Effect of cooling water flow direction on performance of an acetylene fuelled HCCI engine

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This paper deals with experimental investigations carried out on an acetylene-fuelled engine operated in homogeneous charge compression ignition (HCCI) mode to study effect of cooling water flow direction on intake charge temperature and heating requirements, and performance of the engine. In this study, in a possible load range, experiments are conducted on a single-cylinder, water-cooled, acetylene fuelled HCCI mode with conventional and reverse cooling water flow directions using different intake charge temperatures at different loads. At every load condition, best possible intake charge temperature is determined based on brake thermal efficiency. Results obtained in two cases of cooling water flow directions of HCCI mode are compared with those of conventional diesel CI mode. From the results, it is found that, reverse cooling water flow direction shows about 14 to 50% reduction in external intake charge heating at different load conditions as compared to conventional cooling water flow direction. There is an improvement in brake thermal efficiency by about 5 to 10% at different load conditions. Nitric oxide (NO) and smoke emissions are very low in both cases. However, hydrocarbon (HC) and carbon monoxide (CO) levels are higher than conventional CI mode in both the cases.

Keywords: HCCI, Acetylene, Reversed cooling water flow

Conventional compression ignition (CI) engines show higher brake thermal efficiency, but they emit high levels of nitric oxides (NOx) and smoke emissions. Conventional spark ignition (SI) engines show comparatively lower brake thermal efficiency with high NOx, HC and CO emissions. Today, HCCI is an alternative combustion system as it can reduce both NOx and smoke simultaneously to very low levels with good brake thermal efficiency. Onishi et al. first introduced controlled autoignition combustion concept in a two-stroke engine in order to reduce instability at part-load operations and they achieved reduction of emissions and fuel consumption. Later, Najt and Foster extended the HCCI combustion into a four-stroke engine using combinations of high and low octane fuels. Thring investigated on a four-stroke gasoline HCCI engine to find important parameters required for successful operation at part-loads. Ryan and Callahan used a port fuel injection (PFI) injector to supply diesel into an intake air stream with high intake air temperature and compression ratio to achieve the HCCI combustion. This resulted in very early heat release and they concluded that low compression ratio was most suitable for PFI diesel HCCI engine. Swami Nathan et al. tried diesel HCCI with manifold injection. They found smoke emissions were higher than those of conventional CI engine due to low volatility of diesel and also high-load condition was limited by early heat release rate. Intake charge heating, variable compression ratio (VCR), variable valve timing (VVT) and use of secondary low octane fuels as an ignition improver methods tried in the past for achieving the HCCI.

Researchers have tried various alternative fuels also with high autoignition temperatures for HCCI combustion, viz., natural gas, methanol, and ethanol, liquid petroleum fuel (LPG). Christensen et al. successfully investigated on HCCI combustion using natural gas with an intake air heating. Christensen et al. used different combination of high and low octane fuels with VCR keeping air-fuel ratio constant. They investigated that any liquid fuel can be used for the HCCI combustion with VCR. Shi et al. achieved direct in-cylinder injection diesel (at end of exhaust stroke) HCCI with VVT. Their results showed that combustion stability at lower and higher load conditions were worst due to misfiring and knocking. Chen and Konno introduced dual-fuel HCCI

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The combustion of natural gas with dimethyl ether (DME). They found that by optimizing proportions of DME and natural gas, NOx emissions could be lowered to near zero levels. Dual-fuel operation gave higher thermal efficiency than that of the CI mode operation.

Swami Nathan et al. adopted acetylene as a fuel for HCCI engine because of its moderate autoignition temperature and high flammability limits. They varied intake charge temperature to control combustion phasing. They achieved brake thermal efficiency comparable to that of conventional CI mode by using proper intake charge temperature.

Traditionally, acetylene is produced from calcium carbide. Calcium carbide reacts with water producing acetylene and calcium hydroxide. Principal raw materials for calcium carbide production are limestone and coke. Acetylene can also be produced from thermal cracking of methane. This shows that acetylene can be renewable in nature. Gaseous nature of acetylene helps easy mixing with air and eliminates wall wetting problems in internal combustion (IC) engines. Autoignition temperature of acetylene is higher than that of diesel. Thereby, standard compression ratio of conventional CI engine itself can be used. The properties of acetylene are given in Table 1.

From previous investigations, it seems that there is a good scope for acetylene fuelled HCCI engine. Therefore, this study focuses on reduction of intake charge heating by external means in acetylene fuelled HCCI engine.

### Experimental Study

In this study, a single-cylinder, water-cooled, CI engine is operated in HCCI mode with acetylene as fuel. An external electrical heater is used in order to heat intake charge to achieve HCCI combustion if required. Engine speed and cooling water outlet temperature are kept constant. The engine is also operated in conventional CI mode at different loads. Influence of coolant-water-flow direction on intake charge heating, engine performance and emission characteristics at different loads in HCCI mode are studied. In this study, coolant water flow direction in HCCI mode is changed from bottom-up to top-down (Fig. 1) flow. This is done in order to reduce heat transfer from cylinder liner to cooling water thereby higher cylinder liner temperature will be maintained.

In conventional water cooled engines, cooling water flows full through cooling passages from bottom of cylinder liner towards cylinder head and maintain cylinder liner temperature at required level. When cooling water flow direction is reversed, i.e., it flows from cylinder head towards cylinder liner, cylinder liner temperature will be at higher level. When cooling water first enters high temperature cylinder head, it absorbs most of heat energy. Later when it passes through cylinder liner, amount of heat energy lost to cooling water reduces because it is comparatively at higher temperature. Therefore, cylinder liner temperature will be maintained at higher level than the conventional cooling water flow direction. This helps reduce amount of intake charge heating required in acetylene HCCI mode.

### Experimental set-up

A single-cylinder, water-cooled, CI engine is used to run both in acetylene HCCI and conventional CI modes. The engine specifications are given in Table 2. The engine is coupled to an eddy current dynamometer for loading and measurement purpose.

![Schematic of conventional and reverse cooling water flow directions](image-url)
Other end of engine shaft is connected to an induction motor through a magnetic clutch for starting purpose. The engine is also fitted with an electrical intake charge heating system, which is developed in-house and used to control intake charge temperature. The acetylene is inducted into intake manifold after the electrical heater under slightly higher than intake manifold pressure. Provisions are made to measure flow rates of fuel, air and cooling water, and to measure temperatures of intake air and intake charge, cooling water outlet and cylinder liner. Exhaust gas analyzers working on FID (flame ionization detector) for HC, NDIR (non-dispersive infrared) for CO and CLD (Chemiluminescence detector) for NO are used. A Bosch smoke meter is used for measurement of smoke emissions. Fig. 2 shows schematic of experimental set-up developed and used in this study.

**Experimental procedure**

First, experiments are conducted in a conventional diesel CI mode from zero to full load of operation. In this case, static fuel injection timing is maintained at 21 crank angle degree (CAD) before top dead center (TDC). Nozzle opening pressure (NOP) of 220 bar and cooling water outlet temperature of 35°C are maintained.

Second, the engine is run with conventional cooling water flow direction (bottom-up flow) in acetylene HCCI mode. At entire HCCI mode of experiments, cooling water outlet temperature and engine speed are kept constant at 50°C and 1500 rev/min. Load on the engine is varied by varying quantity of acetylene supplied to it. Overall load range of the engine is set by misfiring and knocking limits. At every intake charge temperature, the engine is operated from

![Fig. 2—Schematic of experimental setup](image-url)
misfiring to knocking limit to determine the operating load range. In this study, rate of pressure rise more than 10 bar/CAD is considered as knocking. At every load, intake charge temperature is varied in range of 40-110°C depending upon load which is again set by the misfiring and knocking limits. By this way, it is possible to operate the engine in HCCI mode from 0.5 to 3 bar BMEP under different intake charge temperatures. Then, at every load, best intake charge temperature is determined based on brake thermal efficiency. Next, experimental procedure as explained above in HCCI mode is repeated with reverse cooling water flow direction (top-down flow) keeping all the other parameters same. Also, in this study, external heat addition ($H$) for a particular configuration of the engine is calculated as follows:

$$ H = M_a * C_p * (T_h - T_a) \text{ (W)} \quad \ldots(1) $$

Where, $M_a$ is mass flow rate of air, $C_p$ is specific heat of air at constant pressure, $T_a$ is atmospheric temperature and $T_h$ is temperature of air after the electrical heater. Percentage variation of external charge heating is calculated from amounts of heat addition in actual and reverse cooling water flow directions.

Results and Discussion
In following figures, CD and RD refer to acetylene HCCI mode with conventional and reverse cooling water flow directions.

Effect of cooling water flow direction on brake thermal efficiency, heat release rate and intake charge heating

Figure 3 shows variation of brake thermal efficiency (BTE) with BMEPs for conventional CI mode, and HCCI with conventional and reverse cooling water flow directions. In Fig. 3, BTE values of HCCI engine are corresponding to best possible ones at a given load (at best intake charge temperatures). From Fig. 3, it is seen that BTE in acetylene HCCI mode, for entire load range, is comparable with those of conventional CI mode. In addition, acetylene HCCI operation with reverse cooling water flow direction shows higher BTE than that of conventional one. It may be due to advanced combustion phasing in case of reverse cooling water flow direction. This can be seen from Fig. 4. In Fig. 4, at 2.5 bar BMEP, with conventional flow direction, peak heat-release occurs at about 364 CAD, whereas for reverse flow direction, it occurs at about 358 CAD.

With reverse flow direction, combustion phasing occurs closer to TDC, whereas in conventional case, it occurs far away after compression TDC, thereby reducing net work output. It may be attributed to higher cylinder liner temperatures (Fig. 8) with reverse flow direction in which heat transfers into cylinder gases. Also, peak heat-release in former case is about 80 J/CAD, whereas in later case, it is about 150 J/CAD. Similar behaviour is observed at other BMEPs with reverse flow direction. These factors may be responsible for higher BTE in case of reverse flow direction. There is an improvement in BTE by about 5 to 10% with reverse flow direction as compared to that of conventional one.

Figure 4 shows heat release rates with CADs for both directions of cooling water flow. In Fig. 4, heat-release-rate curves shown are for best BTE conditions (best intake charge temperatures). From Fig. 4, it is seen that, start of combustion (SOC) with reverse flow direction is advanced than that of conventional flow direction. Thereby, it advances occurrence of peak-heat-release point also. This is attributed to higher
heat transfer occurring to cylinder gases due to higher cylinder liner temperatures with reverse flow direction, whereas in conventional flow direction, low temperature of cylinder gases due to low cylinder liner temperature retards the SOC and thereby reducing BTE. In addition, it also helps reduces amount of intake charge heating required by external means. From Fig. 4, it is also seen that, intake charge temperature of 50°C is required at 3 bar BMEP for HCCI operation with conventional flow direction, whereas with reverse flow direction, only 40°C is enough.

It is observed that, when cooling water flow direction is reversed, at all BMEPs, level of intake charge temperature required reduces. This is again due to higher heat transfer to cylinder gases due to higher cylinder liner temperature as discussed earlier. Reduction in levels of intake charge temperatures at different BMEPs are shown in Table 3. Also, reduction in electrical heating requirements at different BMEPs is shown. Reduction in electrical heating requirements is more at higher BMEPs than at lower ones. This is because at higher BMEPs, intake charge temperatures are less due to higher combustion temperatures. At 3 bar BMEP, reduction in electrical heating as high as 50% is possible. This saves lot of external energy supply.

### Table 3—Reduction in inlet charge heating with reverse cooling water flow direction

<table>
<thead>
<tr>
<th>Load (bar)</th>
<th>Inlet charge temperature (conventional cooling) (°C)</th>
<th>Inlet charge temperature (reversed cooling) (°C)</th>
<th>Reduction in inlet charge heating (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.5</td>
<td>110</td>
<td>100</td>
<td>14</td>
</tr>
<tr>
<td>1.0</td>
<td>100</td>
<td>90</td>
<td>16</td>
</tr>
<tr>
<td>1.5</td>
<td>80</td>
<td>70</td>
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</tr>
<tr>
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<td>70</td>
<td>60</td>
<td>26</td>
</tr>
<tr>
<td>2.5</td>
<td>60</td>
<td>50</td>
<td>33</td>
</tr>
<tr>
<td>3.0</td>
<td>50</td>
<td>40</td>
<td>50</td>
</tr>
</tbody>
</table>

**Effect of cooling water flow direction on emissions**

**NO and smoke emissions**

Figure 5 shows variation of NO emission with BMEP in CI and HCCI modes with conventional and reverse cooling water flow directions. In entire load range, NO emissions are very much lower in HCCI mode of operation with both cooling water flow directions as compared to conventional CI mode. This may be mainly due to lower combustion temperature with HCCI mode. Burning of mixture takes place instantaneously in HCCI mode, therefore it avoids high temperature regions during combustion compared to that of conventional CI mode. Also, compression effects on burned gases by burning gas is also absent in HCCI mode, thereby it avoids localized high temperature regions during combustion. In addition, overall combustion temperature is less due to lean mixture. All these factors reduce NO emissions. NO levels in reverse flow direction are marginally higher in some cases. It may be due to advanced combustion phasing with high heat release rates in those cases. However, the NO levels for entire load range are much lower compared to conventional CI mode. From Fig. 5, it is observed that, NO emissions are higher at 2.5 bar BMEP with reverse flow direction. This may be due to higher intake charge temperature used at this BMEP alone than actually required one because intake charge temperature is varied only in a step of 10°C.

Smoke emissions in both cases of cooling water flow directions are very much lower compared to conventional CI mode.

**Hydrocarbon emissions**

Figure 6 shows variation of HC emissions with BMEPs. Generally, acetylene HCCI mode exhibits more HC emissions at all loads compared to
conventional CI mode because of pre-mixed charge. It may be also due to lower combustion and exhaust gas temperatures in HCCI mode which inhibit oxidation of HC in exhaust system. In HCCI modes, HC emissions are comparatively lower at higher loads than lower loads. This may be attributed to fact that, at higher loads, combustion and exhaust gas temperatures are higher which assist post-combustion oxidation of HC resulting in reduced HC emissions.

The HC emissions in HCCI mode with reverse flow direction are higher than that of conventional flow direction except for 2.5 bar BMEP. This may be due to comparatively higher density of intake charge because of reduced intake charge temperatures in case of reverse flow direction. With reverse flow direction, external charge heating is less because of higher wall temperatures involved. At 2.5 bar BMEP, since intake charge temperature is higher than required one as explained above, it may cause more oxidation and reduce HC emissions at 2.5 bar BMEP.

**Carbon monoxide emissions**

Carbon monoxide (CO) emission is generally an indication of incomplete oxidation of fuel. The acetylene HCCI mode exhibits more CO emission compared to that of conventional CI mode due to prevailing low combustion temperatures for entire load range. It is known that, gaseous fuelled HCCI engine emits lower levels of CO compared to liquid fuel operation due to proper mixing of gaseous fuel with the air. Figure 7 shows variation of CO emissions with BMEPs.

At low load conditions, CO emission is comparatively higher due to lower combustion temperatures. Whereas at higher loads, CO emission is comparatively lower because of higher combustion temperatures. It is observed from Fig. 7 that, CO emissions in case of reverse flow direction are higher than those of conventional flow case except for 2.5 bar BMEP. Reason for this is similar to that for HC emissions. With reverse flow direction, with reduced external heating of charge, intake charge density increases. Therefore, when high-density charge present in crevices and quench layer enters into bulk gases during expansion stroke, it may burn partially leading to higher CO emissions.

**Effect of cooling water flow direction on cylinder liner temperature and cooling water flow rate**

Figures 8 and 9 show effect of cooling water flow directions on cylinder liner temperature and cooling water flow rate. From Fig. 8, it is seen that cylinder liner temperatures are higher for reverse flow direction for the entire load range. It may be due to the higher heat transfer to cooling water from cylinder head because cooling water first enters cylinder head and then to cylinder liner in reverse flow direction.
Even though, cylinder liner temperatures increase with reverse flow of cooling water, however they are within tolerable limits of cylinder material, which is cast iron in this case\textsuperscript{11}.

At lower loads, higher intake charge temperatures cause comparatively higher cylinder liner temperatures. At higher loads, they increase again due to higher combustion temperatures. From Fig. 9, it is seen that cooling water flow rate is lower for reverse flow direction indicating reduction of heat losses to the cooling water. In reverse flow direction, it is seen that, the cylinder liner temperatures are higher. This is attributed to reduced heat transfer from the cylinder liner to the cooling water. In reverse flow direction, the cooling water enters the high temperature cylinder head first and heated to higher temperature as explained earlier. Therefore, heat transfer from the cylinder liner reduces due to reduced temperature gradient. Since cooling water outlet temperature is maintained constant (50°C), mass flow rate of the cooling water is required to be reduced as depicted in Fig. 9.

**Conclusions**

In this study, experimental investigations are carried out on conventional CI and acetylene fuelled HCCI modes with conventional and reverse cooling water flow directions to evaluate performance and emission characteristics. From analysis of results, following conclusions are drawn:

(i) Reverse-cooling-water flow direction results in reduction of about 14 to 50% in external intake charge heating at different load conditions.

(ii) Reverse flow direction shows an improvement in brake thermal efficiency of the acetylene fuelled HCCI mode by about 5 to 10% compared to conventional flow direction.

(iii) Brake thermal efficiency of the acetylene fuelled HCCI mode is comparable to that of conventional CI mode for the entire load range.

(iv) In both cooling water flow directions, acetylene fuelled HCCI mode, NO and smoke emissions reduce almost to zero levels for entire load range compared to conventional CI mode.

(v) HC and CO emissions are higher in acetylene HCCI mode compared to conventional CI mode. Reverse flow direction shows higher levels of HC and CO emissions compared to conventional flow direction.

**References**