficiency was first calculated by a zero-dimensional thermodynamic simulation. Then, to test and verify the heat release history, a single-cylinder engine equipped with three common-rail injectors was utilized for engine performance and emissions tests. The injection setup was designed to concurrently reduce heat loss to the wall and to improve exhaust emissions.

## OPTIMUM TIME-RESOLVED HEAT RELEASE RATE CALCULATION BY ZERO-DIMENSIONAL THERMODYNAMIC MODEL

The theoretical thermal efficiency can be improved by a higher degree of constant volume combustion, higher compression ratio and/or higher specific heat ratio. However, a subsequent increase in brake thermal efficiency is not always obtained if these parameters are increased and exhaust loss is decreased, since heat and mechanical losses also typically increase. To quantitatively simulate not only thermal efficiency but also each energy loss, i.e. heat loss, mechanical loss, and exhaust loss, a zero-dimensional thermodynamic diesel engine model was developed [10]. This model allowed observation of the effect of the heat release rate history, engine specifications, and operating conditions on the energy balance, to find an optimum heat release rate profile.

Figure 1 shows a process diagram of calculations performed by the zero-dimensional model. This model is based on standard fuel-air reciprocating engine cycles like Otto, limited pressure dual cycle (Sabathe cycle) and Diesel cycle. First the P-V diagram is selected and the pressure ratio and cut-off (isobaric combustion) ratio are adjusted to maintain the appropriate input energy. Theoretical IMEP (indicated mean effective pressure), bulk mean temperature, and heat release rate can be calculated from this diagram. Heat loss was

determined by Newton's law of cooling, in which the heat transfer coefficient is estimated by Woschni's correlation [11]. Apparent (gross) IMEP was calculated by subtracting the heat loss from the theoretical IMEP. Mechanical loss and FMEP (friction loss mean effective pressure), were then calculated by the Chen-Flynn friction model [12], which depends on maximum cylinder pressure and mean piston speed. And finally, BMEP (brake mean effective pressure) was determined by subtracting FMEP from apparent IMEP. Other assumptions for calculation of heat balance are as follows:

- PMEP (pumping loss mean effective pressure) was neglected by assuming in-cylinder temperature and pressure as initial value from EVO (exhaust valve open) to IVC (intake valve close) timing.
- Instantaneous in-cylinder pressure and temperature were calculated at each crank angle presuming the fuel-air cycle. Maximum in-cylinder temperature did not exceed adiabatic flame temperature, which takes thermal dissociation into account.
- Specific heat ratio (gamma) was calculated by utilizing the approximation formula based on the temperature and gas composition at the prior step.
- To limit the rate of pressure rise in the constant volume combustion phase, its duration was set to 1 degree crank angle.
- The combustion efficiency was considered as 100%. That is, the injected fuel was completely transformed into heat energy, given by the fuel's LHV (lower heating value).
- Combustion commenced at 1 degree before top dead center (BTDC), except the Diesel cycle started at TDC.

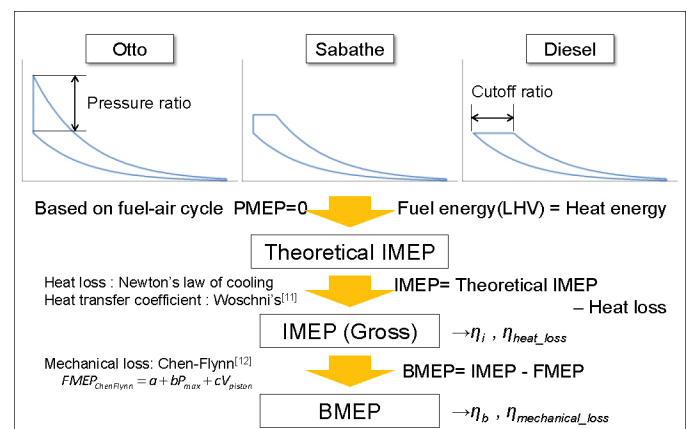


Fig. 1. Process diagram of zero-dimensional thermodynamic simulation model

Figure 2 shows a comparison of experimental and calculated energy balance and cylinder pressure history at an engine speed of 1000rpm, injection quantity of 240mm<sup>3</sup>/stroke (BMEP of about 1.9MPa in this experimental data), boost pressure of 391.3kPa (absolute), intake temperature of 323K and EGR rate of 50%. In calculation, the pressure ratio was determined from the maximum in-cylinder pressure (Pmax) to match experimental data. The cut-off ratio was then calculated by subtracting the total heat energy inputs during the constant volume combustion phase from the total fuel energy. Figure 2 shows that although PMEP and unburned fuel were not accounted for in the model, the calculation results coincide well with measurements.

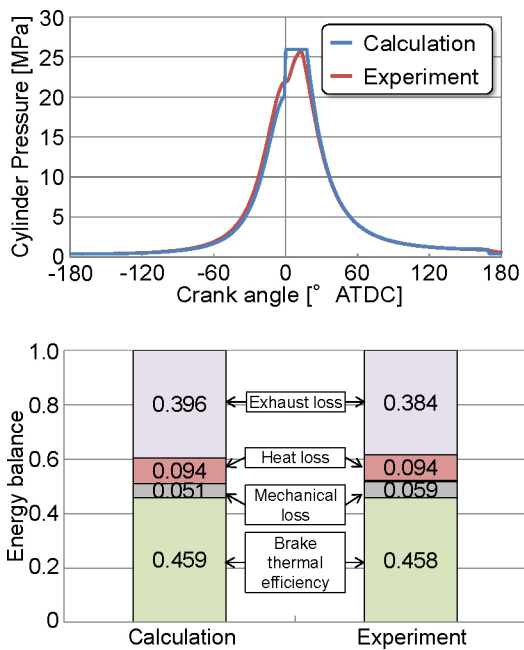


Fig. 2. Comparison of experimental and calculated cylinder pressure histories and energy balance

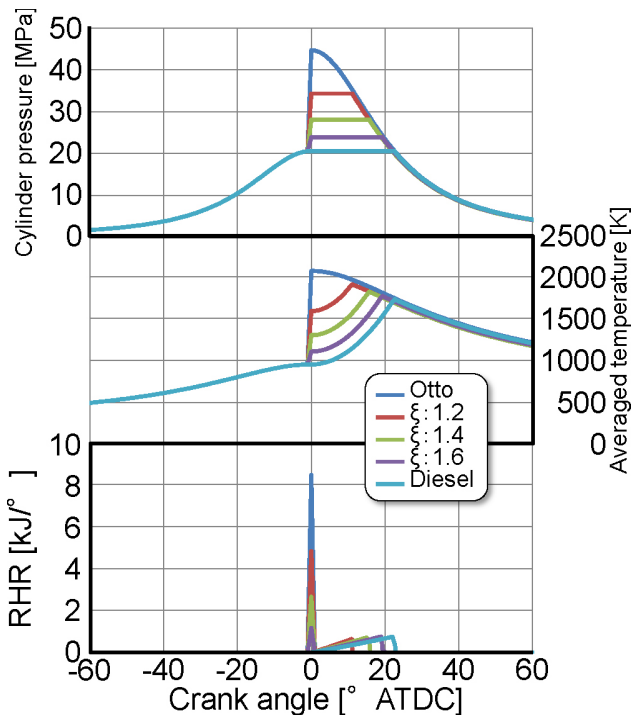


Fig. 3. Comparison of calculated cylinder pressure, temperature and heat release rate for the different idealized cycles and cut-off ratios

Figure 3 shows a comparison of calculated in-cylinder pressure, bulk temperature, and heat release rate at various combination of pressure ratio and cut-off ratio (volume ratio in constant pressure combustion term). These conditions span the Otto, Sabathe, and Diesel cycles, and are carried out under the same conditions as in Fig. 2. Cylinder pressure and temperature rise sharply with heat release rate at top dead center for Otto cycle. The dual cycle heat release rate consists of

the initial acute part and the following “delta” part. Furthermore, as the cut-off ratio increases, the initial acute heat release rate diminishes (finally becoming zero for the Diesel cycle), and the delta heat release period lengthens. The transition from Otto cycle to Diesel cycle anticipates a simultaneous reduction in mechanical loss and heat loss by suppressing the rapid increase in pressure and temperature, even though the proportion of constant volume combustion decreases.

Consideration of compression ratio is appropriate for investigation into the optimum combustion profile to achieve higher brake thermal efficiency (BTE). One should be aware that higher compression ratio may be associated with higher heat loss and lower mechanical efficiency due to increased pressure and temperature. So, the combined effects of compression ratio and heat release rate history on the BTE were then investigated by means of the zero-dimensional model under the same operating conditions as above. Figure 4 shows the effects of compression ratio and cut-off ratio on theoretical thermal efficiency, (gross) indicated thermal efficiency, heat loss and mechanical loss (again normalized by total fuel energy). Theoretical thermal efficiency increased when the compression ratio was increased from 14.0:1 to 22.0:1 and/or cut-off ratio was decreased. However, both heat and mechanical losses also increased, as presumed above.

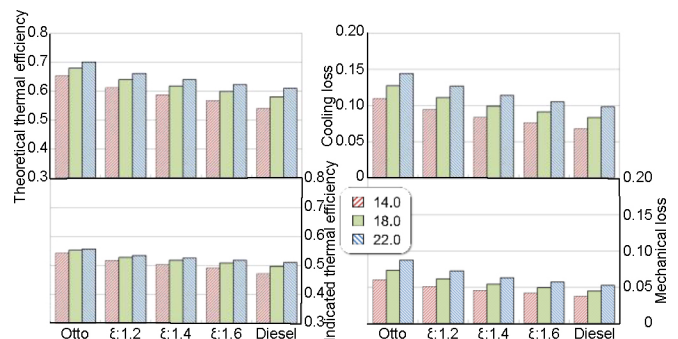


Fig. 4. Effect of compression ratio and cut-off ratio on theoretical thermal efficiency, heat loss, indicated thermal efficiency and mechanical loss

Figure 5 shows a comparison of BTE and Pmax for various combinations of compression ratio and cut-off ratio. These results show that a higher BTE is achieved at a compression ratio of 18.0:1 and cut-off ratio of 1.6 (right-most red crown) under the constraint of Pmax = 25MPa, compared to the combination of lower compression ratio and higher cut-off ratio. The center (red) line corresponds to 25MPa, the current maximum level for heavy-duty commercial vehicles. Thus, the optimum heat release rate profile to achieve the highest BTE depends heavily on not only Pmax constraints but also on each energy loss as a function of the instantaneous pressure and temperature. It is readily apparent that the Otto-like cycle could not result in optimum BTE because of the increase in other energy losses. Thus it becomes a kind of chicken-and-egg problem to determine the optimum heat release rate profile. So, given more degrees of freedom and real-time control of the heat release rate profile to maximize the cycle efficiency concurrent with minimizing energy losses at any operating condition, BTE could be improved further.