

# Influence of coolant temperature on the performance of a four stroke spark ignition engine employing a dual circuit cooling system

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**Abstract:** Diesel engines have attracted attention because of their higher thermal efficiency and lower carbon dioxide emissions than gasoline engines. On the other hand, oxides of nitrogen (NO<sub>x</sub>) and particulate matter (PM) in the exhaust from diesel engines are difficult to be reduced simultaneously because a decrease in one is likely to result in an increase in the other. In this paper, an attempt has been made to compare the effect of the heat lost to the coolant in the cylinder block, and in the cylinder head with a conventional cooling circuit with corresponding effects in an engine with a dual cooling circuit. A dual cooling circuit employs separate cooling circuits for the cylinder block, and for the cylinder head. A single cylinder, four-stroke, water cooled, naturally aspirated spark ignition engine test rig was developed for the purpose. The influences of the cylinder block and cylinder head temperatures on brake specific fuel consumption (bsfc) and on exhaust emissions were measured. The results show that raising the temperature of the coolant in the engine block can produce significant improvements in bsfc with a corresponding reduction in the hydrocarbon (HC) emissions. Similarly, lowering the coolant temperature in the cylinder head can increase the knock limit of the engine with a corresponding reduction in the levels of NO<sub>x</sub> in the exhaust emissions. The objective of this investigation was to access the magnitude of the likely benefits of the dual circuit cooling system.

**Keywords:** dual circuit cooling system, spark ignition engine, heat transfer, engine cooling

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## 1 Introduction

Controlling the engine cooling in a flexible and controlled way, compared to the conventional cooling system significantly improves fuel economy of spark ignition (SI) engines (Pang and Brace, 2004). Increasing and adjusting cylinder block wall temperature by increasing the coolant temperature at the engine inlet, substantially reduces HC emissions (Finlay et al., 1989). These improvements are attributed to increase cylinder wall temperature specifically in the lower region of the cylinder block. The higher wall temperature delays flame quenching on the wall as the quench layer

thickness gets reduced. Also, corresponding improvement in the expansion work of the engine has been reported due to improved HC combustion in this stage (Finlay et al., 1989). With the dual cooling system a separate cooling circuit was used to cool the cylinder head. Higher cylinder head temperature results in a direct effect on the NO<sub>x</sub> emissions as they are temperature sensitive (Heywood, 1988). Slight reduction in the combustion chamber temperatures by reducing coolant temperature in the upper region of the cylinder head reduces NO<sub>x</sub> which is temperature governed. Also, volumetric efficiency is improved in the case that the inlet manifold temperature falls due to extra cooling of the cylinder head. This is due to the increase in air density at lower temperature (Heywood, 1988). This is because that the inlet manifold is

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connected to the cylinder head, and thus, lowering the temperature in this region reduces the manifold temperature. The results suggest that by using a dual cooling circuit very important benefits are derived like increased knock resistance, improved bsfc, reduced HC and NO<sub>x</sub> emissions. Further, the literature suggests that higher cylinder block temperature reduces the frictional losses associated with the piston and ring pack, and this also leads to reducing fuel consumption, especially at part load (Kobayashi et al., 1994). Finlay (1988) experimented with a precision cooling system in which 40% less coolant flow rate is required. In precision cooling finer coolant galleries in the cylinder head increase the coolant flow speed from a maximum of 1.4 m/s in conventional galleries to more than 4 m/s. This can be attributed to increase convective heat transfer (Finlay et al., 1988).

The concept of control component temperature cooling (etc.) is another important method being explored by many researchers (Pang and Brace, 2004). Up to 20% in fuel savings are obtained at part of load conditions, and also HC emissions are reduced. This method is similar to a dual circuit system which the cylinder liner is maintained at a higher temperature with respect to the head. (Willumeit, 1984)

Chanfreau et al. (2003) reported that fuel savings of up to 2%-5%, a 20% reduction of carbon monoxide (CO), and 10% reduction in HC with a higher coolant temperature set point of 110°C, in comparison to the 90°C in a conventional cooling system. Metal temperature is consistently 10°C higher than that in a conventional system when running at steady-state conditions.

Changing the heat transfer mode lowers fuel consumption as well as the auxiliary power requirement. Ap et al. (1999) replaced the mechanical coolant pump, rated at about 1 kW, for a 1.2 L gasoline engine with a smaller electric coolant pump, rated at 30-80 W, by employing an evaporative and cooling system. The advanced cooling system results in a large step reduction in the coolant flow rate and circuit pressure, intensifying the evaporative effect. The low flow rate and circuit pressure reduced hydraulic losses significantly, allowing the system to have at least 25% of the original flow rate

even when pump power was less than 10% of the original system (Ap et al., 1999).

Chanfreau et al. (2003) reported the benefit of raising the operating temperature obtained by using an electric water-pump rated at 600 W for a 3.8 Liter gasoline engine with a power output of 180 hp, in place of a mechanical coolant pump, typically rated at 2-3 kW. The reduction in coolant flow rate with a smaller coolant pump was made possible by raising the coolant operating temperature from about 90°C to 110°C. This represents significant fuel savings, particularly at part load conditions where a mechanical coolant pump would draw a significant portion of engine power to cool the engine even when it was not required. They also reported a 15% reduction in CO output and a 17% reduction in unburned HC, with a higher operating temperature set point.

Couetouse and Gentile (1992) targeted a maximum 140°C operating temperature for the engine oil by regulating coolant temperature within the range of 90-115°C, depending on the engine power output. This allowed a reduction of up to 10% in fuel consumption during a part-load operation.

Nishano et al. (2004) reduced the knock characteristic by intensifying the cooling rate in the head of the engine. New coolant passages were added among the inlet valves and the thickness of the wall was reduced at various points in the head. This resulted in decreased temperature of the end gas by about 5°C that helped to allow an increase in spark advance angle of about 1°C.

Vagenas et al. (2003) suggested the possible benefit of a controlled cooling jet in the engine cooling gallery, and the results showed a significant reduction in the coolant flow rate and saved a part of the energy that was otherwise wasted for cooling. In addition to these,, the system responded to temperature changes very quickly. Subsequently, using an internal combustion engine cooling gallery simulator a specific series of experiments were undertaken to demonstrate the enhancement of heat transfer conditions at the liquid/metal interface and also the potential for reduced coolant volumes.

Kubozuka et al. (1987) conducted a fundamental heat transfer study which heat was removed from the engine

through the boiling process in the water jacket and subsequently radiated to the air through a condenser. As reported, the heat transfer rate was around 1,000 kW/m<sup>2</sup> and a more uniform temperature distribution over the cylinder was observed.

## 2 Materials and methods

In a split, dual cooling system, the cylinder head and the cylinder block of the SI engine are cooled by independent coolant circuits. This system facilitates flexibility in regulating the temperature of the cylinder block and the cylinder head of the engine. A split cooling system gives a unique advantage to the engine as it allows each section of the engine to operate at its optimum temperature set point, maximizing the overall effect of the cooling system on engine performance. Each circuit operates with a different coolant temperature set point or flow rate to create the desired temperature distribution in the engine.

### 2.1 Experimental setup

The engine used for the development of the test rig to conduct the investigation was a naturally aspirated four stroke, single cylinder water cooled diesel engine. Engine details are in Table 1. The diesel engine was converted to SI engine mode with reduction in the compression ratio from 17.5:1 to 9:1. A blind metal plate with required outer dimensions of the engine was used to cover the top of the cylinder block to stop the coolant flow from the block to the cylinder head. The metal plate thickness was 9mm which was calculated to get the required decrease in the compression ratio. The experimental setup, along with the photograph of the test rig is shown in Figure 1 and Figure 2 respectively. Different thermocouples were fixed to the wall of the cylinder block and on the inner surface of the cylinder head. The locations of these thermocouples are shown in the Figure 3.

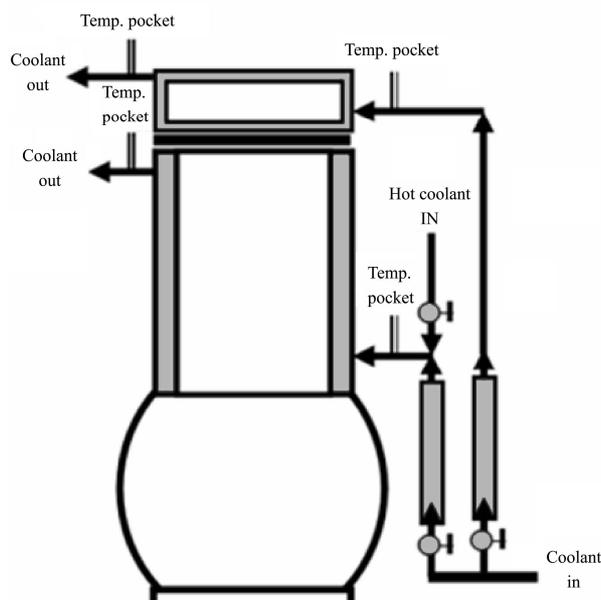


Figure 1 Experimental setup

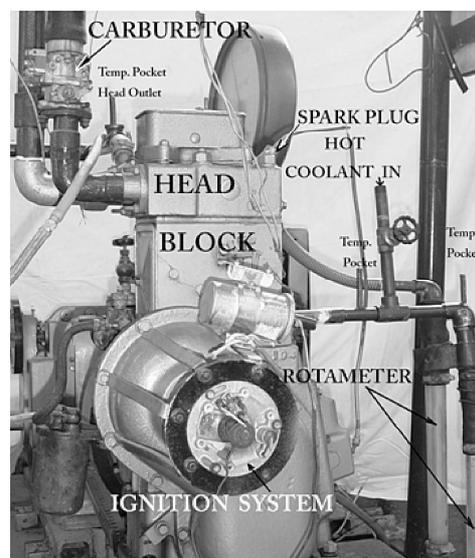


Figure 2 Photograph of the test rig

Fuel flow rate was measured by a measuring buret type fuel flowmeter. Exhaust emissions of HC and CO were measured by an AVL-444 exhaust gas analyzer.

Table 1 Details of the engine

| S.No. | Engine            | Specifications       |
|-------|-------------------|----------------------|
| 1.    | Bore              | 87.5 mm              |
| 2.    | Stroke            | 110 mm               |
| 3.    | No. Of Cylinders  | One                  |
| 4.    | Compression ratio | 17.5:1/(9:1 SI)      |
| 5.    | Power & Speed     | 5.5kW at 1,500 r/min |

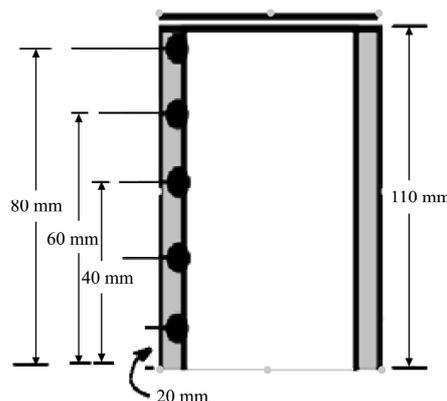


Figure 3 Location of the thermocouples on the cylinder wall

### 2.1.1 Ignition system

The cylinder head opening for the diesel injector was used for fitting in a small diameter sleek spark plug as shown in Figure 2. A sleeve was used to compensate for the difference in the diameter of the injector and the spark plug. The sleeve was designed to seal the combustion chamber of the engine. A cam shaft was designed and manufactured for operating the contact breaker type of ignition system. A shaft coupling was used to mount the contact breaker cam shaft as an extension to the engine

cam shaft which is used for operating the engine valves. The contact breaker assembly was supported on a bracket fitted on the engine body as shown in Figure 4. The ignition system was operated using a battery-ignition system with the contact breaker cam shaft fitted on the engine. The spark timing was set with reference to maximum brake torque adjusted with the help of the contact breaker while conducting a separate test not presented in this discussion.

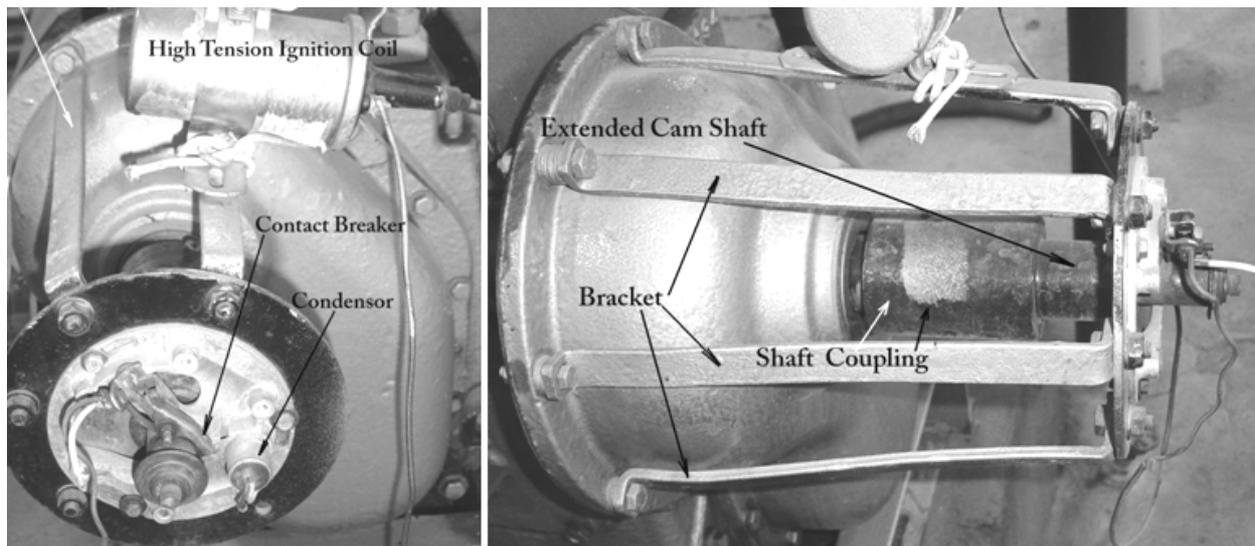


Figure 4 Ignition system mounted on a bracket with extended cam shaft

### 2.1.2 Carburetion

Preparation of the fuel and air mixture was done in the carburetor. A suitable carburetor to be matched with the engine capacity was mounted on the inlet manifold. Figure 2 shows the location of the carburetor on the engine.

### 2.1.3 Dual circuit cooling test rig

The existing conventional cooling system was modified for dual circuit cooling. As stated earlier the coolant flow from the lower cylinder block was stopped by placing a blind plate sandwiched between two engine head gaskets with the block on the lower side and the head on the top. Water passage was provided with its entry at the lowest section of the cylinder block, similar to the conventional system, but an outlet was provided at a point 100 mm above the bottom dead centre. Similarly, a separate inlet to the head was provided, and the outlet for the coolant in the cylinder head was the

same as that the conventional engine cooling system. The coolant supply inlets to both the circuits were fitted with an individual rotameters to observe the coolant flow rate to the two circuits. The block coolant circuit was provided with the heated coolant input to manipulate and increase coolant temperature in the block, as shown in Figure 1. All supply lines were regulated with the valves to control the flow rate. Similarly, Figure 1 and Figure 2 show the coolant inlet and outlet temperature pockets which were provided to monitor these temperatures.

## 2.2 Test procedure

Before conducting the test on the cooling system the ignition timing was optimized for maximum brake torque and the tuning of the carburetor was done to optimize the bsfc. The optimization results are not shown here. Tests were conducted in steps in the range 5% to 85% of the rated load conditions. These tests were carried out

with controlled cooling of the cylinder head without any significant temperature change from the values obtained during the tests conducted with the conventional system. However, the cylinder block temperature was maintained at desired higher wall temperature to conduct the test. Keeping the cooling rate in the head nearly fixed, the block outlet temperature was set to an increased level. The bsfc was reduced as well as the exhaust emissions of CO and HC. The procedure adapted for the tests at part load conditions involved first setting the coolant temperature, leaving the cylinder head at 10°C lower than the conventional system. The coolant temperature entering the block was then set which gave the desired cylinder wall temperature. All tests reported here were conducted at (1500±15) r/min. Engine speed was maintained by adjusting the throttle valve on the carburetor, as the engine was loaded by a Heenan Froud hydraulic dynamometer.

### 3 Results

The effects of load on bsfc, as well as on HC and CO emissions were compared between dual circuit and conventional cooling systems as shown in Figure 5 to 7. The temperature readings were obtained by using thermocouples and the temperature profiles which had been plotted in Figure 8 for dual circuit cooling in comparison with conventional cooling.

#### 3.1 Effect of load on bsfc

As shown in Figure 5, the bsfc of the engine decreased for both cooling systems as the amount of air-fuel mixture combusted increased in order to maintain a constant speed with increasing load. The values of bsfc at low loads were high because most of the energy from the burning of the charge supplied to the engine was needed to overcome frictional forces in the engine. In this case at a constant engine speed, the friction was nearly constant irrespective of load. Consequently, as engine loads increased an increased proportion of the energy from the burning of the charge was delivered as brake power, causing bsfc to decrease. The dual circuit cooling system performed better with lower bsfc values during mid-engine-load conditions. It is evident in Figure 5 that at low and high load levels the bsfc is

comparable for the two systems. At lower loads the bsfc obtained with the dual circuit cooling system was related to a 10°C lower temperature in the cylinder head, which affected fuel vaporization and subsequent combustion. Combustion of the premixed charge was affected by the vaporized fuel to air ratio (Heywood, 1988). Lower cylinder head temperature affected the percentage of vaporized fuel in the premixed charge during the lower load conditions. Figure 5 shows similar bsfc values for both the cooling systems at higher loads. In this range of engine operation, the 10°C lower temperature of the cylinder head as compared to that with the conventional cooling system had a diminishing effect on bsfc, whereas the higher block temperature dominates the combustion process. Taylor (1968) and Heywood (1988) have discussed the effect of end charge temperature on knocking as well as the effect of ignition timing on engine performance. The effect on bsfc in the higher load operating range could be attributed to higher end charge temperature, due to higher wall temperature near the top dead centre location and also to ignition timing optimization. The coolant temperature in the head and block used in this present work could be further manipulated and the ignition timing which was optimized for the mid-speed range could be further varied for higher load operations.

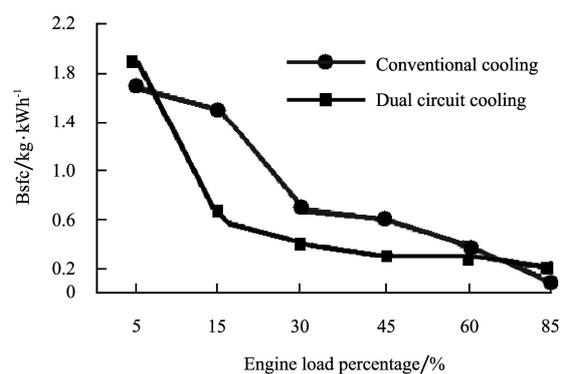


Figure 5 Effect of load on bsfc at constant speed

An appreciable reduction in bsfc was observed between 15% and 60% of rated load. This could be attributed to higher temperatures on the wall of the cylinder block. The higher wall temperature delayed flame quenching on the cylinder wall as the quench layer thickness got reduced. Corresponding improvement in

the expansion work of the engine has been reported due to improved HC combustion in this stage, which is evident in Figure 6. An extended after- burning phase reduced bsfc and reduced HC emissions. Specific fuel consumption for the dual circuit cooling system being lower than with the conventional cooling system was reflected in the improvement in performance of the system (Figure 5).

**3.2 Effect of load on HC emissions**

Figure 6 shows the effect of load on HC emissions. As the load was increased, the HC emissions increased for both the systems. Due to the wall quenching effect, the fuel particles near the wall surface did not take part in the combustion process, and escape combustion. Heywood (1988) has related HC emissions to crevice volumes near the cylinder wall and to the oil absorption/desorption effect. During dual circuit cooling higher wall temperature results in extended combustion duration, and thus the HC levels were lower than with the conventional system of cooling (Guillemot et al., 1994).

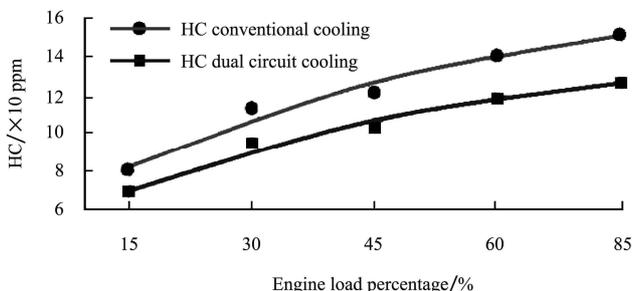


Figure 6 Effect of load on HC emissions at constant speed

**3.3 Effect of load on CO emissions**

Figure 7 shows the effect of load on CO emissions. Increase in the load resulted in the rise in CO emissions in both the systems. The quantity of CO emissions (by percentage volume) with dual circuit cooling was less than that with conventional cooling. CO emissions in SI engines were mainly due to incomplete combustion. Improving fuel combustion not only reduced HC emissions, but also had a direct effect on CO emissions. Another reason for lower CO emissions could be attributed to higher volumetric efficiency which was due to a 10°C lower temperature in the cylinder head.

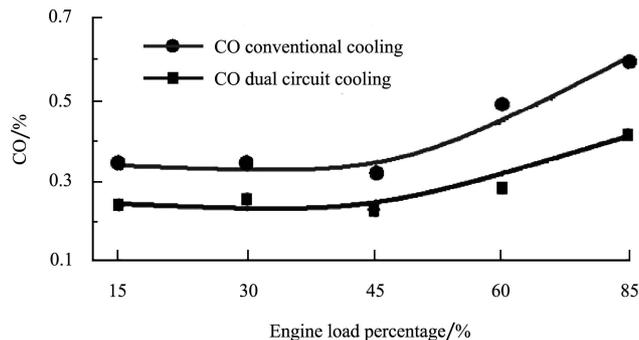


Figure 7 Effect of load on CO emissions at constant speed

**3.4 Comparison of average temperature profiles between dual circuit and conventional cooling systems**

Figure 8 shows a plot of average temperatures calculated from the observations taken from different thermocouples located at different positions from the top dead centre position as shown in Figure 3. The thermocouples were embedded on the inner wall of the cylinder block on the coolant side. Thermocouple observations were taken for both of the cooling systems at different loads. However, Figure 8 shows only the average temperatures calculated for load levels of 5% and 85% of the rated load. The average temperatures plotted for the dual circuit cooling were higher than those with conventional cooling at different load conditions. Figure 8 shows that the dual circuit cooling system had higher wall temperatures at all loads.

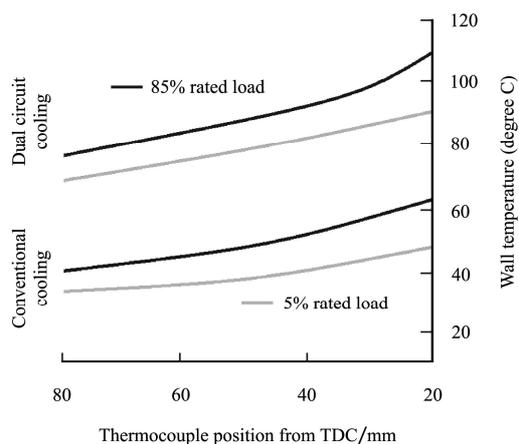


Figure 8 Average cylinder wall temperature profiles for dual circuit and conventional cooling systems at lowest (5%) and highest (85%) of rated load

**4 Conclusions**

The following conclusions can be drawn from the

experiments on dual circuit cooling in an SI engine:

1) There is a decrease in bsfc due to a decrease in the friction losses and better combustion since higher temperature is maintained in the block with the dual circuit system.

2) Raising the temperature of the walls to around 80°C to 100°C reduces the bsfc in the range of 4%-6%.

3) At 5% of the rated engine load the bsfc with the dual circuit cooling system is higher than that with the conventional cooling system. This can be improved by providing reduced cooling in the cylinder head to improve vaporization of the fuel before combustion.

4) At 85% of the rated engine load the bsfc with the dual circuit cooling system is again higher than with the conventional cooling system. This situation can be improved by further reducing the cylinder head temperature from the present fixed set point. In addition

to this, at higher engine loads the ignition timing can be varied and optimized for that range of operation.

5) With dual circuit cooling lower HC emissions are obtained due to a reduced wall quenching effect. Results reveal about 12%-15% reductions in HC emissions.

6) Since a lower cylinder head temperature is maintained with dual circuit cooling, as opposed to with conventional cooling, the temperature around the spark plug, as well as the inlet and exhaust valves is also reduced, thus the knock limit of the engine will be extended.

7) The results obtained from the experiments show that by using a dual circuit cooling system; the temperature of the cylinder block in the lower region towards bottom dead centre is about 30-40°C higher than for the case of the conventional cooling system.

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