

A Study on Reduction of Heat Loss by Optimizing Combustion Chamber Shape

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ABSTRACT

A method to improve fuel consumption in diesel engines is to enhance their theoretical thermal efficiency by increasing their compression ratio. However, this results in an increase in heat loss due to the elevation of the concomitant in-cylinder temperature and the expansion of the impingement area between fuel spray and chamber wall. Therefore, reducing heat loss to the chamber wall is important to effectively benefit from a high compression ratio.

To meet this challenge, in this study, we optimized the combustion chamber shape using the three-dimensional computational fluid dynamics (CFD) simulation software, CONVERGE. A rationale proposed by the University of Wisconsin-Madison was selected to outline the shape and combined with a multiobjective optimization software, modeFRONTIER. The calculations produced a shallow dishlike combustion chamber comprising a plateau at its center that may reduce heat loss. In this system, a portion of fuel spray remained at the center of the combustion chamber because of wall impingement. The lower half of this spray developed along the chamber wall, partially blocking air entrainment. These phenomena resulted in weak premixed combustion, lowering the combustion temperature and thus reducing heat loss.

Furthermore, experiments using a single-cylinder engine were performed to evaluate the effect of the optimized combustion chamber shape on heat loss. The optimized combustion chamber improved fuel consumption under high load and advanced injection timing conditions. A weak premixed combustion was observed, as predicted by the CFD calculation. The thermal balance analysis also revealed that heat loss from the cylinder decreased while exhaust loss increased.

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INTRODUCTION

Solutions to global warming issues and depleting fossil-oil resources hinge on improved fuel consumption. The diesel engine has recently attracted considerable research and development attention as a power source for heavy duty vehicles and passenger cars. The improvement of the indicated thermal efficiency and the reduction of friction loss are essential for enhancing brake thermal efficiency. A higher indicated thermal efficiency requires an enhanced theoretical thermal efficiency, combustion efficiency, and degree of constant volume combustion. Reducing heat loss to the chamber wall also leads to better thermal efficiency. Figure 1 shows heat balances under high load conditions calculated using a zero-dimensional combustion model [1,2,3]. Higher compression ratios improve brake work because of the increased theoretical thermal efficiency. Similarly, advanced injection timing boosts brake work by augmenting the degree of constant volume. However, these parameters augment heat loss because they increase the in-cylinder temperature and expand the impingement area between fuel spray and chamber wall. Therefore,

reducing heat loss to the chamber wall is important to effectively obtain an optimal theoretical thermal efficiency and a higher degree of constant volume.

Various recent studies have focused on decreasing heat loss from cylinders. In the Temperature Swing Heat Insulation concept [4], the combustion chamber wall is coated with low heat conductivity and low heat capacity materials for the chamber wall surface temperature to follow the transient gas temperature. The resulting decrease in the temperature difference between the chamber wall and gas lowers the heat loss from the cylinder without degrading the volumetric efficiency. However, this method is hampered by surface coating cost and durability issues.

On the other hand, approaches to reduce heat loss by controlling intake air conditions and injection strategy have been studied. An advantage of these approaches is that they do not need hardware upgrade. Combustion can decrease heat loss under high exhaust gas recirculation (EGR) conditions [5] because of a lower flame temperature. However, it degrades the degree of constant volume because of a prolonged duration.

Multiple injections reduce the flame impingement area to the chamber wall and keep high-temperature gas away from the wall [6]. Despite lowering heat loss, this approach is difficult to implement in high compression ratio engines because these engines display a smaller combustion chamber volume than their low compression ratio counterparts.

In this study, a diesel engine with a high compression ratio was selected for its high theoretical thermal efficiency. The combustion chamber shape was optimized using three-dimensional computational fluid dynamics (CFD) simulations to reduce heat loss. Several computational studies have dealt with the optimization of combustion [7,8,9,10]. Most of them have focused on fuel consumption and exhaust emissions, such as nitrogen oxides (NO_x) and soot. On the other hand, this study addressed the effect of the combustion chamber shape on the degree of constant volume and heat loss.

First, the combustion features of re-entrant- and shallow dish-type chambers were investigated. Once the combustion chamber shape was optimized, the reason why the optimized shape has a lower heat loss was studied. Finally, experiments using the single-cylinder engine were performed to evaluate the effect of the optimized combustion chamber shape on heat loss.

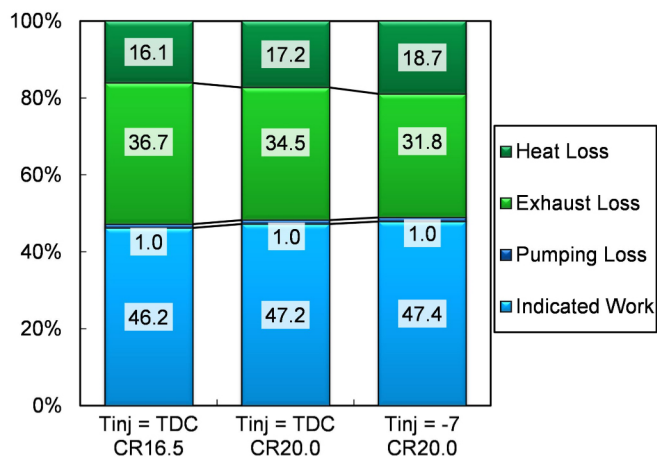


Figure 1. Calculated heat balances under high load conditions

COMPUTATIONAL METHOD

Engine Modeling by Computational Fluid Dynamics

Fuel consumption and exhaust emissions were predicted using the three-dimensional combustion simulation software called CONVERGE (Convergent Science, Inc). Primary computational submodels are shown in Table 1. To optimize the calculation time, Shell [13] and characteristic time combustion (CTC) models [14] were used for ignition and combustion, respectively. Similarly, the basic mesh size of the adaptive mesh refinement was set to 4 mm and the minimum mesh size was set to 0.5 mm. Moreover, a sector mesh was used for the mesh number.

Table 1. Submodels for combustion simulation

Breakup Model	KH-RT [11]
Wall Film Model	O'Rourke and Amsden [12]
Ignition Model	Shell [13]
Combustion Model	CTC (Characteristic Time Combustion) [14]
Turbulence Model	RNG k-ε [15]
Wall Heat Transfer Model	O'Rourke and Amsden [16]
NO _x Model	Extended Zeldovich Mechanism [17]
Soot Model	Hiroyasu [18]

Evaluation of Prediction Accuracy

Calculation and experimental results were compared to determine the accuracy of the predictions. Table 2 shows engine specifications and operating conditions. Engine speed and engine load were selected to maximize brake thermal efficiency in multi-cylinder engine. The limits of thermal efficiency in diesel engines are of interest in this study, because the actual fuel consumption of vehicles can be improved by the increase in the usage of high-efficiency operating conditions.

First, the effect of the wall heat transfer model on calculations was investigated. Calculated and experimental heat loss rates were compared (Figure 2). Experimental heat loss was calculated by thermal balance analysis. The Han and Reitz model [19] overestimated heat loss, whereas the Angelberger model [20] underestimated it. On the other hand, the O'Rourke and Amsden model [16] agreed with experimental results and was therefore selected.

Figure 3 shows a comparison between experimental and calculated cylinder pressures and heat release rates (ROHRs). The calculated heat release rate displayed a slightly less intense peak than the experiment. The calculated combustion duration was also longer than the experiment. Moreover the spike in the calculated ROHR at around 22° after top dead center (ATDC) was from combustion of after injection fuel. For the experimental result, there was no clear peak at around 22° ATDC. This might be because a detailed chemistry solver wasn't used. As the after injection fuel burns in lower oxygen concentration atmosphere, combustion of after injection might not be well simulated by the Shell and CTC model.

Table 2. Engine specifications and operating conditions

Engine Type	4 stroke, 6 cylinder engine	
Bore x Stroke	115 mm × 125 mm	
Compression Ratio	16.5	
Combustion Chamber Type	Re-entrant	
Nozzle Hole Diameter and Number	0.137 mm, 9	
Swirl Ratio	1.8	
Engine Speed	1200 rpm	
Injection Quantity	Main	146.0 mm ³ /st.
	After	13.0 mm ³ /st.
Injection Timing	Main	-5 °ATDC
	After	19 °ATDC
Injection Pressure	130 MPa	
IMEP	2.2 MPa	

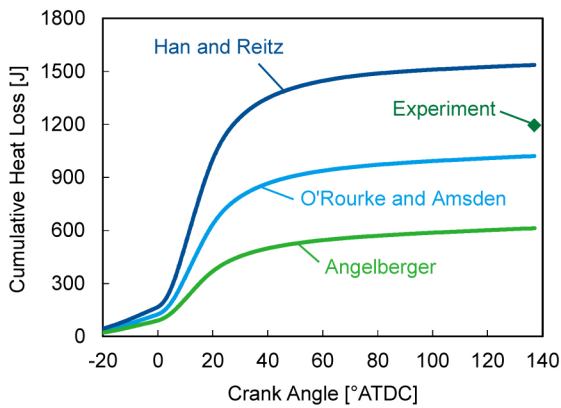


Figure 2. Comparison between experimental and calculated heat losses for various wall heat transfer models

Indicated specific fuel consumption (ISFC) and indicated specific NOx exhaust emission (ISNOx) are compared in Table 3. Calculation and experimental results are relatively compared. Experimental values were predicted within several percent of error. These comparisons demonstrate that the selected engine modeling approach is sufficiently accurate for this study. With respect to the soot emission, as simulation results couldn't capture experimental results well, the soot emission wasn't focused in CFD study.

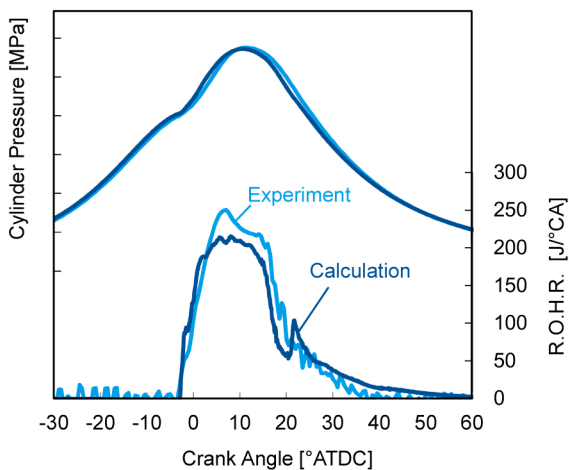


Figure 3. Comparison between experimental and calculated cylinder pressures and heat release rates

Table 3. Comparison between the experimental and calculated values of ISFC and ISNOx. Calculation and experimental results are relatively compared.

	ISFC [-]	ISNOx [-]
Experiment	1.00	1.00
Calculation	1.04	1.07

CALCULATION RESULTS

Effects of Chamber Shape and Compression Ratio on Heat Loss

As a preliminary study, the effects of combustion chamber shape and compression ratio on heat loss was investigated by CFD. One re-entrant- and two shallow dish-type chambers (shallow dish A and

B) were compared. Figure 4 shows their computational mesh. Shallow dish A-type chamber was narrow and deep, whereas shallow dish B-type chamber was wide and shallow. Engine specifications and operating conditions were the same as mentioned above (Table 2), except for chamber shape, compression ratio, and injection condition. Single injection was set and injection quantity of after injection was added to main injection. Therefore injection quantity was 159 mm³/st. and injection timing was -5° ATDC.

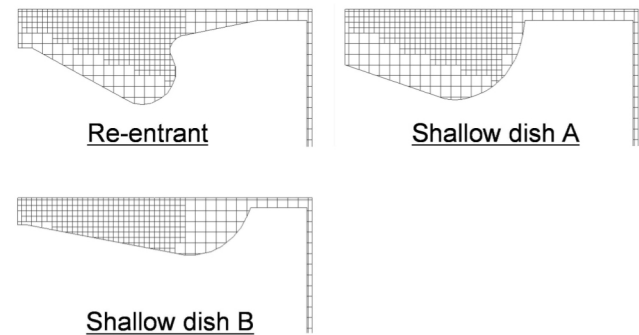


Figure 4. Computational mesh of re-entrant-type, shallow dish A-type, and shallow dish B-type combustion chambers

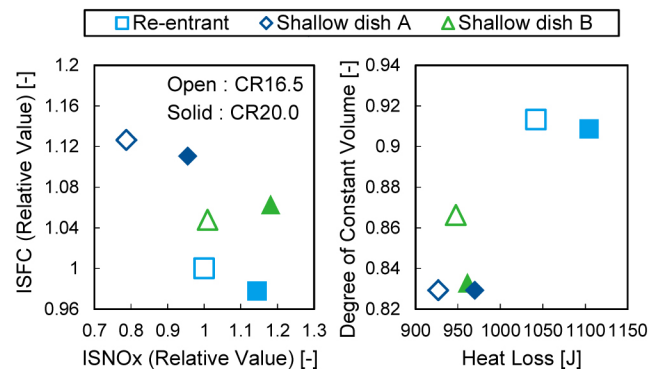


Figure 5. Effects of compression ratio and combustion chamber shape on ISFC, ISNOx, heat loss, and degree of constant volume. Open and solid symbols represent calculation results for compression ratios of 16.5 and 20, respectively.

Figure 5 shows simulated ISFCs, ISNOx, heat losses, and degrees of constant volume combustion for all three shapes. ISFC and ISNOx are relative values based on the results obtained using a compression ratio of 16.5 for a re-entrant-type chamber. Open and solid symbols represent results for compression ratios of 16.5 and 20, respectively. The degree of constant volume combustion η_{gl} was calculated by the following equation using theoretical thermal efficiency of Otto cycle η_{th} , cumulative apparent heat release Q , stroke volume V_h , clearance volume V_c , ratio of specific heats κ , in-cylinder volume V_θ , and apparent rate of heat release $dQ/d\theta$.

$$\eta_{gl} = \frac{1}{\eta_{th} \cdot Q} \int \left\{ 1 - \left(\frac{V_h + V_c}{V_\theta} \right)^{1-\kappa} \right\} \cdot \frac{dQ}{d\theta} d\theta \quad (1)$$

Results of figure 5 indicate that for re-entrant and shallow dish A-type chambers, ISFC improved with increasing compression ratio. This is because the increase in the theoretical thermal efficiency exceeds heat

loss degradation. On the other hand, the increase in compression ratio worsened ISFC for the shallow dish B-type chamber. This may stem from a lower degree of constant volume.

Figure 6 compares heat release rates and cylinder gas temperatures between the different chambers at a compression ratio of 20.0. In the re-entrant-type combustion chamber, the fuel spray was divided into two areas because of wall impingement. Consequently, heat loss increased due to the large impingement area. On the other hand, when air was utilized effectively in the cylinder, the combustion state was good, and the degree of constant volume was high. In the shallow dish A-type chamber, combustion occurred slowly after wall impingement because the fuel spray was distributed in the circumferential area and fresh air present in the central area was not used. Therefore, combustion state worsened during later combustion phases, resulting in low heat loss but low degree of constant volume combustion. The distance to the wall impingement was longer in the shallow dish B-type chamber than that in the shallow dish A-type counterpart, providing a good combustion before wall impingement. On the other hand, the combustion state degraded, and the combustion was prolonged after wall impingement because the fuel spray was distributed in the circumferential area and fresh air existing in the central area was not consumed, similar to the shallow dish A-type chamber.

Consequently, for the designs explored here a high compression ratio engine designed for higher theoretical thermal efficiency produces worse combustion state because of difficulties in mixing air and fuel. Moreover, an increased in-cylinder gas temperature and an enlarged impingement area between fuel spray and chamber wall increased heat loss. Therefore, reducing heat loss to the chamber wall with a high degree of constant volume and improved combustion is essential for fully benefiting from the high compression ratio.

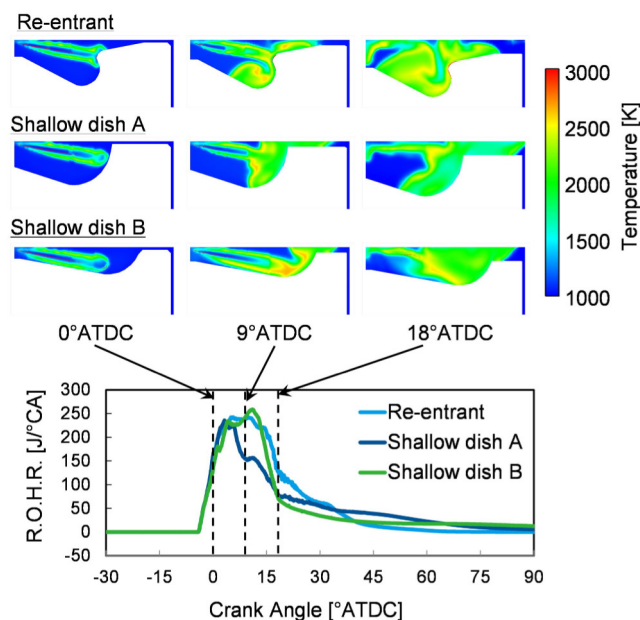


Figure 6. Calculated heat release rates and cylinder gas temperatures of re-entrant- and shallow dish-type combustion chambers for an engine with a 115 mm bore and a 125 mm stroke

Effects of Engine Size and Combustion Chamber Shape on Heat Loss

To investigate the effect of engine size on heat loss, an engine with 115 mm bore and 125 mm stroke (Table 2) was compared with a larger engine with 147 mm bore and 154 mm stroke. The combustion chamber shape of the second engine was defined to achieve a compression ratio of 20.0. Operating conditions were the same as in Table 2. Therefore, for the larger engine, indicated mean effective pressure (IMEP) was reduced by half.

Figure 7 shows calculated ISFC, ISNOx, heat loss, and degree of constant volume for both engines. ISFC and ISNOx values were relative to the results obtained for the re-entrant-type 115 mm bore engine. Open and solid symbols correspond to 115 mm and 147 mm bore engines, respectively. These results revealed that ISFC was improved for the larger engine but ISNOx increased.

For re-entrant-type chambers, engine size did not impact heat loss significantly, but an increase in the degree of constant volume improved ISFC. For shallow dish A-type chambers, elevated cylinder gas temperatures increased heat loss because of a better combustion state. Nevertheless, ISFC was significantly improved when the degree of constant volume increased. A reduced heat loss and an increased degree of constant volume significantly improved ISFC in shallow dish B-type chambers. Furthermore, engine enlargement increased NOx emissions for all chamber shapes. There is more in-cylinder oxygen relative to the fuel for the larger engine than the smaller engine. Therefore, improvement of combustion states was assumed to boost the combustion temperature.

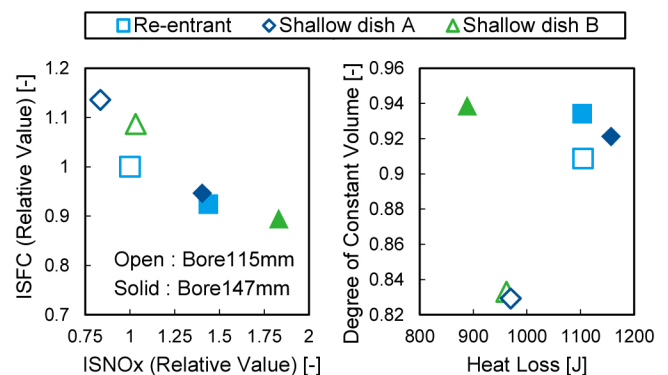


Figure 7. Calculated ISFC, ISNOx, heat loss, and degree of constant volume for 115 mm (open symbols) and 147 mm bore engines (closed symbols).

Figure 8 compares heat release rates and cylinder gas temperatures of re-entrant- and shallow dish-type combustion chambers for large bore engines. The later phase of combustion was activated, and the combustion was shortened for all chamber shapes, suggesting that the combustion was almost complete before wall impingement because of the longer impingement distance to the wall. The impingement area between fuel spray and chamber wall was smaller for shallow dish B-type chambers than that for re-entrant- and shallow dish A-type chambers. Therefore, combustion may be achieved with a lower heat loss and a higher degree of constant volume.