ANALYSIS OF AN I.C. ENGINE USING WATER COOLING TECHNIQUE

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ABSTRACT

The current scenario is focused on the cooling of an IC engine when it runs at high speed for a long time. The many parameters are affected there would not be proper cooling of the engine. The one of the most important parameter is the efficiency and power of the engine. In the present paper, a non-intrusive thermometry method using full spectral analysis of cooling of an IC engine is developed for the analysis the performance of two and four stroke engine. The investigation is performed as a proof of concept, focusing on reliability and accuracy of the method. The performance is based on the various input parameters like compression ratio, Maximum possible temperature in the cylinder and speed of engine. The results have been drawn for two stroke and four stroke diesel engine by developing software in C++ and followed by the graphs plotted in Origin 6.1. The results have been shown Indicated power, Mechanical Efficiency, Mass of cooling required and thermal efficiency.

1. INTRODUCTION

1.1 Petrol Engines

In petrol engines the air-fuel ratio (AFR) is maintained at an approximately constant value of 14-16:1 by the carburetor or fuel injection system. The top temperature (T3) and the torque is determined by the amount of air-fuel mixture admitted by the throttle. Hence petrol engines are described as being quantity governed.
efficiency can be improved. Research results show that the recovery efficiency of exhaust gas energy is mainly limited by exhaust gas temperature. The maximum bottom cycle power can reach 19.2 kW and IC engine thermal efficiency can be improved by 6.3% at 6000 r/min. All those can prove this novel bottom cycle.

1.2 Diesel Engines

In diesel engines varying amounts of fuel, in the form of very fine droplets, are injected into approximately the same amount of air, irrespective of the engine’s speed, to control the top temperature and the torque. The AFR therefore varies (typically 20-100:1), hence Diesel engines are described as being quality governed. Fuel burns (after a slight delay) on injection. The ideal (or 'true') diesel cycle is shown below in which the process 2-3 is constant pressure heat transfer to the cycle.

The bottom cycle concept is designed on a four-cylinder naturally aspirated IC engine: with three cylinders taken as ignition cylinder, the last one is used for steam expansion cylinder; IC engine exhaust pipe is coupled with a Rankine steam cycle system which uses the high temperature exhaust gas to generate steam; then, the steam is injected into steam expansion cylinder and expands in the cylinder. In this way, the Otto cycle (or diesel cycle) of traditional IC engine and the steam expansion cycle (open Rankine cycle) are coupled on IC engine. On this basis, the energy recovery potential of this bottom cycle is studied by cycle processes calculation and parameters analysis. The research results show that the recovery efficiency of exhaust gas energy is mainly limited by exhaust gas temperature. The maximum bottom cycle power can reach 19.2 kW and IC engine thermal efficiency can be improved by 6.3% at 6000 r/min. All those can prove this novel bottom cycle.
concept has larger potential for energy saving and emission reduction on IC engine. Jianqin Fu, et al. [20] have proposed an open steam power cycle used for IC engine exhaust gas energy recovery in order to improve IC engine energy utilization efficiency. P. Raman and N.K. Ram [31], have studied the performance of an internal combustion engine fueled with 100% producer gas was studied at variable load conditions. The engine was coupled with a 75 kWe power generator. Producer gas generated from a downdraft gasifier system was supplied to the engine. V. Pandiyarajan, et al. [21] have evaluated and reported the performance parameters pertaining to the heat exchanger and the storage tank such as amount of heat recovered, heat lost, charging rate, charging efficiency and percentage energy saved. The exhaust gas from an internal combustion engine carries away about 30% of the heat of combustion. The energy available in the exit stream of many energy conversion devices goes as waste, if not utilized properly. Zbyszko Kazimierski and Jerzy Wojewoda [23] have presented an old idea of how to increase the internal combustion (IC) piston engine. Such an engine has a 2-side action piston, which divides a single cylinder into two working parts. In these parts there are two usual 4-stroke engines which constitute one double IC engine, equipped with a slider-crank mechanism. The new engine may use bio-fuels as well. D. Descieux and M. Feidt [24] have proposed a generic methodology to simulate the static response of an IC engine and show the influence of several engine parameters on the power and efficiency. Moreover it puts in evidence the existence of two optimal engine speeds one relative to maximum power and the second to maximum efficiency. The thermodynamic performance of an air standard diesel engine with heat transfer and friction term losses is analyzed. Z.G. Sun et al. [25] have presented Energetic efficiency evaluation of the combined system of cold and heat, where primary energy rate (PER) and comparative primary energy saving are used. A combined system intended to be an alternative to provide cold and heat for buildings was set up. Comparison of energetic efficiency of the combined prototype system with contemporary conventional separate systems of cold and heat shows that energy utilization efficient is improved greatly, and there is a large potential for energy saving by the combined system of cold and heat. G. De Nicolao et al. [27] have examined the benefits and limitations of black-box approaches and compared. The problem considered here can be viewed as a realistic benchmark for different estimation techniques. The volumetric efficiency ($\eta_v$) represents a measure of the effectiveness of an air pumping system, and is one of the most commonly used parameters in the characterization and control of four-stroke internal combustion engines.

2. ENGINE COOLING SYSTEM

2.1 Necessity for cooling

In an internal combustion engine, the fuel is burned within the engine cylinder. During combustion, high temperatures are reached within the cylinder, for example in a compression ignition engine as high as 2000-25000C is reached. About one third of the heat energy liberated b the burning fuel is converted into power. Another one third goes out through the exhaust pipe unused. The remaining one third flows into the various components of the engine, namely, cylinder, cylinder head, valves, spark plug (in SI engines), fuel injector (in CI engines) and pistons. This heat flow takes place during combustion and expansion processes.

2.2 Optimum cooling

To avoid overheating, and the consequent ill effects mentioned above, the heat transferred to an engine component (after a certain level) must be removed as quickly as possible and be conveyed to the atmosphere. It will be proper to say the cooling system as a temperature regulation system. It should be remembered that abstraction of heat from the working medium by way of cooling the engine components is a direct thermodynamic loss.
2.3 Thermosyphon cooling system

When a vessel of cold water is heated, the hot water will tend to rise (by virtue of its lower density) and its place will be taken over by cold water. This causes a definite circulation within the water mass from top to bottom and vice versa. This is called natural convection.

The thermosyphon circulation cooling system is shown in fig. 4 works on this principle. In the thermo-syphon circulation cooling system, when the engine is cold the whole water is at the same temperature and is at rest. When the engine is operating, the water around the cylinder heads and cylinder walls gets heated and flow up to the top header tank of the radiator. Now the cold water from the lower part of the radiator flows and fills the coolant spaces in the cylinder head and the cylinder block. The hot water flows downward through the radiator tubes. This water is cooled by the stream that flows past the tubes. Air is sucked by the fan which is driven by the engine crankshaft. In this system, the water circulation through the cylinder, cylinder head and the radiator is by natural means. For success in operation, the passages through the jackets and radiator should be free and the connecting pipes large. Further the jackets should be placed as low as possible relatively to the radiator, in order that the hot leg shall have as great a height as possible. During operation the water level must on no account be allowed to fall below the level of the delivery pipe to the radiator top. If this happens, water circulation will cease. In general the thermo-syphon system requires a larger radiator and carries a greater body of water than the pump circulation system. Further, a somewhat excessive temperature difference is necessary to produce the requisite circulation. On the other hand, to some extent it automatically prevents the engine being run too cold. The thermo-syphon circulation cooling system is simpler in construction and operation. This system is used in some motor cars.
2.4 Heat Transfer

The methods adopted for the calculation of quantity of water or air required for the cooling of the rated engine will be calculated on the basis of the basic equations used ion heat transfer. Conduction, Convection and Radiation are three modes of heat transfer in this case also.

\[ Q = h_a A (T_1 - T_2) \]

\[ Q = \text{Quantity of convective heat transfer} \]
\[ h_a = \text{Coefficient of convective heat transfer} \]
\[ A = \text{Area of surface} \]
\[ (T_1 - T_2) = \text{temperature difference between the fluid and the surface} \]

Coefficient of heat transfer \( h_a \) is defined as the amount of heat transmitted for a unit temperature difference between the fluid and the unit area of the surface in unit time. The value of \( h_a \) depends on the types of fluid, their velocities and temperatures, dimension of the pipe and the types of problem. Since \( h_a \) depends upon several factors, it is difficult to frame a single equation to satisfy all the variation, however a dimensional analysis gives an equation for the purpose which is given under:

\[ \frac{h_a D}{k} = C \left( \frac{\nu D}{\mu} \right)^a \left( \frac{c_p \mu}{k} \right)^b \left( \frac{D}{L} \right)^c \]

\[ N_u = Z(R_e)^a (P_r)^b \left( \frac{D}{L} \right)^c \]

\[ N_u = \text{Nusselt number} \frac{h_a D}{k} \]
\[ R_e = \text{Reynolds number} \left( \frac{\nu D}{\mu} \right) \]
\[ P_r = \text{Prandtle number} \left( \frac{c_p \mu}{k} \right) \]
\[ \frac{D}{L} = \text{Diameter to length ratio} \]
\[ C = \text{a constant determined experimentally} \]
\[ c_p = \text{Specific heat at constant pressure} \]
\[ K = \text{thermal conductivity} \]
\[ \rho = \text{Density} \]
\[ \mu = \text{Dynamic viscosity} \]
\[ V = \text{velocity} \]
The overall heat transfer coefficient (U) is given by:

\[ U = \frac{1}{\frac{1}{h_a} + \frac{1}{k} + \frac{1}{h_b}} \]  
(5)

Cooling water requirement:

The heat rejected to the coolant greatly depends upon the type of the engine. It can be seen for a small high speed engine the heat rejected of the coolant can be as high as 1.3 times the B.P. developed, while for an open chamber engine it is only about 60% of the B.P. developed.

The quantity of water required for cooling is given by

\[ Q_w = \frac{Z \times B.P.}{\Delta t_w} \]  
(6)

\( \Delta t_w \) = permissible increase of temperature of cooling water

Z is the constant which depends upon the fuel consumption and the compression ratio

Area \( (A) = \frac{\pi}{4}D^2 \)  
(7)

k=0.5 for 4 stroke engine

k=1 for 2 stroke engine

\[ C_p = (1.0189 \times 10^3) - 0.13784 \times T_a + (1.9843 \times 10^{-4}) \times T_a^2 + 4.2399 \times 10^{-7} \times T_a^3 - (3.7632 \times 10^{-10} \times T_a^4) \]  
(8)

\[ T_x = \frac{T_3}{C_R \times T_2} \]

\[ T_4 = T_3 \times (T_x)^{\gamma - 1} \]

For Diesel cycle:

\[ \eta = 1 - \frac{1}{\gamma} \left[ \frac{(T_4 - T_1)}{(T_3 - T_2)} \right] \]  
(9)

\[ Q_L = c_v \times (T_4 - T_1) \text{ (kJ)} \]  
(10)

\[ Q_H = c_p \times (T_3 - T_2) \]  
(11)

Efficiency also calculated by

\[ \eta = 1 - \left[ \frac{Q_L}{Q_H} \right] \]

\[ W_{net} = Q_L - Q_H \]  
(12)

\[ V_{max} = \frac{RT_1}{P_1 \times 100} \]  
(13)
\[ V_{\text{min}} = \frac{V_{\text{max}}}{\left[\frac{P_2}{P_1}\right]} \]  
\[ P_{\text{mep}} = \frac{W_{\text{net}}}{100 \times (V_{\text{max}} - V_{\text{min}})} \]  
\[ IP = \frac{n \times P_{\text{mep}} \times L \times A \times N \times k \times 10}{6} \text{ (kW)} \]  
\[ BP = \frac{2 \times n \times N \times T_1}{60000} \text{ (kW)} \]  
\[ FP = IP - BP \text{ (kW)} \]  
\[ \eta_{\text{mech}} = \frac{BP}{IP} \]  
\[ \eta_p = 1 - \frac{1}{\left[\frac{P_2}{P_1}\right]} \]  

Specific output (SO) is given by Brake output per unit of piston displacement
\[ SO = \frac{BP}{A \times L} \]  

Mass of fuel in cylinder for combustion
\[ MFB = \frac{SFC \times BP}{60} \text{ (kg/min)} \]  

Air consumption (AC) in cylinder for a given fuel
\[ AC = MFB \times AFR \text{ (kg/min)} \]  
\[ V_s = \frac{\eta_c \times R \times 1000 \times T_1 \times N}{2 \times P_1 \times 100000 \times AC} \]  
\[ SFC = \frac{MFB}{BP} \text{ (Kg/kWh)} \]  
\[ \eta_l = \frac{IP}{MFB \times CV} \]  
\[ \eta_B = \frac{BP}{MFB \times CV} \]  

For petrol engine, heat supplied is given by
\[ HS = MFB \times CV \]  
Heat absorbed in indicated power
\[ IPE = IP \times 60 \text{ (kJ/min)} \]  
Heat taken away by cooling water
\[ Q_W = Q_H \times \frac{PE}{100} \]

Mass of water circulated for cooling of engine is given by

\[ M_W = \frac{n \times Q_W}{C_W \times (T_3 - T_1)} \text{ kg/s} \quad (27) \]

\[ Q_W = \frac{M_W \times C_W (T_3 - T_1)}{60} \text{ kJ/s} \quad (28) \]

Heat taken away by exhaust gases is given by

\[ HG = MG \times C_{PG} \times (T_e - T_r) \text{ kJ/min} \quad (29) \]

Dual Cycle Thermal Efficiency

\[ \eta_{th} = \frac{W_{net}}{Q_{in}} = \frac{Q_{in} - Q_{out}}{Q_{in}} \]

\[ \eta_{th} = 1 - \left( \frac{T_5 - T_1}{(T_3 - T_2) + \gamma (T_4 - T_3)} \right) \]

\[ c_p \frac{\gamma}{c_v} = \gamma, \text{ hence } \eta_{th} = 1 - \frac{T_5 - 1}{T_1} \left( \frac{T_4}{T_1} \right)^\frac{T_5 - 1}{T_1} \]

\[ \eta_{th} = 1 - \frac{1}{r_c^{\gamma+1}} \left[ \frac{a(\beta)^{-1}}{1 + \gamma a(\beta + 1)} \right] \quad (30) \]

The dual cycle approximates to the processes within a modern high speed diesel engine. In petrol engines, and large slow speed (< 500 RPM) diesels, the pressure rise because of combustion is comparatively rapid (relative to piston speed) and the cycle can be reasonably approximated by the constant volume or OTTO cycle.

3.13 Engine Performance

The basic performance parameters of internal combustion engine (I.C.E) may be summarized as follows:

**Indicated power (IP):**

IP power could be estimated by performing a Morse test on the engine.

\[ IP = P_m L A N \]

where N is the number of machine cycles per unit times, which is 1/2 the rotational speed for a four-stroke engine, and the rotational speed for a two-stroke engine.
Brake power (BP)
The break power is given by:

\[
BP = \frac{2aNT}{60} \text{ kW}
\]

where \( T \) is the torque

Friction power (FP) and Mechanical efficiency (\( \eta_m \))

The difference between the IP and the BP is the friction power FP and is that power required to overcome the frictional resistance of the engine parts,

\[
FP = IP - BP
\]

The mechanical efficiency of the engine is defined as

\[
\eta_m = \frac{BP}{IP}
\]

\( \eta_m \) is usually between 80% and 90%

Indicated mean effective pressure (imep)

It is a hypothetical pressure which if acting on the engine piston during the working stroke would result in the indicated work of the engine. This means it is the height of a rectangle having the same length and area as the cycle plotted on a p- v diagram.

\[
\text{imep (Pi)} = \left(\frac{\text{Net area of the indicator diagram}}{\text{Swept volume}}\right) \times \text{Indicator scale}
\]

Consider one engine cylinder:

Work done per cycle = \( \text{Pi AL} \)

where \( A \) = area of piston; \( L \) = length of stroke

Work done per min = work done per cycle \times \text{active cycles per min.}

\[
\text{IP} = \text{Pi AL} \times \text{active cycles/ min}
\]

To obtain the total power of the engine this should be multiplied by the number of cylinder (n), i.e.:

Total IP = \( \text{Pi AL Nn/2 for four- stroke engine and} \)

= \( \text{PiALNn for Two- stroke engine} \)
Brake mean effective pressure (bmep) and brake thermal efficiency

The bmep (Pb) may be thought of as that mean effective pressure acting on the pistons which would give the measured BP, i.e.

\[ BP = Pb \times AL \times \text{active cycles/ min} \]

The overall efficiency of the engine is given by the brake thermal efficiency, \( \eta_{BT} \), i.e.

\[ \eta_{BT} = \frac{\text{Brake power}}{\text{Energy supplied}} \]

\[ \eta_{BT} = \frac{BP}{m_f \times Q_{net}} \]

where \( m_f \) is the mass of fuel consumed per unit time, and \( Q_{net} \) is the lower calorific value of the fuel.

Specific fuel consumption (sfc)

It is the mass of fuel consumed per unit power output per hour, and is a criterion of economic power production.

\[ sfc = \frac{m_f}{B.P.} \text{ Kg/kWh} \]

Low values of sfc are obviously desired. Typical best values of bsfc for SI engines are about 270g/kWh, and for C.I. engines are about 200g/kWh.

Indicated thermal efficiency (\( \eta_{IT} \))

It is defined in a similar way to \( \eta_{BT} \)

\[ \eta_{IT} = \frac{IP}{m_f \times Q_{net}} \]

\[ \eta_m = \frac{BP}{IP} \]

Volumetric efficiency (\( \eta_v \))

Volumetric efficiency is only used with four-stroke cycle engines. It is defined as the ratio of the volume if air induced, measured at the free air conditions, to the swept volume of the cylinder:

\[ \eta_v = \frac{v'}{v_s} \] where air volume \( v' \) may be refereed to N.T.P. to give a standard comparison.

RESULTS AND DISCUSSION

In the following sections, the results have been calculated by developing software in C++ and the graphs plotted by using menu driven software “origin 50”. The tables shows the input values taken for the different results and by varying one or two parameters and keeping other parameters constants.
Table 1: Input values used for calculation

<table>
<thead>
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<th>S. No.</th>
<th>Description</th>
<th>Values</th>
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<tr>
<td>1</td>
<td>Compression Ratio (CR)</td>
<td>10</td>
</tr>
<tr>
<td>2</td>
<td>Initial Temperature T1</td>
<td>300 K</td>
</tr>
<tr>
<td>3</td>
<td>Torque (Tq)</td>
<td>100 N-m</td>
</tr>
<tr>
<td>4</td>
<td>Speed (N)</td>
<td>1500 rpm</td>
</tr>
<tr>
<td>5</td>
<td>Cylinder Diameter (D)</td>
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</tr>
<tr>
<td>6</td>
<td>Cylinder Length (L)</td>
<td>0.14 m</td>
</tr>
<tr>
<td>7</td>
<td>Constant (K)</td>
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</tr>
<tr>
<td>8</td>
<td>Initial Pressure (bar)Pa</td>
<td>1.01325,</td>
</tr>
<tr>
<td>9</td>
<td>Lower Calorific Value (CV)</td>
<td>42000 kJ/kgK</td>
</tr>
<tr>
<td>10</td>
<td>Mass of Cooling (Mw)</td>
<td>15 kg/min</td>
</tr>
<tr>
<td>11</td>
<td>Maximum Temperature (T3)</td>
<td>1500-2000K</td>
</tr>
</tbody>
</table>

Figure 5: Variation of Indicated Power with maximum temperature for different no. of cylinders

Figure 6: Variation of mechanical efficiency with cylinder temperature for no. of cylinders
Figure 7: Variation of Indicated Power with maximum temperature

Figure 8: Variation of Mechanical Efficiency with maximum temperature

Figure 9: Variation of Indicated Power with Compression ratio
Figure 10: Variation of Mechanical Efficiency with Compression Ratio

Table 2: Input values taken for results

<table>
<thead>
<tr>
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<tr>
<td>8</td>
<td>Initial Pressure (Pa)</td>
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</tr>
<tr>
<td>9</td>
<td>Lower Calorific Value (CV)</td>
<td>42000 kJ/kgK</td>
</tr>
<tr>
<td>10</td>
<td>Mass of Cooling (Mw)</td>
<td>15 kg/min</td>
</tr>
<tr>
<td>11</td>
<td>Compression Ratio (CR)</td>
<td>4-24</td>
</tr>
</tbody>
</table>

Figure 11: Variation of Indicated Power with Compression Ratio
Figure 12: Variation of Mechanical Efficiency with Compression Ratio

Table 3: Input values used for results

<table>
<thead>
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<th>Values</th>
</tr>
</thead>
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<td>2000 K</td>
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<td>Compression Ratio (CR)</td>
<td>10</td>
</tr>
</tbody>
</table>

Figure 13: Variation of Indicated Power with Engine Speed
In the following paragraphs, the graphs have been plotted for the requirement of water for the cooling of IC engines for various configurations.
Figure 17: Variation of Cooling water required with maximum temperature

Figure 18: Variation of Cooling water required with Compression Ratio

Figure 19: Variation of Efficiency with Compression Ratio
**Figure 20:** Variation of Cooling water required with No. of Cylinders

**Figure 21:** Variation of Power with Engine Speed

**Figure 22:** Variation of Power with Compression Ratio
Figure 23: Variation of Power with Engine Speed for different no. of cylinders

CONCLUSION

On increasing the maximum temperature of the cylinder, the indicated power increase for a given engine. On increasing the number of cylinders it increases. For 6 cylinder engine it is more compared to 2 or 3 cylinder engine.

Mass of cooling water required is decreases with increasing the compression ratio. On increasing the number of cylinders for a given engine, mass of cooling water required also increases. Indicated power increases on increasing the shaft speed for a given engine. Power developed in 2 strokes and 2 cylinders is more than the power developed in 4 stroke and 2 cylinders engine with other parameters same. Mechanical efficiency decreases on increasing the compression ratio. At low compression ratio it decreases more rapidly but on higher values of compression ratio it is almost constant.

REFERENCE


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