Effect of the Piston Crown Contour on the Fluid Flow of Diesel Engine using Biodiesel B30 Based on Simulation

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***Abstract* - diesel engines are categorized into internal combustion engines. On the internal combustion engine, combustion occurs in the combustion chamber. The combustion process requires three main elements: fuel, air, and heat from compression. The combustion process on the internal combustion engine will affect engine performance or emissions. The engine used is Yanmar TF85MH, one cylinder, four-stroke, direct-injection with B30 fuel palm oil. This study analyzes the effect of the piston bowl depth on the resulting fluid flow. To get the data, the author using the ANSYS software. The current piston bowl depth is 15,5 mm. In this study, model variation with addition and reduction of +2, +1, 0, -1, and -2 (in mm). The results of the simulation show that the highest swirl ratio occurs in case 5 with a value of -1,15, the highest temperature occurs in case 4 with a value of 2373°K, and the highest heat release rate occurs in case 4 with a value of 3,25x10^{8} erg/degree.**

Keywords - computational fluid dynamics, flow fluid, piston bowl.

I. Introduction

The diesel engine is an internal combustion engine where combustion in a diesel engine occurs due to fuel injection, air supply, and heat due to compression in the combustion chamber [1]. Until now, diesel engines are still improving their performance and thermal efficiency. Besides, diesel engine exhaust emissions must also be adjusted to recognized standards. The emissions referred to are NOx and soot. NOx emissions occur as a result of the reaction of nitrogen and oxygen during the combustion process. NOx emissions can occur due to the inhomogeneity of the fuel and air mixture. The high temperature in the combustion chamber can also increase the NOx emission content [2].

There is a relationship between the entire exhaust gas with a mixture of fuel and air in the combustion process [3]. The combustion of fuel can be influenced by several factors, for example, injection pressure, piston shape, exhaust pattern, air rotation, the amount of fuel injected, etc. [4]. One journal [5] also claims that in the combustion process, the right mixture of air and fuel is needed. The airflow in the combustion chamber will also affect the combustion process in the diesel engine. There are two types of flow in the combustion chamber, namely, swirl and tumble [6]. Swirl flow is a flow of air that forms a loop or coil, while tumble flow is a flow of air that is perpendicular to the cylinder. In the combustion process, swirl flow is expected because it can increase the turbulent intensity [7,8]. The high turbulent intensity causes combustion in the combustion chamber to occur more thoroughly. Swirl flow can be obtained by changing the shape of the piston crown [9].

Several forms of piston crown have been analyzed, namely HCC (hemispherical combustion chamber), SCC (shallow depth combustion chamber), and TCC (toroidal combustion chamber). In his research, the piston with the shape of the TCC (toroidal combustion chamber) has the highest swirl ratio [1].

With this research, the most ideal geometry of the piston crown will be found so that there is no decrease in combustion quality when using B30 biodiesel fuel.

II. Method

A. Research Problem

At this stage, the authors formulate problems based on the implementation of biodiesel B30 in Indonesia. The use of biodiesel B30 can reduce engine performance. Therefore, we need a way to improve engine performance, one of which is by modifying the piston.

B. Collecting Data

In this research, several literature studies are needed in the simulation process. Likewise, the data required in this study are as follows:

1. Piston Geometry

In this study, the variation used is the depth of the piston bowl. However, to maintain the engine compression ratio, modification of the bowl piston diameter and TDC clearance is required.

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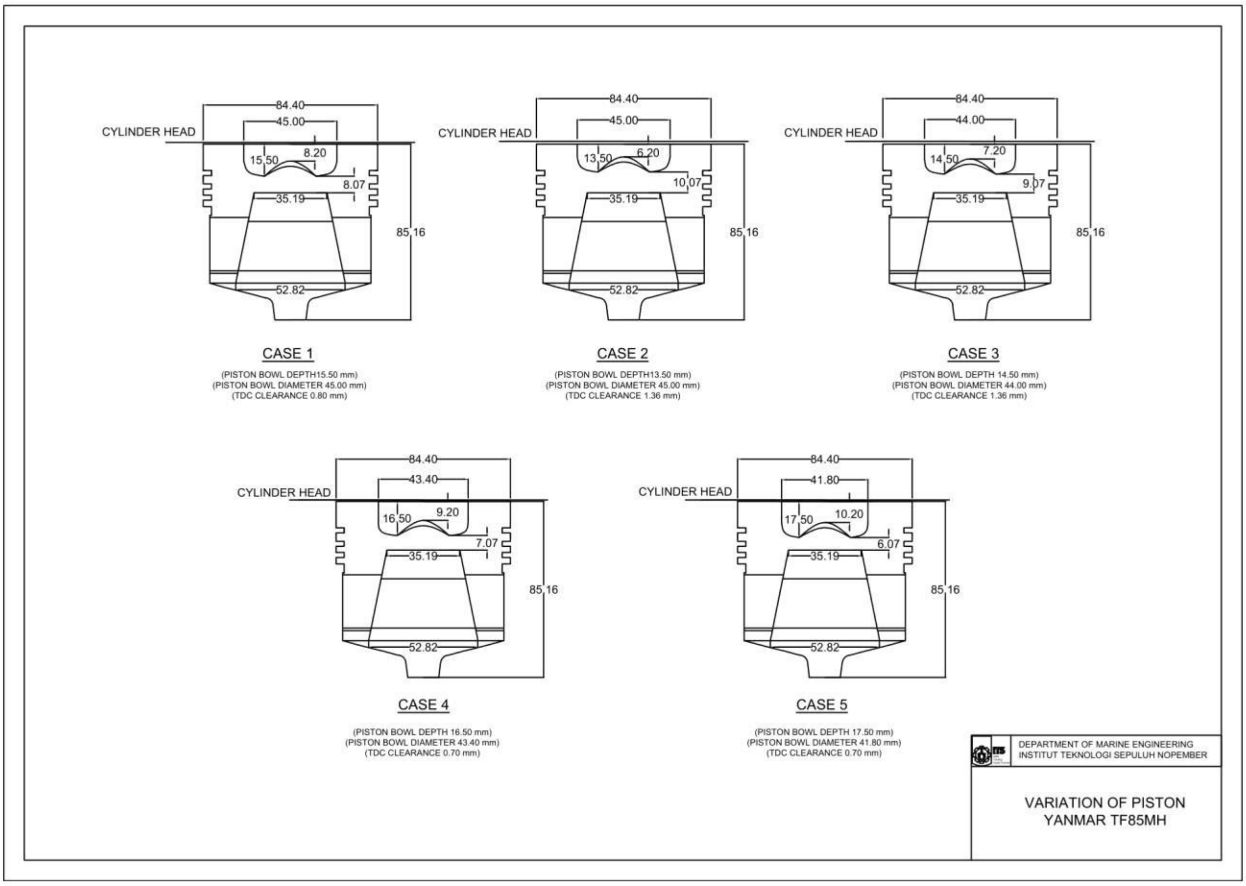

Figure. 1. Piston Variations

TABLE 1.
PISTON CROWN GEOMETRY

CASE	Piston Bowl Depth	Piston Bowl Diameter	TDC Clearance
Case 1	15,50 mm	45,00 mm	0,80 mm
Case 2	13,50 mm	45,00 mm	1,36 mm
Case 3	14,50 mm	44,00 mm	1,36 mm
Case 4	16,50 mm	43,40 mm	0,70 mm
Case 5	17,50 mm	41,80 mm	0,70 mm

2. Engine Specification

   Engine specifications are obtained from the Yanmar TF85MH manual.

TABLE 2.
YANMAR TF85MH SPECIFICATION

1	Number of cylinders	One Cylinder
2	Injection timing	18 before TDC
3	Bore x Stroke	85 mm x 87 mm
4	Volume of cylinder	493 cc
5	Compression ratio	18

3. Fuel Characteristics

   As data input in the ANSYS software, the fuel composition to be used is required, namely palm oil biodiesel B30. The composition of B30 biodiesel is 70% HSD fuel and 30% Methyl Palmitate or Fatty Acid Methyl Ester (FAME).

4. Air Composition

TABLE 3.
AIR COMPOSITION

O_2	13,69 %
N_2	74,73 %
CO_2	7,89 %
H_2O	3,69 %

C. CFD Modelling

After determining the geometry modeling of the piston bowl, it is necessary to manufacture the main components in the combustion chamber (cylinder head, cylinder liner, and piston) in 3D with dimensions as shown above. The software used by the writer in modeling is SOLIDWORKS. Then these components need to be carried out by the assembly process at SOLIDWORKS and continued by making a negative volume of the combustion chamber with each predetermined piston model. To be simulated using Ansys Forte, it is necessary to convert the data to Parasolid (*. X_t).

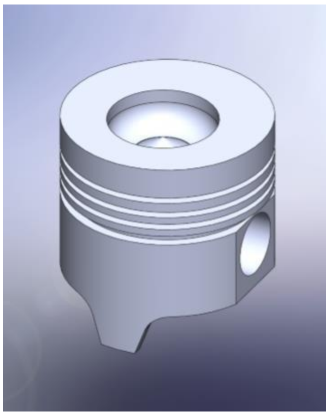

Figure.2. Isometric View Piston Standard 3D

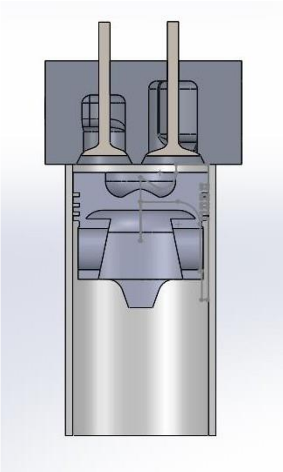

Figure. 3. Front View Combustion Chamber 3D

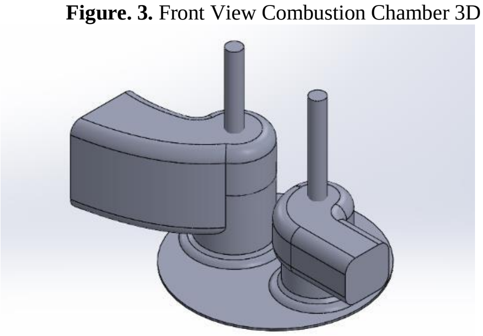

Figure. 4. *Flow Domain* Yanmar TF85MH in *Parasolid(*.x_t)*

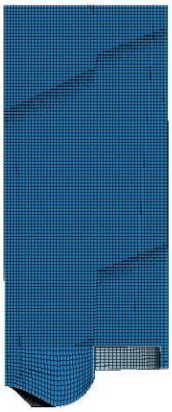

Figure. 5. Meshing on Forte Mesh Generator

D. CFD Simulation

This simulation uses ANSYS ICE FORTE software using the Yanmar TF85MH engine with a variety of piston models and uses palm oil B30 biodiesel as a fuel. At this stage, some data has been required that need to be input, such as the spray model, initial conditions, and boundary conditions. The data used can be seen in the following table:

TABLE 4.
SPRAY MODELS

Inflow droplet temperature	305 ˚K
Mean cone angle	11˚
Nozzle diameter	0,025 cm
Start of injection	18 ˚CA BTDC
Duration of injection	36˚
Total injected mass	22 mg

TABLE 5.
BOUNDARY CONDITIONS

Piston temperature	500 ˚K
Cyl. head temperature	500 ˚K
Cyl. Liner temperature	430 ˚K

TABLE 6.
INITIAL CONDITIONS ON 60˚ BTDC

Temperature	500 ˚K
Pressure	7,55 bar

A. Output Simulation

After the CFD simulation using ANSYS ICE Forte software is complete, several outputs are obtained that will answer the research objectives. From the simulation results, a graph of the swirl ratio, maximum temperature, and heat release rate will be obtained using several variations in the depth of the piston bowl. In addition, image visualization of the velocity of fluid and temperature in the combustion chamber is also obtained.

B. Discussion and Conclusion

When all the necessary data have been collected, it is followed by an analysis that will answer all the research objectives set by the author. Analysis can be carried out when all the necessary data have been collected. The results of the analysis must answer all of the research objectives set by the author.

III. RESULTS AND DISCUSSION

A. Swirl ratio

Based on the iteration results obtained from the simulation, case 1 produces a swirl ratio of -1.105 at 3˚ crank angle before TDC, case 2 produces a swirl ratio of -0.98 at 5˚ crank angle before TDC, case 3 produces a swirl ratio value of -0.99 at 4.94 crank angle before TDC, case 4 produces a swirl ratio value of -1.13 at 4˚ crank angle before TDC, while case 5 produces the highest swirl ratio value of -1.15 at 4, 5˚ crank angle before TDC.

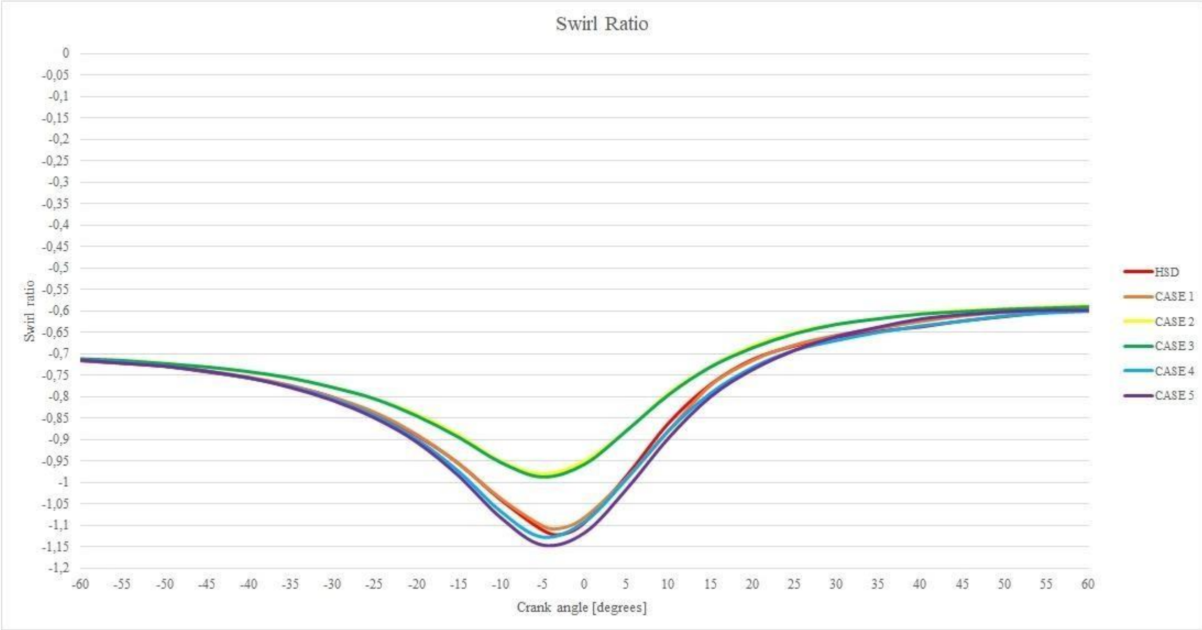

Figure. 6. Swirl Ratio

However, when the TDC position, case 1 has a swirl ratio value of -1.08, case 2 has the lowest swirl ratio value of -0.95, case 3 has a swirl ratio value of -0.958, case 4 has a swirl ratio value of - 1,092, and case 5 has a swirl ratio of -1,117. From the simulation results, it can be seen that the swirl ratio in case 2 and case 3 has decreased. Whereas in case 4 and case 5 it is bigger than the normal piston fueled by biodiesel B30 (case 1).

From these results, it can be seen that changes in the geometry of the piston bowl with a reduction in the depth of the piston bowl and the addition of TDC clearance affect the decrease in the swirl ratio. Meanwhile, changing the piston model by increasing the depth of the piston bowl along with a reduction in the TDC clearance can increase the swirl ratio value.

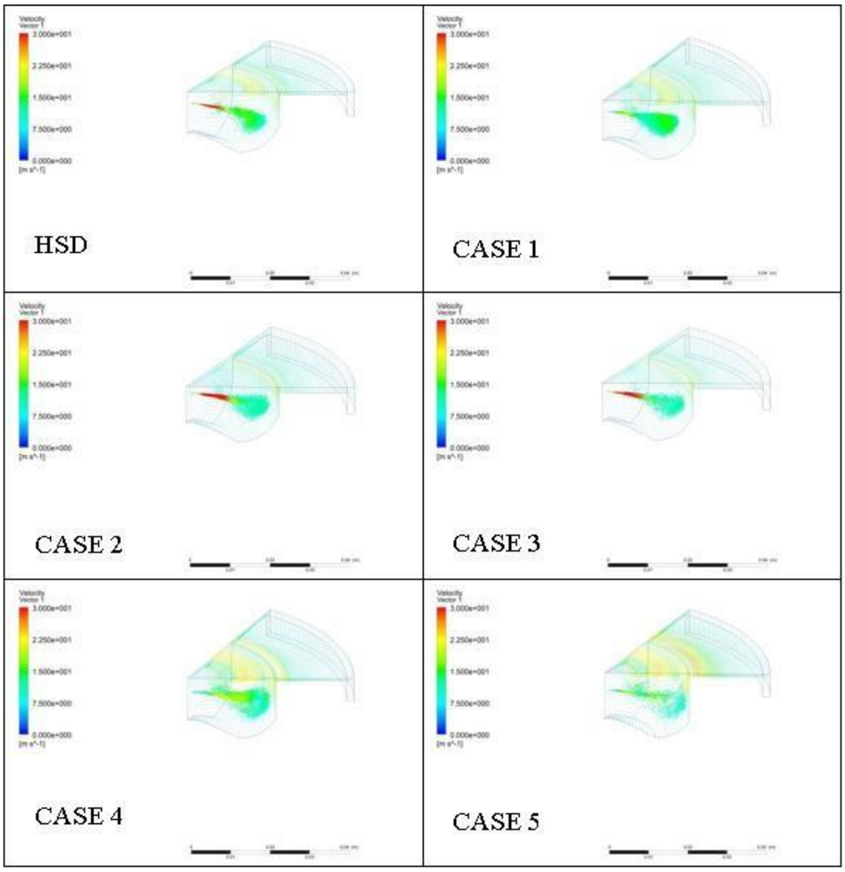

Figure. 7. Velocity of Fluid at TDC

B. Velocity

Figure 7 is a visualization of the velocity of the fluid in the combustion chamber at TDC. From the velocity of fluid visualization, it can be seen that the faster the airflow, the redder the color will be seen on the velocity vector. It can be seen in the figure that changing the geometry of the piston crown can change the shape of the fuel injection from the injector. From the picture above, it can be seen that the reduction in the depth of the piston bowl and the addition of TDC clearance (in case 2 and case 3) indicates that the fuel distribution is not good, whereas with the addition of the piston bowl depth and the reduction in TDC clearance (in case 4 and case 5) may cause more diffuse injection of fuel through the injector. This is due to the change in the swirl ratio that affects the spread of fuel injection.

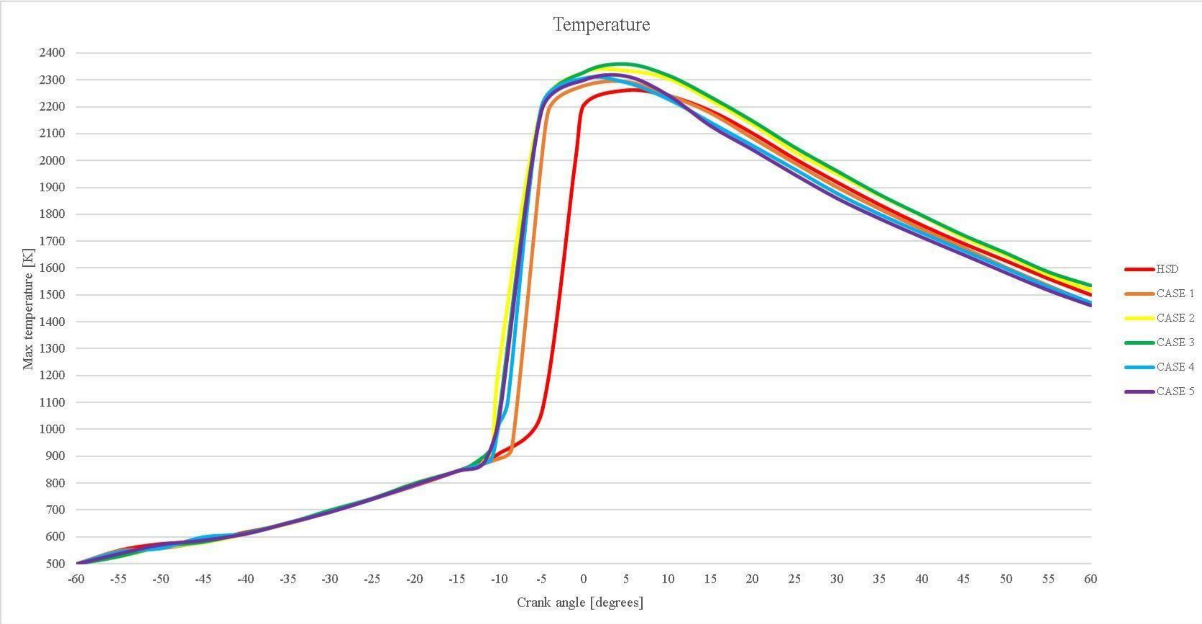

Figure. 8. Temperature

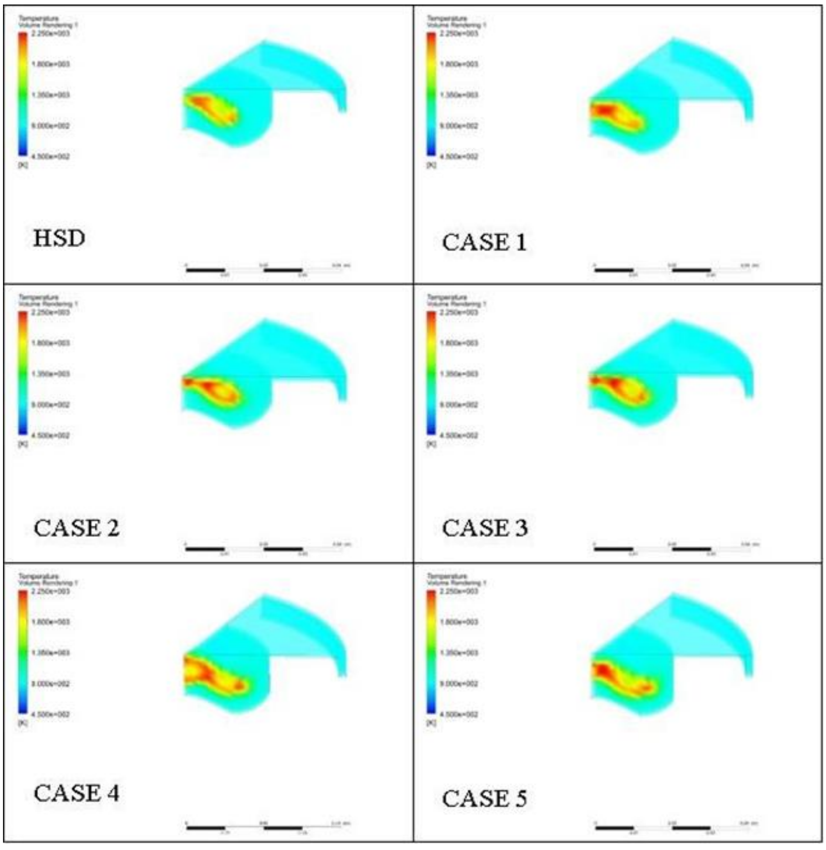

Figure. 9. Temperature Distribution at TDC

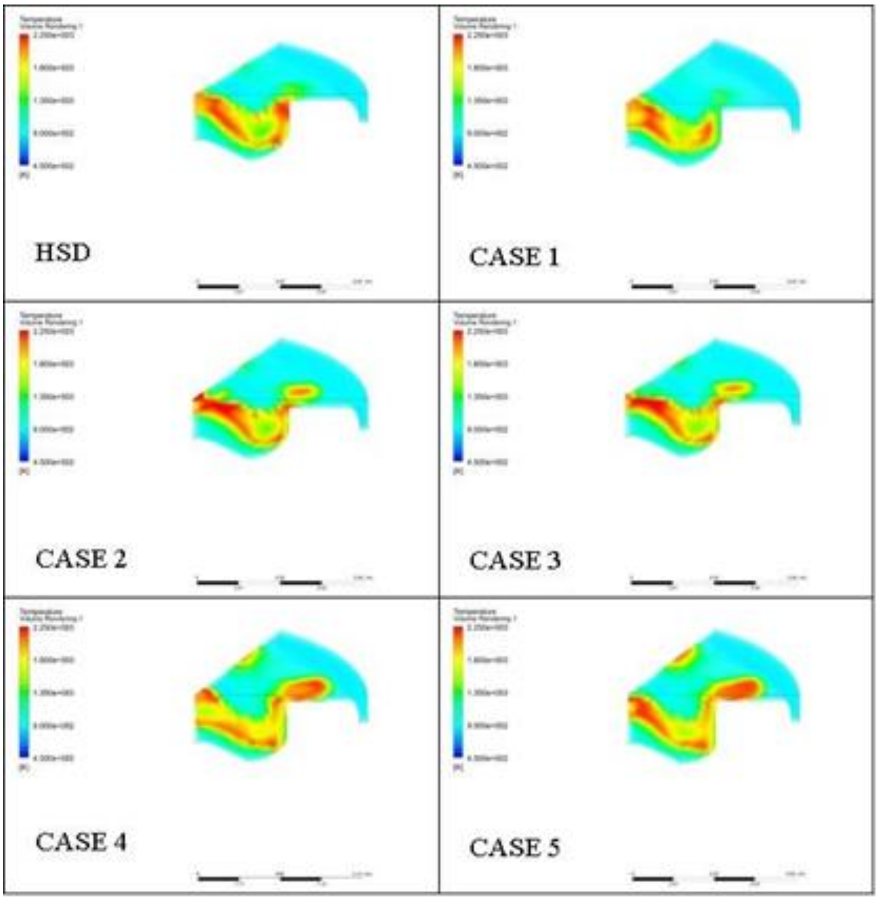

Figure. 10. Temperature Distribution at TDC

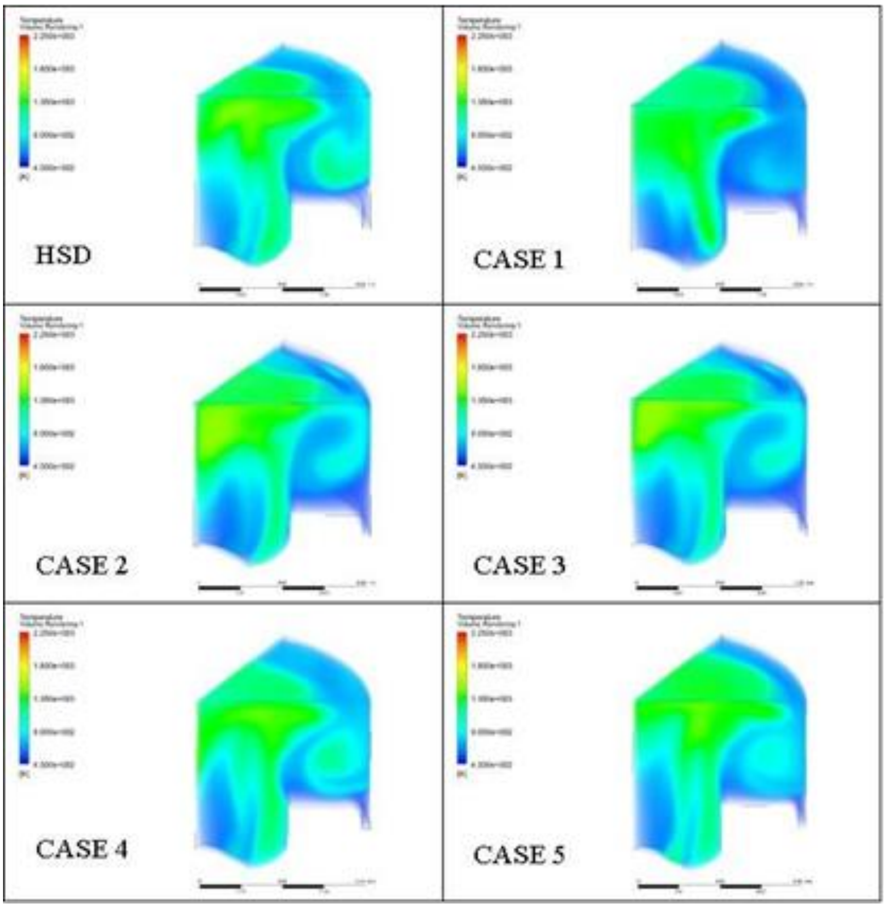

Figure. 11. Temperature Distribution on 60˚ CA ATDC

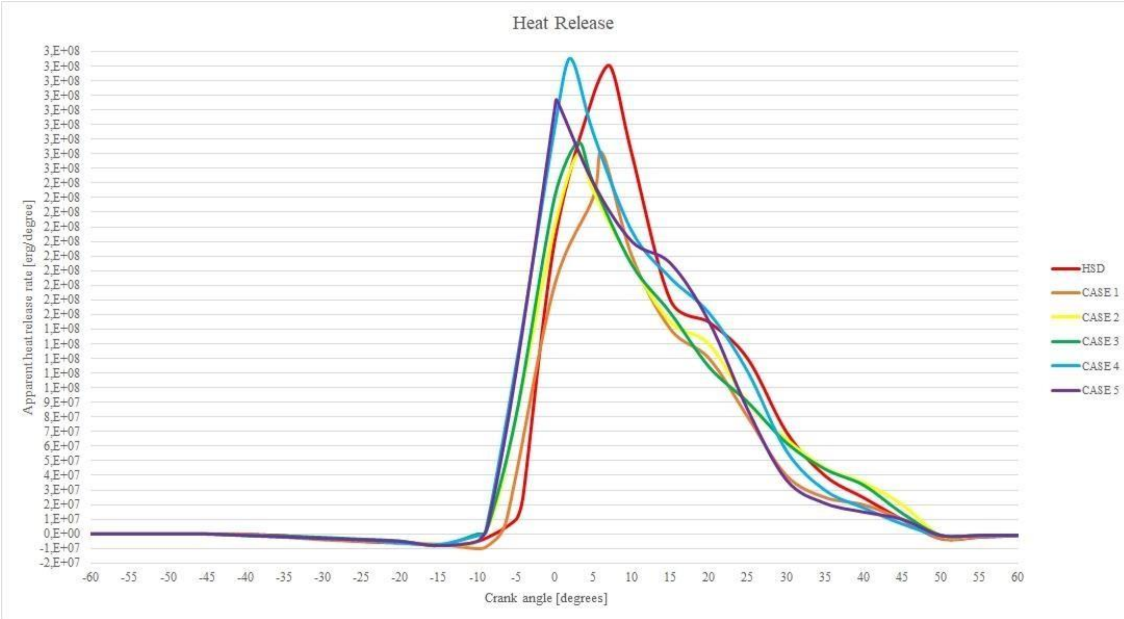

Figure. 12. Heat Release Rate

C. Temperature

Figure 8 is a temperature graph with a variety of models. From the results of the iteration, case 1 produces a maximum temperature of 2290 ˚K at 5˚ crank angle after TDC, case 2 produces a maximum temperature of 2345 ˚K at 4.11˚ crank angle after TDC, case 3 produces a maximum temperature of 2357 ˚K at 4.04˚ crank angle after TDC, case 4 produces the highest maximum temperature of 2373 ˚K at 5˚ crank angle after TDC, and case 5 produces a maximum temperature of 2318˚K at 5˚ crank angle after TDC.

However, when the TDC position, as in the table, case 1 has a maximum temperature of 2270 ˚K, case 2 has the highest maximum temperature of 2328 ˚K, case 3 has a maximum temperature of 2322 ˚K, case 4 has a maximum temperature of 2305 ˚K, and case 5 has a maximum temperature of 2290 ˚K. This is due to differences in the fuel ignition period.

From the simulation results, the normal piston ignition process using HSD fuel starts from 5˚ crank angle before TDC, whereas if using biodiesel fuel B30 (case 1) starts from 9˚ crank angle before TDC, case 2 starts from 10˚ crank angle. Before TDC, case 3 started at 11˚ crank angle before TDC, case 4 started at 12˚ crank angle before TDC, case 5 started at 12˚ crank angle before TDC. From these results, it can be concluded that the fuel can affect the ignition delay. Ignition delay in case 1 with HSD fuel is longer when compared to biodiesel fuel B30.

With this simulation, visualization of temperature distribution is obtained which is taken from -60˚ crank angle before TDC to 60˚ crank angle after TDC.

Figure 9 is a visualization of the temperature in the combustion chamber at TDC. It can be seen in the picture above that HSD fuel is more difficult to burn than biodiesel fuel B30. The high temperature is caused by the ignition process as a result of injecting fuel into the combustion chamber.

Figure 10 is a visualization of the temperature in the combustion chamber at a 10 ˚ crank angle after TDC. At this position, the temperature distribution on the HSD case and case 1 is getting closer to the cylinder head. Meanwhile, the spread of heat from case 2 and case 3 has flowed between the top piston and cylinder head towards the cylinder liner, while the temperature distribution in case 4 and case 5 is more spread out compared to other types of pistons.

D. Heat Release Rate

Based on the iteration results obtained from the case 1 simulation results in a heat release rate of 2,6 x 10^8 erg/degree at 6,2˚ crank angle after TDC, case 2 results in a heat release rate of 2,64 x10^8 erg/degree at 3,02˚ crank angle after TDC, case 3 produces a heat release rate value of 2,67 x10^8 erg/degree at 3,15˚ crank angle after TDC, case 4 produces a heat release rate of 3,2 x10^8 erg/degree at 2˚ crank angle after TDC, while case 5 produced the highest heat release rate value of 2,97 x10^8 erg/degree at 0,175˚ crank angle after TDC.

But when the TDC position, case 1 has a heat release rate value of 1,7x10^8 erg/degree, case 2 has the lowest heat release rate value of 2,35x10^8 erg/degree, case 3 has a heat release rate value of 2,3x10^8 erg/degree, case 4 has a heat release rate value of 2,77x10^8 erg/degree, and case 5 has a heat release rate value of 2,8x10^8 erg/degree.

As seen in the graph, the trend of the ignition process when using B30 biodiesel fuel is faster than using HSD fuel. If a normal piston uses Biodiesel B30 fuel, the maximum heat release rate value decreases by 6x10^7 erg/degree which results in the smallest heat release rate value occurring in case 1.

However, with the modification process in the piston bowl, the heat release rate can be increased. As in the graph, case 4 has a heat release rate of 3,25x10^8 erg/degree at a 2˚ crank angle after TDC.

IV. Conclusion

The conclusions that can be drawn from this research are as follows:

1. Different contours of piston crowns with the same compression ratio can affect the swirl ratio changes. With the reduction of the piston bowl depth and the addition of TDC clearance, it can reduce the swirl ratio. On the other hand, the increase of the piston bowl depth and reduce the TDC clearance, can increase the swirl ratio. The highest swirl ratio is seen in Case 5 with a maximum value of -1,15 at 4.5˚ crank angle before TDC. Changes in the swirl ratio value can also affect the spread of fuel injection.
2. Different contours of piston crown with the same compression ratio can affect the temperature distribution in the combustion chamber. The highest temperature distribution can be seen in Case 4 with a maximum temperature value of 2373 K at 5˚ crank angle after TDC.
3. Changing the fuel from HSD to Biodiesel B30 can reduce the heat release rate of the engine with a decrease of $6x10^7$ erg/degree. With the modification of the piston bowl, it can be concluded that Case 4 has the highest heat release rate of $3,25x10^8$ erg/degree at a 2˚ crank angle after TDC.

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