The Effect of Thermal Boundary Conditions on Knock Characteristics in a Single Cylinder Spark Ignited Engine

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Abstract: In order to develop a high efficiency spark-ignited engine in future, increasing the compression ratio is essential. Cooling strategy is one of the most practical ways for knock mitigation. Characteristics of knock occurrence under various cooling conditions were investigated in this study.

Using downhill simplex (Nelder-Mead) method for optimization and 3-D stress analysis, the linkage system with secured durability was developed to facilitate the piston temperature measurement and the flame detection. Using the developed device, knock mitigation effect of the coolant water was observed. The load increase (DBL expansion) was tried, while maintaining the same knock frequency with independent cooling strategies on the head, cylinder and the liner.

As a result, 31.4% and 18.4% of DBL expansion was achieved at 1500rpm and 2000rpm, respectively. And within the same load, ignition timing advance potential was also investigated, through further knock suppression with cooling. Specification of knock spot in some conditions with in-cylinder visualization and flame propagation shape with gasket ion probe were conducted in this study.

1. Introduction

One of the great issues in the history of spark-ignited engine development, is the increase of compression ratio, since it has a direct relationship with the efficiency. Recently, as norms for emission and fuel efficiency are getting more stringent globally, companies with mass production and a lot of researchers are endeavoring to develop engines with higher efficiency.

However, increasing compression leads to a more knock-prone in-cylinder condition, with respect to the combustion process. If the compression ratio is increased, due to the auto-ignition occurrence of the higher end-gas temperature during the expansion stroke, the engine certainly gets damaged and the customers may also feel annoyed with the ringing sound derived from the knock phenomena.

Recent mass-produced engines have already reached brake thermal efficiency of over 40%. And through massive efforts on knock mitigation under high compression ratio conditions, a 50% efficiency is now being targeted.

There are some practical methods to mitigate the knock phenomena under high compression ratio operating conditions. One is the enhancement of flame speed in the combustion process. This simply means making the flame propagation process faster, to augment efficiency and suppress knock occurrence. Using an advanced ignition system such as high energy ignition [1, 2, 3, 4, 5] is one of the ways to introduce fast combustion. Introduction of high-tumble port to augment the turbulent energy at the end of compression stroke is also one of the most effective ways [6, 7, 8, 9, 10].

The second way is to improve the EGR system. Introducing EGR is also a promising method to suppress knock behaviour, by reducing the temperature and reactivity of the unburned end-gas in the cylinder [10, 11]. EGR system is being applied very aggressively and many studies have shown the effect. Improvement of the distribution of EGR of each cylinder, optimization of manifold features [12], use of cooled EGR and improvement of the EGR cooling unit [1, 11] have been investigated.

The third one is mitigation with engine cooling strategy. This method is one of the most practical ways for mass-produced engines. Many studies have investigated the effect of structural change of the water jacket, improvement of the temperature distribution [11, 12] and the enhanced cooling of combustion chamber, like the introduction of hollow valve [1]. Knock mitigation with split cooling system or dual-loop cooling system has already been identified by many studies. With efficient cooling, further knock mitigation, while reducing piston ring pack friction loss, has been achieved. [12]

This study examines the effects of the cooling water on the head and the block under high compression ratio of 12.5. A total of 12 thermocouples were used in this study, including 5 points of the piston, to measure the steady state temperature of the combustion chamber, to determine how it affects the knock behaviour. Piston temperature was also controlled by applying oil supply to piston gallery experimentally [1, 13, 14, 15]. Further, the relationship between temperatures and the location of knock occurrence, is shown with in-cylinder knock visualization method and flame propagation shape measurement. In order to measure data like temperature and flame shape, optimized linkage system and ion probe flame detection system were used.

2. Linkage system optimization

In order to measure signals of thermocouple or strain gauge of high speed moving system, Telemetry system [16] or linkage system is needed, generally. However, telemetry system is expensive and the frequency is not enough to measure signals which have multiple channels and frequencies over 1kHz, like flame ionization signals. Furuhama *et al.* developed L-link type linkage system to measure the piston temperature distribution. [17, 18] However, additional devices to prevent wire fracture have defects and need a major change in the base shape of the engine, like putting a device on the piston pin.
The basic design of the linkage system is to minimize the disconnection of signal lines without changing the base structure. Thus, in order to prevent the fracture from being broken through a typical folding, wires are designed to receive only torsion from each link. To determine the structure of the linkage system for minimizing the fracture of sensor signal wires, downhill simplex (Nelder-Mead) method [19] was applied for the unconstrained non-linear optimization. The maximum value of the torsion angle on each link for wire in the specified engine geometry was set as the objective function and minimized by the optimization process. Totally, 27 sections for the optimization were derived from dividing each parameter (lengths of each link) into three length ranges.

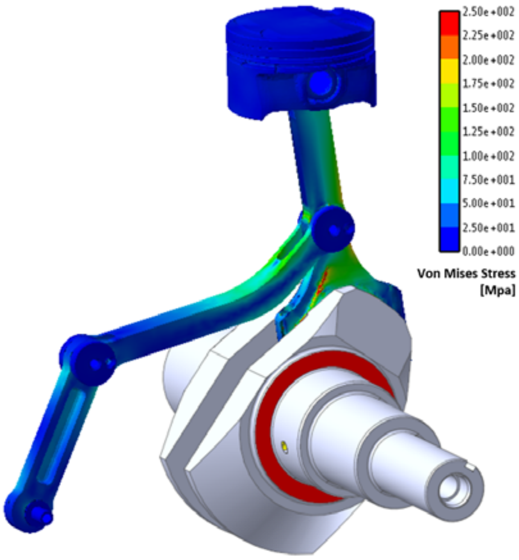


Figure 1: Result of 3-D stress analysis at 6000rpm

Even though the optimization process can give the minimized objective function, which is the structure that minimizes the signal wire fracture, in terms of the stress on the structure, specified parameters (linkage lengths) still may not the best. Thus, 3-D dynamic stress analysis was conducted to find the best structure. RecurDyn V8R4 analysis tool was used for the modal and nodal analysis. As a result, 87mm for Linkage1, 155mm for linkage2, 63.5mm for Rev.Joint were the best and showed a safety factor over 2, at 6000rpm.

3. Experimental Setup and Conditions

3.1 Engine setup and measurement

Table 1 shows the detailed engine specification for the experiment in this study. A 495cc single cylinder naturally-aspirated spark-ignited engine was used. A new design for the piston was used to increase the compression ratio up to 12.5.

Engine type	Single-cylinder N/A
Displacement	495cc
Bore	81 mm
Stroke	96 mm
CR	12.5 : 1
EVO / EVC	bBDC 68° / aTDC 1°
IVO / IVC	aTDC 10° / aBDC 67°
Max valve lift	10mm (both)

Table 1: Experimental engine specifications

In-cylinder pressure was measured with a Kistler 6056A flush-mounted piezoelectric pressure sensor to avoid cavity resonance [20]. AVL Indi Micro IFEM was used for the amplifier of in-cylinder pressure signal. Intake manifold pressure was measured with a Kistler 4045A2 absolute pressure sensor and amplified with Kistler type 4603 pieszo-resistive amplifier. Cylinder pressure pegging was conducted by setting the mean pressure at intake BDC (± 2 CA) equal to the mean pressure of the intake port, assuming the air flow stops. The pressure signals was meausred with AVL Indimodule device and total 300 cycles were logged to caclulate the result values. Pressure data were sampled with 0.1CA resolution, using 3600 teeth encoder, for precise frequency analysis of the knock phenomena.

3.2 Test conditions with temperature control

The head and liner temperature conditions were changed by controlling the temperature of the coolant, independently. The coolant temperature was controlled from 80° to 60°C for each. The flow rate was sufficient to introduce the uniform outer thermal boundary condition, as much as possible. The temperature difference between the inlet and outlet of the two types of coolant was kept below 2°C under operation.

Engine speed	1500, 2000 rpm	
Coolant temperature control	Head	80, 60, 40°C
	Liner	80, 60, 40°C
	Piston oil-jet	On / off
Intake air temperature	30 ± 3°C	
Fuel inlet condition	30 ± 3°C / 4 bar, dual MPI	

Table 2: Test conditions

Using K-type thermocouples, on the head, 5 points were measured; the middle of in/ex valve positions, front and rear, besides the spark plug. In the liner, front and rear points were measured. Due to the structural problem, sadly, both the in/ex sides of the liner couldn't be measured in this study. All spots were exactly 1mm outside the combustion chamber surface. 5 points of the piston were also measured. In order to decrease the possibility of wire disconnection, specialized thermocouple wires were used for the piston with the linkage system. Approximate locations for each of the measured points are shown in Figure 2.

Under the steady state condition, while the temperature of the liner coolant was varied from 80°C to 40°C, the temperatures of head spots were changed to within 3.2°C. But while the temperature of the head coolant was decreased to 40°C, the liner temperature showed a temperature drop of more than 10°C. It is shown that the increase in heat loss to the head occurs simultaneously with the decrease of heat loss through the liner. In effect, the heat loss of the cylinder head is much bigger than that of the liner during combustion; the change of heat loss through the liner doesn't have a significant effect on the total heat loss of the head.

In all cases in this study, variation of the head coolant temperature didn't affect the piston temperature that much. But piston was also cooled down to around 142°C from 160°C, when the liner coolant temperature was decreased from 80°C to 40°C. This shows that the heat transfer from the piston to the liner through the ring pack is not insignificant.

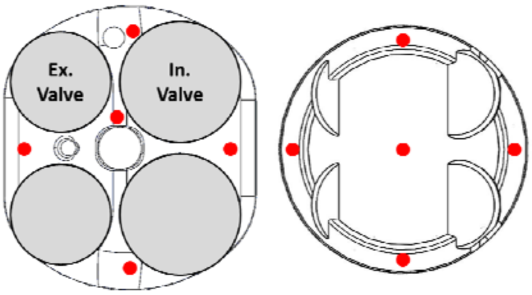


Figure 2: Temperature measurement location; head from bottom view, piston from top view

For piston thermal boundary condition, the temperature was controlled by switching the oil injection to the oil gallery on and off, which is shown in Figure 3. The oil was maintained to 80°C, at the same temperature as the engine supply oil. Without the oil-jet, if the liner coolant temperature was 80°C, the piston center showed 160°C, approximately. If the oil was applied to gallery, the temperature was cooled down to around 120°C.

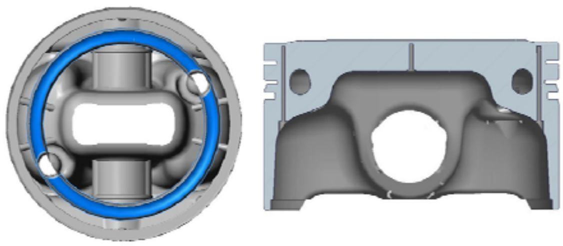


Figure 3: Cooling gallery piston for CR12.5

Figure 5 shows the difference in temperature distribution between two extreme cases. The left side is an example, where the coolant temperature of the head and liner are kept at 80°C, without oil-jet oil injection. In this case, the load was 5.95 bar of IMEP. Liner boundary temperature was 102.5°C at front, 107.3°C at rear. The right side of the figure shows the distribution, while the coolant temperature of the head and liner were maintained at 40°C, simultaneously with oil-jet injection. This case shows the most excessive cooling in this study, but the heat generation from the combustion chamber was bigger than the former case.

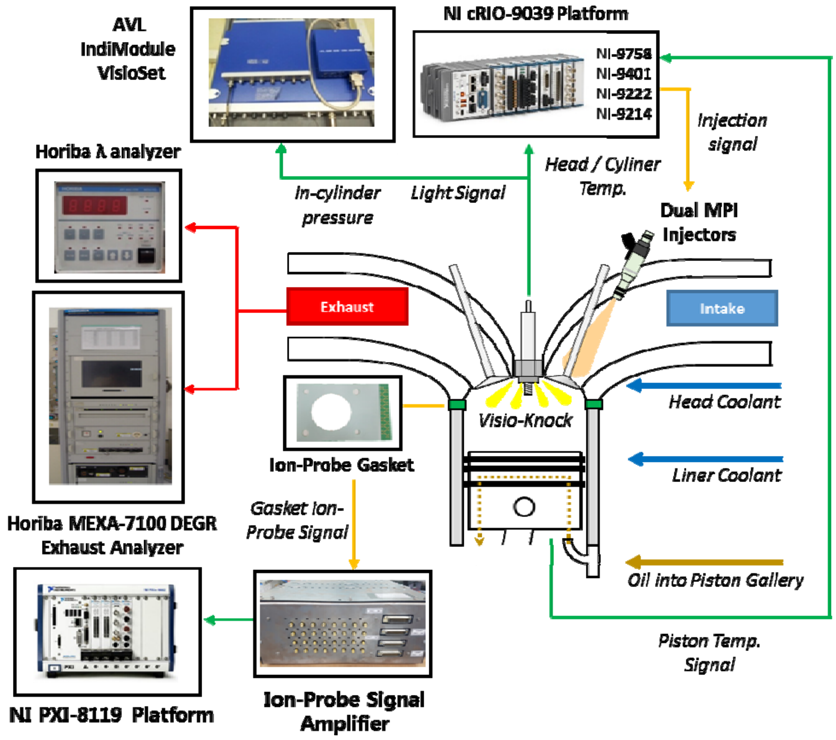


Figure 4: Schematic diagram of the experimental system

The load was 7.7 bar of IMEP which could be increased due to further knock suppression. The liner temperature was 74.5°C at front and 70°C at rear. It shows that the temperature of the rear side of both the head and the liner is a bit higher than that of the front side. It is thought that it is because of the head coolant, which flows from the front side to the rear.

The condition with cooled oil-jet injection (20°C) was tried as well. However, it did not show a better effect because the heat transfer occurs between cylinder block and the oil, while the oil passes the passage in the block. In order to achieve excessive cooling effect of the piston, it is thought that independent oil supply is necessary.

3.3 Knock and flame detection method

In this study, using 4kHz HPF (high-pass filtered) in-cylinder pressure signal, TVE (threshold value exceeded) method was used. If the magnitude was over 0.9 bar, the cycle was decided as knock-occurring cycle. Although measurements using one in-cylinder pressure sensor had been reported to have differences depending on the location [21], it is still the most widely-used method, because of the low cost and reliability for analysis. Knock onset angle was defined with simple TVE method as well, as the crank angle when the HPF signal exceeds 0.2 bar of amplitude. This method generally gives an accuracy of within 0.5CA. [22]

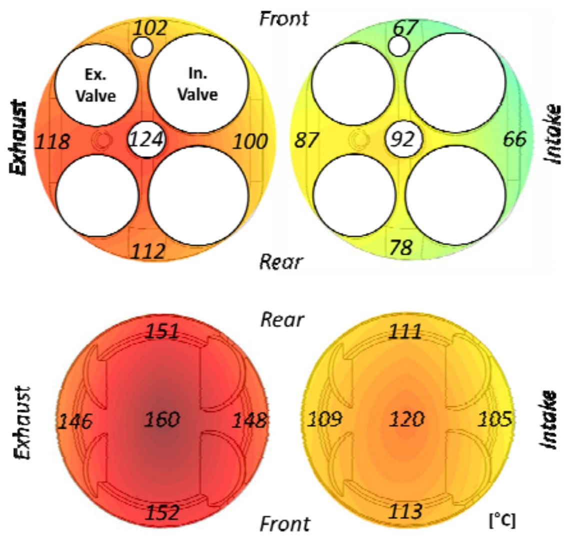


Figure 5: Temperature of thermal boundaries

Using AVL VISIO system, which has 8 windows targeting 30° down from the horizontal line to measure the light intensity of end-gas region, the knock occurring spot could be found statistically. Ion-probe head gasket [23, 24, 25] was also introduced to detect the flame propagation, in order to pinpoint where the knock spot is, during combustion. It has 16 probes with 22.5° for each around the cylinder bore and the original head gasket was replaced to ion probe head gasket in this study. The current of the ion probe signal is very weak, so a specially designed amplifier was used to get the voltage signal. The bias voltage between the detection probe was 250V, and the threshold of the flame signal detection was set to 1V.

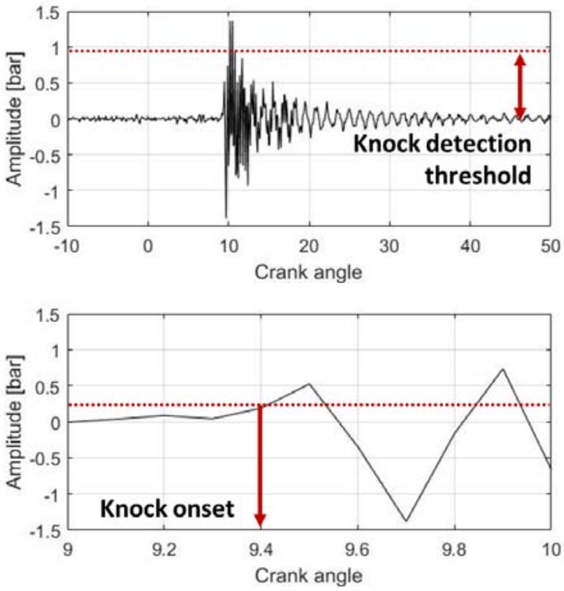


Figure 6: Determination of knock

4. Results and Discussions

4.1 DBL expansion

It is very important to improve the knock resistance for enhancing engine thermal efficiency. Cooling down the enigine helps to increase anti-knock characteristics. In this study, knock mitigation effect was obtained and observed by controlling the temperatures of each component of the cylinder head, liner and the piston. Expansion of the detonation border line by reducing knock occurrence was tried in this study.

In this study, KF refers to knock frequency and represents the percentage of knock-occurring cycles among all the cycles. Since the occurrence of a knock during the operation is obviously irregular, the conditions are set for ±5 % of the specified knock frequency. Each of the cases has reached the ignition timing to be MBT timing to get maximum load condition.

Figure 10 shows the result of changing the temperature of parts, at 1500 rpm condition. As previously explained, the upper surface temperature control of the piston was achieved by the oil cooling gallery and the injection of the oil jet. And the head and liner temperatures were controlled by controlling the coolant temperature.

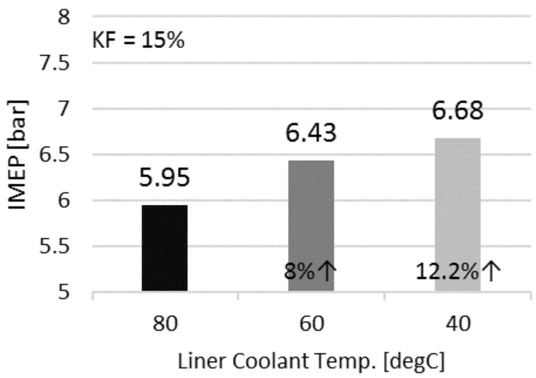
Figure 10(a) shows the result of liner coolant cooling. Head coolant temperature was fixed to 80°C and the liner coolant temperature was varied from 80°C to 60°C and 40°C. Piston cooling wasn't used and the temperature of the piston center spot decreased to approximately 160,154,146.3°C in this case. The temperature of the head remained constant at approximately 100°C. It is shown that liner wall temperature significantly affects piston temperature. As the result, 12.2 % of the load limit expansion was achieved, while maintaining the KF 15 %. The decrease of the temperature at the end of compression stroke, derived from the decrease of wall temperature, led to knock mitigation. The load was increased, thus, the efficiency was maintained, in spite of the temperature decrease of the combustion chamber wall.

The result of reducing the coolant temperature of the head from 80°C to 40°C and maintaining liner coolant at 80°C is shown in Figure 10(b). Compared to Figure 10(a), this tells us that the cooling the head coolant has a bigger effect than the cooling coolant of the liner, on knock mitigation. Head area is bigger than the other parts during the combustion process (over than 53% of total surface area at knock onset angle, 12aTDC CA) and hence could reduce the temperature of the chamber more efficiently. The temperature of the piston was maintained at approximately 160°C in each case and is shown to be more sensitive to changes in the temperature of the liner, rather than the temperature of the head, due to the direct heat conduction through piston ring pack, as mentioned above.

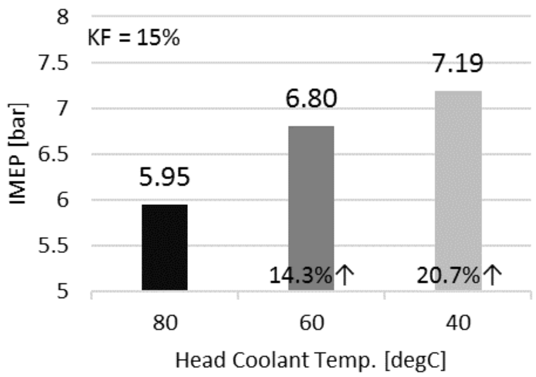
Figure 10(c) is the result of simultaneous cooling of the head and liner. It showed approximately 26.9 % of load expansion was possible. The lower the temperature of the coolant, the higher the temperature difference between the the boundary temperature and the coolant was observed In all cases. It means that as the coolant temperature decreased, heat loss of the combustion chamber increased. However, increase of the load towards the sweet-spot brought the increase of efficiency and measured the same level within the experimental error, eventually.

In case of each KF is 0 and 30% are shown in figure 10(d) and figure 10(e), respectively. In these cases, the temperatures of the head and liner coolant were changed simultaneously. As KF increased, a slight decrease of effects was observed. It is thought that knock occurrence might lead to abnomal thermal distribution in the cylinder and become an obstacle for increase of load.

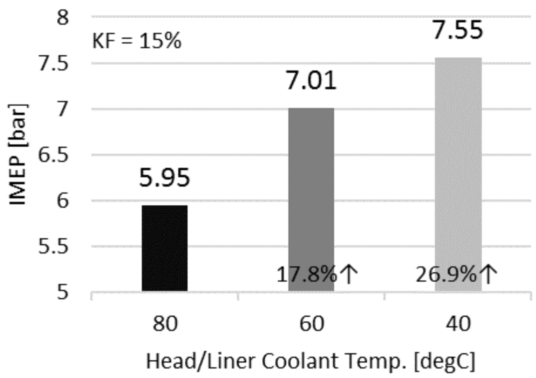
Figure 10(f) shows the case where the coolant temperature of the head and the liner was maintained at 80°C and the piston center temperature was cooled down from 160°C to 132°C, with oil cooling gallery. A load increase of 10.6% was achieved in this case. As previously mentioned, cooled oil injection strategy was tried. Injected oil was cooled down to 20°C from the outside of the engine, but the effect was not big enough due to the heat transfer between block and the oil when the oil passes the passage.



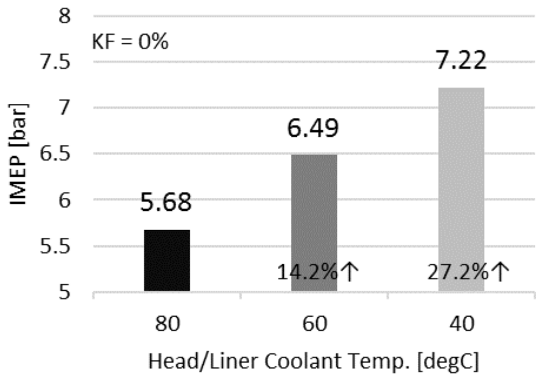
(a) Liner : 80, 60, 40°C
Head : 80°C (fixed)
KF = 15%



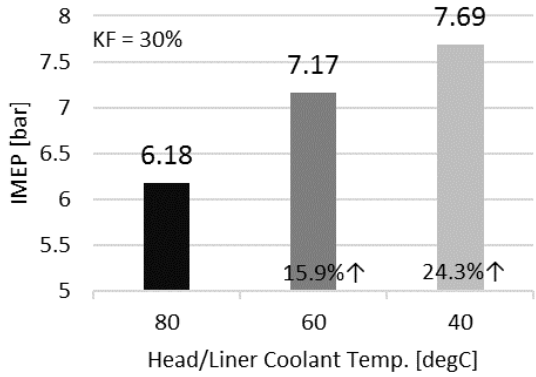
(b) Liner : 80°C (fixed)
Head : 80, 60, 40°C
KF = 15%



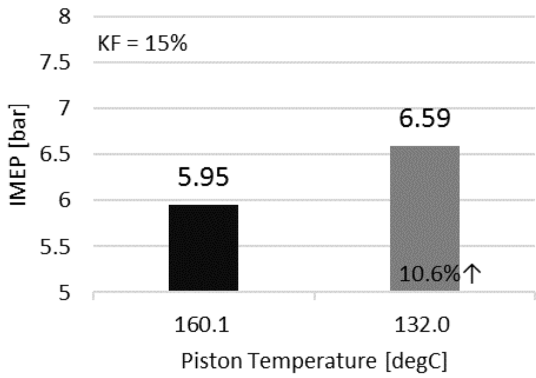
(c) Liner : 80, 60, 40°C
Head : 80, 60, 40°C
KF = 15%



(d) Liner : 80, 60, 40°C
Head : 80, 60, 40°C
KF = 0%



(e) Liner : 80, 60, 40°C
Head : 80, 60, 40°C
KF = 30%



(f) Liner : 80°C (fixed)
Head : 80°C (fixed)
KF = 15%
oil-jet cooling : none, full

Figure 10: DBL expansion result at 1,500rpm

Figure 11 is a brief summary of the cases at 1500 rpm under various KF. Up to 31.44 % of the load increase was achieved with aggressive cooling, including the oil jet. With the increase in load, even though there was a small reduction in combustion efficiency, the indicated efficiency was maintained at the same level. But as it is well known [11, 12, 26], under real vehicle driving and engine operating conditions, excessive cooling leads to a greater loss of efficiency, calling for further parametric study or optimization.

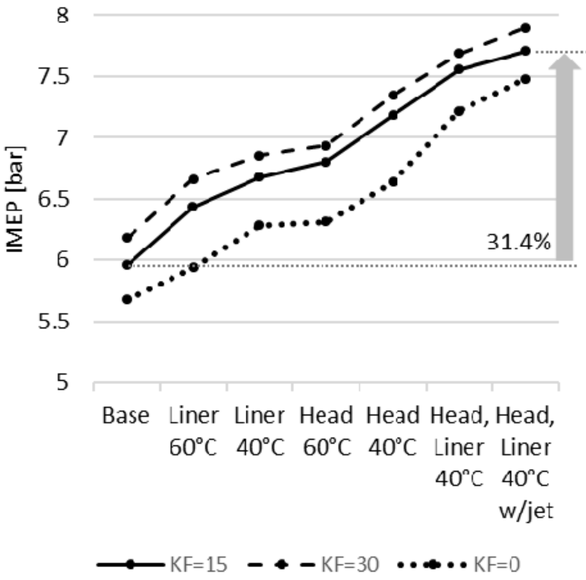


Figure 11: Load expansion summary at 1500rpm

With the same level of load, for example, if the piston is more cooled, the temperature of the other parts (head and liner) were slightly decreased as well. Due to the MPI injection characteristics, it is believed that combustion efficiency is not affected much by the wall temperature decrease. So it can be thought that if the load is the same, the total amount of heat generated from the combustion chamber is also similar.

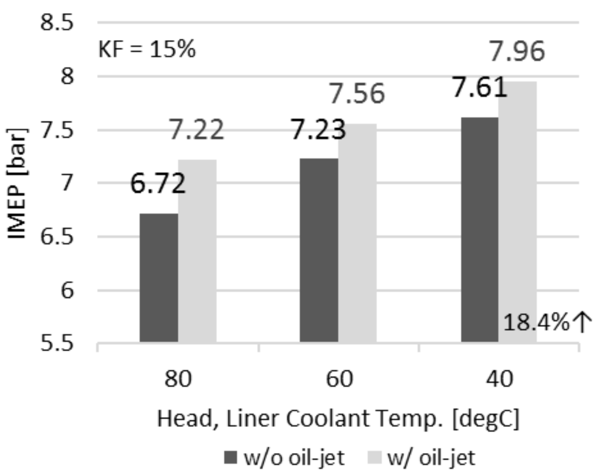


Figure 12: DBL expansion result at 2000rpm

Verification of the various cooling effects was achieved at 2000 rpm as well and the results are illustrated in Figure 12. Accordingly, a maximum load expansion of 18.4% was achieved.

Through implementation under various loads, effects on the knocking of each object could be indirectly estimated, in terms of the thermal boundary condition. For example, while the thermal boundary conditions of the head and the piston were maintained, load increase amount by the liner boundary temperature decrease per unit could be measured. Likewise, the effect of the head and the piston could be indirectly estimated as well. Even though it is not an exact value due to various operating conditions and differences, it seems that the liner boundary temperature decrease has the greatest effect (~0.04 bar of IMEP/°C), two times more than the head and the piston (0.022~0.024 bar of IMEP/°C), in terms of knock phenomena. It is thought that during the intake process, due to the tumble motion, lots of fresh air first hits the liner on the exhaust side and begin heat transfer first in the other parts of the combustion chamber. Furthermore, in the intake process, the liner surface area is the biggest among the three. This can make the air-fuel mixture colderd at IVC timing. D. Dakahashi *et al*. [26] had investigated with 3-D simulation and suggested that strategic cooling on the exhaust side could enhance engine performance. However, further study is necessary to evaluate the effect precisely in the future.

Experiments using the VISIO system could not be carried out, particularly under 40°C coolant temperature condition of the head. Due to the low temperature of the head, there was less vaporization of the fuel from the MPI injectors, leading to excessive PM deposit on the optical probes. However, 40°C condition of the liner coolant showed not much difference on PM emission.

By measuring the perturbation of the light intensity signal derived from the pressure oscillation, under weak knock-occurring condition, it is not easy to capture the location of knock and analyse it statistically. Thus, to make the in-cylinder condition more knock-prone, ignition timing was advanced to bTDC 23CA. In each case, KF showed approximately 40%. The load was increased, while the coolant temperature of the liner was decreased. It showed statistically uniform knock occurrence around the cylinder bore under heavy knock condition regardless of the location as increase the load.

Figure14 shows the gasket ion probe result. The knock locations were found under heavy knock condition, while the head coolant temperature was decreased. It shows one example for each condition of 80°C and 40°C liner coolant temperature. The line with dashes indicates the knock onset angle and the solid line indicates the angle at which the flame reached after the ignition timing. A total 16 directions, 22.5° for each, were measured and were splined, as indicated in the figure. The radius means the angle in the polar graph; the bigger radius indicates that the flame reached late. The coloured area shows possible knock location, where the flame reached later than knock onset. It is shown that knock occurs simultaneously around the cylinder bore. Even though the condition of 40°C liner coolant temperature also showed simultaneous occurrence of knock around the cylinder bore, the intake side shows slightly more knock occurrence compared to the exhaust side for whole operating cycles, statistically. It was also found in optical result, knock occurrence was more simultaneous around the cylinder under higher liner temperature condition compared to low temperature condition, statistically.

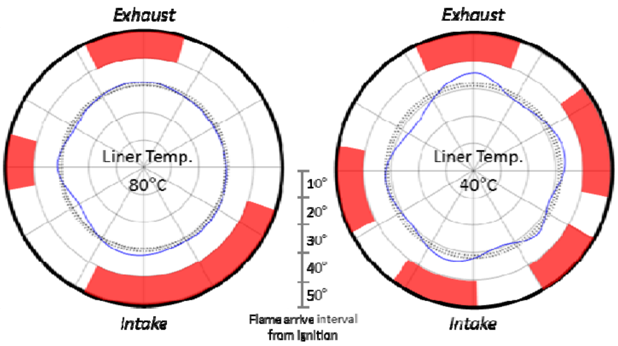


Figure 14: Flame propagation and knock spot under heavy-knock condition

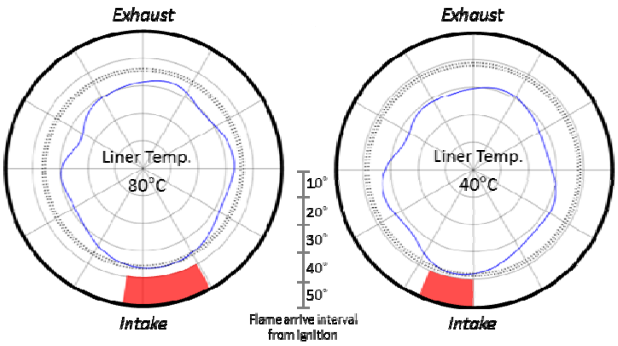


Figure 15: Flame propagation and knock spot under weak-knock condition

Figure 15 shows the result under weak knock condition. As the KF was decreased, load was decreased as well. At 80°C of liner coolant temperature condition the load was 6.16 bar and was 6.5bar at 40°C condition. Ignition timing was maintained at 21 bTDC CA in order to achieve MBT. It can be thought that if the time of latest flame arrival exceed or is consistent with the knock onset angle, knock occurred in that direction. Average results of 300 total cycles showed different flame speed for each direction. Significant difference was found in the flame speeds between the exhaust and the intake side. In all cases, under weak knock condition which is relatively low load condition, flame speed to the exhaust side was faster than the intake side regardless of the liner coolant temperature. It can be thought that the knock occurred on the intake side majorly under weak knock condition.

4.2 Ignition timing advance

The temperature variation effect of the cooling water was studied to determine its influence on the advance of injection timing. In all cases, KF was maintained at 15%, and load was maintained as 7.15 bar of IMEP at 1500rpm and 7.5 bar of IMEP at 2000rpm. Since the load was constant, combustion phase was advanced as the ignition timing advanced toward MBT. Adjusting the oil flow rate faciliated the control of piston temperature. Upon adjustment of the oil flow rate, piston central temperature showed approximately 160°C at zero flow rate(w/o jet), 145°C at medium state(half jet) and 130°C at maximum flow rate(full jet), respectively. In this study, excessive cooling condition was excluded due to its unreality, as a lower-than-40°C coolant temperature may cause additional loss of the vehicle.

Figure 16 shows the result of ignition advance and CA50 during cooling down the head coolant temperature, while liner coolant temperature was maintained 80°C. As the head coolant temperature decreases by 1°C, approximately 0.125CA advance of ignition timing was obtained, regardless of the piston temperature. Ignition timing advance led to the increase of overall efficiency, even though the boundary wall temperature was lower.

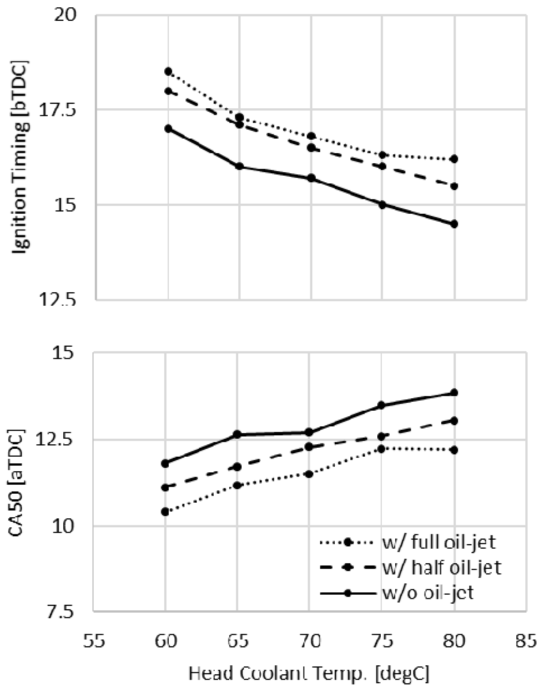


Figure 16: Ignition timing advance at 1500rpm

Figure 17 shows the result of simultaneous changes of coolant temperature of the head and the liner. It is indicated with solid line that the case without changing the liner coolant temperature. Additional knock reductions could result in further ignition timing advance, when the liner was cooled simultaneously. For each case, 0.125, 0.175, 0.2, 0.24CA of ignition timing advance was achieved per 1°C of coolant temperature decrease. The absolute slope of the line is increasing, while the piston temperature decreases. The effect of piston cooling through the oil jet increased when the flow rate of oil jet was increased. This is considered to be a more effective reduction in heat transfer to air-fuel mixture during the intake process, due to the simultaneous cooling of the piston and the liner. Lowering piston temperature with maximum flow rate and cooling down both head and liner to 60°C facilitated the MBT timing of ignition.

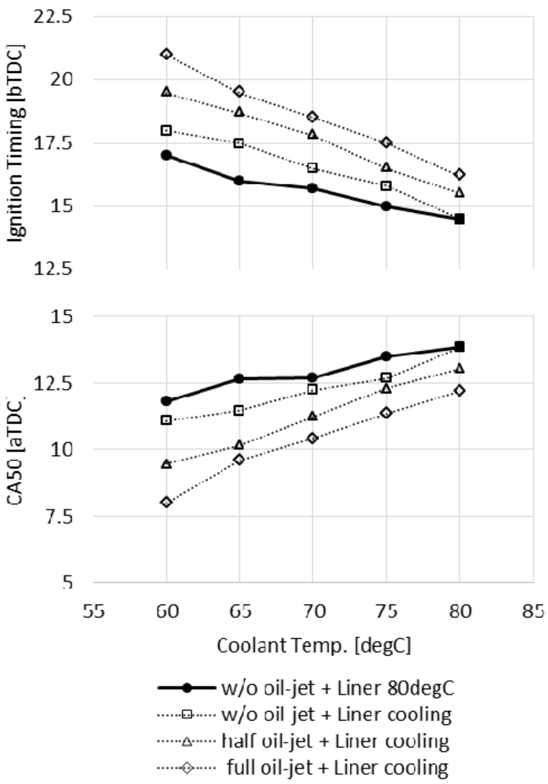


Figure 17: Ignition timing advance at 1500rpm

As shown in Figure 18, the effect at 2000rpm was also investigated. Approximately 0.2CA ignition timing advance per 1°C decrease of coolant temperature was achieved. MBT was achieved with low coolant temperature and piston temperature decrease. However, no further mitigation effect of simultaneous cooling of the piston and the liner was observed.

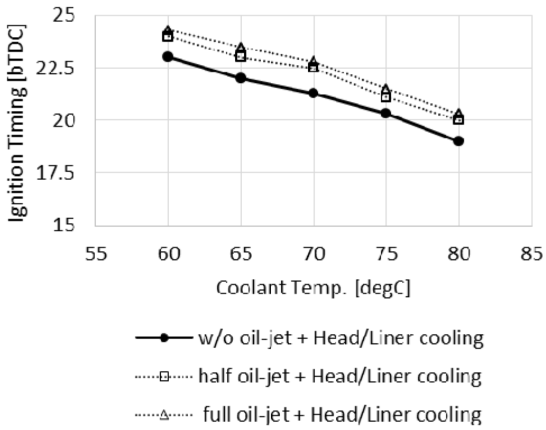


Figure 18: Ignition timing advance at 2000rpm

5. Conclusion

In this study, the effect of thermal boundary condition of combustion chamber on knock characteristics by controlling the head and liner coolant temperatures and controlling oil injection to piston gallery was investigated.
First, with respect to DBL expansion, decrease of boundary temperature showed knock mitigation, which led to load increase. 31.6% and 18.4% of the load increase was achieved at 1500 and 2000rpm respectively while decreasing the coolant temperatures of the head and the liner, also injecting the engine oil to the piston gallery.
To capture the knock location, the visualization and flame ionization detection method were used. Under weak-knock condition, knock spot was seemed mostly locates on the intake side in all cases. But as increasing the load, under heavy-knock condition, it was noticed that knock occurred simultaneously around the cylinder bore.
Second, ignition timing advance was measured while maintaining the load as the boundary condition changed. As a result, low boundary temperature condition showed more ignition advance towards MBT timing. 0.24 and 0.2CA of ignition timing advance per 1°C coolant temperature decrease was achieved at 1500 and 2000 rpm, while cooling down the coolant of the head and the liner to 60°C and using the piston gallery cooling. As the ignition timing was advanced, efficiency increased.
However, excessive cooling such as 40°C of coolant temperature condition of each part can make additional efficiency loss. On this point, especially for optimization, further study is imperative in the future.

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8. Glossary

aTDC: After Top Dead Center

bTDC: before Top Dead Center

aBDC: after Bottom Dead Center

bBDC: befor Bottom Dead Center

bsfc: brake specific fuel consumption

CA: Crank Angle

CR: Compression Ratio

DBL: Detonation Border Line

EGR: Exhaust Gas Recirculation

HPF : High-pass Filter

KF: Knock Frequency

MBT: Maximum Brake Torque

N/A: Naturally Aspirated