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Development of Mathematical Model for Cooling of Internal Combustion Engines

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ABSTACT: *The energy released in the combustion chamber of an internal combustion engine is dissipated in three different ways. About 35 % of the fuel energy is converted to useful crankshaft work, and about 30 % energy is expelled with the exhaust. This leaves about one-third of the total energy that must be transmitted from the enclosed cylinder through the cylinder walls and head to the surrounding atmosphere. The heat generated in the combustion chamber which is transferred to the engine body is cooled down to a reasonable temperature (that permits the continuous operation of the engine) basically for three reasons; firstly to promote a high volumetric efficiency, secondly to ensure proper combustion and thirdly to ensure mechanical operation and reliability. Non continuous cooling of the engine while in operation leads to overheating of the engine. This can affect the mechanical performances of an engine. Firstly, overheating can lead to a loss of strength. Secondly, the top piston ring groove temperature must also be limited to about 200^{0}C if the lubrication is to remain satisfactory. Above this temperature, lubricants can degrade, leading to both a loss of lubrication, and packing of the piston ring groove with products from the decomposed oil. Finally, failure can result through thermal strain. Strain can be caused by either mechanical or thermal loading. The thermal strain is directly proportional to the temperature gradient. This has led to the development of mathematical model for cooling of spark ignition internal combustion engines. The mathematical model was simulated to estimate the thermal system response for the user specified input heat profile, Q_{in}. The maximum heat input supplied to the system is approximately 1800 watts. This magnitude is substantially below the energy released during the actual engine combustion process. The engine, radiator, and junction node temperatures plotted against time for a series of symmetrical 10.0°C disturbances in the temperature set point. For the selected controller gains, the engine temperature displays small steady-state departures from the set point temperature of approximately 0.9°C. The junction temperature profile is very similar to the engine temperature as expected. The radiator temperature changes slightly which reflects the large thermal exhaust capabilities for this node.*

Keywords: *Combustion, Engine, Temperature, Overheating, Radiator.*

I. INTRODUCTION

1.1 Background of Study

There are two aspects to heat transfer within internal combustion engines. Firstly there is heat transfer from within the combustion chamber to its boundaries and secondarily there is heat transfer from the combustion chamber to its cooling media this aspect is discussed in this study.

First of all, the cooling requirements are considered on a global basis, as a function of engine type, load and speed, and cooling system type (that is, liquid or air- cooled). Also included here are a discussion of the heat flow in low heat loss Diesel engines (so –called 'adiabatic' engines) and the use of ceramic components or coatings as insulators. (Leidenfrost and Korenic 1982)

There are three reasons for cooling engines: firstly to promote a high volumetric efficiency, secondly to ensure proper combustion, and thirdly to ensure mechanical operation and reliability. The cooler the surfaces of the combustion chamber, then the higher the mass of air (and fuel) that can be trapped in the cylinder. In general, the higher the volumetric efficiency, the higher the output of the engine.(Mizushina, Ito and Miyashita ,1968)

In the case of spark ignition engines, cooling of the combustion chamber inhibits the spontaneous ignition of the air- fuel mixture. Since spark ignition engines have an essentially homogeneous mixture of fuel and air, then spontaneous ignition can affect a significant quantity of mixture, and the subsequent rapid pressure rise or so- called detonation, generates the characteristic 'knocking' sound. This process destroys the thermal boundary layer, and can lead to overheating of components and ensuing damage. (Tezuka, Takada and Kasai, 1976)

The heat rejected to the coolant in spark ignition engines is a function of the: speed, load, ignition timing and air/ fuel ratio. The large number of dependent variables means that even when comprehensive energy balance data are published, not all the variables are likely to have been investigated systematically, nor will the test conditions have necessarily been fully defined.

Gruden and Kuper (1987) present a series of contour plots for the different energy flows (fuel in, brake power, coolant, oil, exhaust) as functions of bmep and engine speed for a 2.5 litres spark ignition engine. They also present contour plots of the brake, mechanical and indicated efficiency. The brake efficiency results imply that the engine has been tuned for maximum economy at part load, while at full throttle the mixture has been richened to give the maximum power. Their mechanical efficiency is directly affected by the load (with zero mechanical efficiency by definition at no load). Friction dissipates useful work as heat, some of which appears in the coolant and some in the oil. The heat loss recorded to the oil is almost solely a function of speed, with about 5 kW dissipated at 3000 rpm, and 15 kW dissipated at 6000 rpm. Coolant is comparable to the brake power output. In the load range of 8-10 bar bmep, the energy flow to the coolant is about half the brake power output. However, the exhaust energy comprises in part the chemical energy of the partial combustion products: at full load the chemical energy is comparable to the exhaust thermal energy. The effect of the air/fuel ratio or equivalence ratio on the energy flow to the coolant is illustrated by some data obtained from a gas engine (Johnson, Kreid, and Hanson., 1983). The gas engine has an entirely homogeneous air- fuel mixture. So that it is possible to run with a very wide range of equivalence ratios. The equivalence ratio of 0.75 corresponds to an air/fuel ratio of about 20:1 for a gasoline fuelled engine. Firstly, in terms of the total fuel energy supplied, the heat flow to the coolant is nearly constant at 28 per cent of the supplied energy, with a slight fall with rich mixtures to 25 per cent at an equivalence ratio of 1.2. Next, the energy flow to the coolant can be considered as a fraction of the brake power output, but as this remains close to unity (within experimental tolerances), it has not been plotted here. Finally, the absolute values of the energy flow to the coolant have been plotted. From the foregoing discussion, it can be seen that the energy flow to the coolant is a reflection of the way in which the engine brake output responds to the variation of the equivalence ratio. Advancing the ignition timing leads to an increased absolute value of heat rejected to the coolant, if the throttle, speed and air/fuel ratio are fixed. Earlier ignition causes higher temperatures in both the burned and unburned gas, and this leads to higher levels of heat rejection from the combustion chamber.

Raising the compression ratio also increases the in-cylinder gas temperatures, but this does not necessarily lead to an increase in the heat flow, to the coolant. The higher compression ratio increases the work output from each charge that is ignited, and thus the exhaust temperature is lowered. Consequently, the heat rejected to the exhaust valve and exhaust port is reduced, and this can offset any increase in heat flow from the combustion chamber. As the compression ratio continues to be increased, the gains in work output reduce and the surface- to- volume ratio of the combustion chamber deteriorates, so there will be a compression ratio above which the heat flow to the coolant increases.

The material of the engine, and in particular the cylinder head, can affect the engine performance. Tests reported by Gruden and Kuper (1987) included comparisons of the energy balance with cast iron and aluminium alloy cylinder heads that were otherwise identical.

When spark ignition engines were the common form of aircraft propulsion, it was accepted that air-cooled engines had a slightly lower output and efficiency. However, the weight saving associated with an air-cooled engine meant that for journeys of up to five or six hour's duration, the overall engine plus fuel' weight of an air- cooled engine was lower than the weight of a liquid-cooled engine and its fuel (Judge 1967). A direct comparison has been made by Gruden and Kuper (1987) between liquid – and air- cooled engines; they concluded that there were no significant differences.

In recent years, there has been much interest in the application of ceramics to compression ignition engines. The simple argument is that ceramics have a much lower thermal conductivity than metals (one or two orders of magnitude) so that the energy flow to the coolant will be reduced, and the higher combustion temperatures will lead to more expansion work. However, the largest thermal resistance is in the thermal boundary layer adjacent to the combustion chamber, and this will not be affected much. This system can be modeled as a series of thermal resistances to represent the thermal boundary layers on the gas side and the coolant side, and a thermal resistance to control the heat flow through the combustion chamber wall. If the heat flow is considered to be steady, then the thermal resistances are proportional to the temperature differences (cf. voltage differences). If the heat flow (cf. current) is to be changed, then the largest effect will be obtained by changing the largest thermal resistance that is the gas side heat transfer coefficient. Thus an order of magnitude change to the thermal conductivity of the combustion chamber wall does not lead to an order of magnitude change in the heat flow (Web, 1984). It should be noted that the gas side heat transfer coefficient is increased, so that some of the gain in cylinder head insulation is offset. However, because of the time- dependent variation of gas temperature within the combustion chamber, there are other factors controlling the heat flow.

Propagation of heat from combustion chamber of an internal combustion engine can be considered to be in two aspects. Firstly there is heat transfer from within the combustion chamber to its boundaries (surrounding walls) and secondly there is heat transfer from the combustion chamber to its cooling media. The heat generated in the combustion chamber which is transferred to the engine body is cooled down to a reasonable temperature (that permits the continuous operation of the engine) basically for three reasons:

i. Firstly to promote a high volumetric efficiency
ii. Secondly to ensure proper combustion
iii. Thirdly to ensure mechanical operation and reliability.

The cooling requirements are considered on a global basis, as a function of:

i. Engine type
ii. Load and speed
iii. Cooling system type (that is, liquid or air- cooled).

II. PROBLEM STATEMENT

Non continuous cooling of the engine while in operation leads to overheating of the engine. This can affect the mechanical performances of an engine. The engine performance can be affected in the following ways:

i. Overheating which can lead to a loss of strength.
ii. The top piston ring groove temperature must also be limited to about 200^0c if the lubrication is to remain satisfactory. Above this temperature lubricants can degrade, leading to both a loss of lubrication, and packing of the piston ring groove with products from the decomposed oil.
iii. Finally, failure can result through thermal strain. Strain can be caused by either mechanical or thermal loading. The thermal strain is directly proportional to the temperature gradient. Hence, the importance of monitoring the temperature variation of an engine during operation. Modeling is one of the tools for such monitoring.

The cooling requirements are considered on a global basis, as a function of:

i. Engine type
ii. Load and speed
iii. Cooling system type (that is, liquid or air- cooled).

III. OBJECTIVE OF THE STUDY

The main objective of this study is to develop a mathematical model that will predict the temperature of radiator and heater (engine block) with respect to time.

IV. METHODOLOGY

The mathematical modeling was carried out considering the heat propagation from combustion chamber to surrounding air. The convective exchange coefficient between burning gases and combustion chamber wall was determined. The differentiated cooling of CIE (high temperature of cylinder head) was organized in two variants: one by mean of coolant controlled flow rate between cylinder block and cylinder head and other by mean of thermal pipes, transferring a certain amount of heat from exhaust gases to the cylinder head cooling circuit.

The following main parameters were observed:

i. Engine output,
ii. Fuel consumption,
iii. Engine efficiency
iv. Pollutants values.

The governing equations for temperature of radiator, heater and junction are given below:

$$\frac{dT_e}{dt}=\frac{1}{C_e}\left(-\frac{1}{R_1}f(T_e,T_r)-\frac{1}{R_2}f(T_e,T_j)+\frac{1}{R_4}f(T_j,T_e)+Q_{in}\right)$$

$$\frac{dT_r}{dt}=\frac{1}{C_r}\left(\frac{1}{R_1}f(T_e,T_r)-\frac{1}{R_3}f(T_r,T_j)-Q_{out}\right)$$

$$\frac{dT_j}{dt}=\frac{1}{C_j}\left(\frac{1}{R_2}f(T_e,T_j)+\frac{1}{R_3}f(T_r,T_j)-\frac{1}{R_4}(T_j,T_e)+Q_{pump}\right)$$

Where:

C_e **heater capacitance (J/^{o}K)**
C_r **radiator capacitance (J/^{o}K)**
C_j **junction capacitance (J/^{o}K)**
R_1 **resistance between valve-cooler (^{o}K/W)**
R_2 **resistance between valve-junction (^{o}K/W)**
R_3 **resistance between cooler-junction (^{o}K/W)**

Source: (Leidenfrost and Korenic 1982)

V. DISCUSSION

Using analytical (Laplace Transform) method of analysis to solve the Governig Eqautions, the following results were obtained

$$T_e=\frac{C_{pc}\dot m_r\ (298\Sigma B_{pc}\dot m_f-\theta+\theta_{in})}{\alpha_1\ \alpha_2}+\left[\frac{\begin{array}{c}298C_cC_r\alpha_1^2+(\theta_oC_r+298C_{pc}\dot m_f+298C_{pc}C_r\dot m_r)\alpha_1\\+C_{pc}\dot m_r(298\sum C_{pa}\dot m_f-\theta_o+\theta_{in})\end{array}}{3\alpha_1^2-2(\alpha_1^2+\alpha_2)\alpha_1+\alpha_1\,\alpha_2}\right]\ell^{\alpha_1 t}+$$

$$\left[\frac{\begin{array}{c}298C_cC_r\alpha_1^2+(\theta_oC_r+298C_{pc}\dot m_f+298C_{pc}C_r\dot m_r)\alpha_2\\+C_{pc}\dot m_r(298\sum C_p\dot m_f-\theta_o+\theta_{in})\end{array}}{3\alpha_2^2-2(\alpha\ \ +\alpha_2)\alpha_2+\alpha_1\,\alpha_2}\right]\ell^{\alpha_1 t}$$

$$Let\ \ \alpha_1=\frac{-C_{pc}C_c\dot m_r+\sqrt{C_{pc}\dot m_r\ \ (C_c+C_r)^2-4\Sigma C_{pc}C_{pa}(\dot m_r\dot m_f C_cC_r)}}{2C_eC_r}$$

$$Let\ \ \alpha_2=\frac{-C_{pc}C_c\dot m_r(C_e+C_r-\sqrt{C_{pc}\dot m_r\ \ (C_c+C_r)^2-4\Sigma C_{pc}C_{pa}(\dot m_r\dot m_f C_cC_r)}}{2C_eC_r}$$

$$T_r=\frac{298\sum C_{pa}C_{pc}\dot m_f\dot m_r-\theta_o\dot m_rC_{pc}-\theta_{in}(298\sum C_{pa}\dot m_f-C_{pc}\dot m_r)}{\alpha_1\ \alpha_2}$$

$$\frac{\begin{array}{c}+298C_cC_r\alpha_1^2+(298\sum C_{pa}\dot m_f-\theta_oC_e+298C_rC_{pc}\dot m_r+298C_cC_{pc}\dot m_r)+298C_e\sum C_{pa}\dot m_f)\\+298C_e\sum C_{pa}\dot m_f)\alpha_1\\ \\+298\sum C_{pa}\dot m_f\dot m_r-\theta_o\dot m_rC_{pc}-\theta_{in}(\sum C_{pa}\dot m_f-C_{pc}\dot m_f-\sum C_{pa}\dot m_r)\end{array}}{3\alpha_1^2-2(\alpha_1+\alpha_2)\alpha_1+\alpha_1\alpha_2}$$

$$+298C_cC_r\alpha_2^2+(298\sum C_{pa}\dot{m}_f-\theta_oC_e+298C_rC_{pc}\dot{m}_r+298C_cC_{pc}\dot{m}_r)+298C_eC_{pa}\dot{m}_f$$
$$+298C_e\sum C_{pa}\dot{m}_f)\alpha_2$$

$$\frac{+298\sum C_{pa}C_{pc}\dot{m}_f\dot{m}_r-\theta_o\dot{m}_rC_{pc}-\theta_{in}(\sum C_{pa}\dot{m}_f-C_{pc}\dot{m}_r-C_{pa}\dot{m}_r)}{3\alpha_2^2-2(\alpha_1+\alpha_2)+\alpha_1\alpha_2}\ .$$

$$\theta=\frac{\dfrac{V}{K_b}+(\beta_1(L\beta_1+R)\left[0.5cd\,(\dfrac{n_w}{n_m})\;A\Delta P\left(1+0.5cd\,(\dfrac{n_w}{n_m})z\right)\right]+K_mV\ell^{\beta_1 t}}{4J_{eq}L\beta_1^3+(bL+RJ_{eq})\beta_{11}^2+2bR\beta_1+K_bK_m}$$

$$+\frac{\left((\beta_1(L\beta_1+R)\left[0.5d\,(\dfrac{n_w}{n_m})\;A_p\Delta P\left(1+0.5d\,(\dfrac{n_w}{n_m})z\right)\right]+K_mV\right)\ell^{\beta_1 t}}{4J_{eq}L\beta_2^3+3(bL+RJ_{eq})\beta_2^2+2bR\beta_2+K_bK_m}$$

$$+(LS+R)(S^2+bs)+K_mK_b)\tilde{l}=\frac{V}{S}\;(J_{eq}S^2+bs)$$
$$-(LS+R)\left[0.5d\,(\frac{n_w}{n_m})\;A_p\Delta P\;(1+0.5cd\,(\frac{n_w}{n_m})z\right]$$

The mathematical model was simulated to estimate the thermal system response for the user specified input heat profile, Q_{in}. The maximum heat input supplied to the system is approximately 1800 watts. This magnitude is substantially below the energy released during the actual engine combustion process. The engine, radiator, and junction node temperatures versus time are displayed in **Figure 5.1** for a series of symmetrical 10.0°C disturbances in the temperature set point. For the selected controller gains, the engine temperature displays small steady-state departures from the set point temperature of approximately 0.9°C. The junction temperature profile is very similar to the engine temperature as expected. The radiator temperature changes slightly which reflects the large thermal exhaust capabilities for this node.

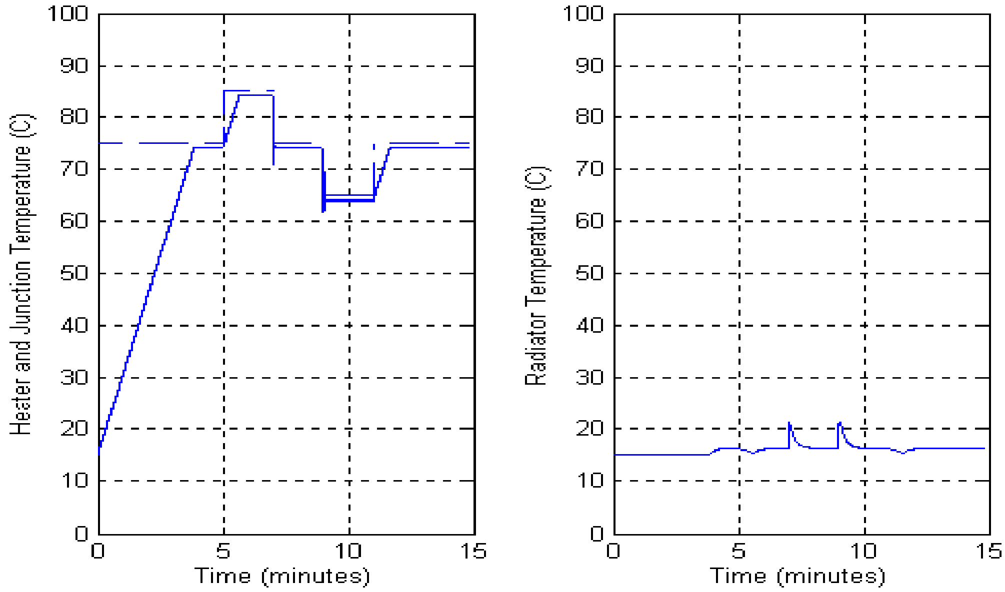

Figure 5.1 Temperature vs. time

Figure 5.2 shows the Experimental scale cooling control system hardware for real-time algorithm studies while **Figure 5.3** shows the experimental cooling system with flow direction.

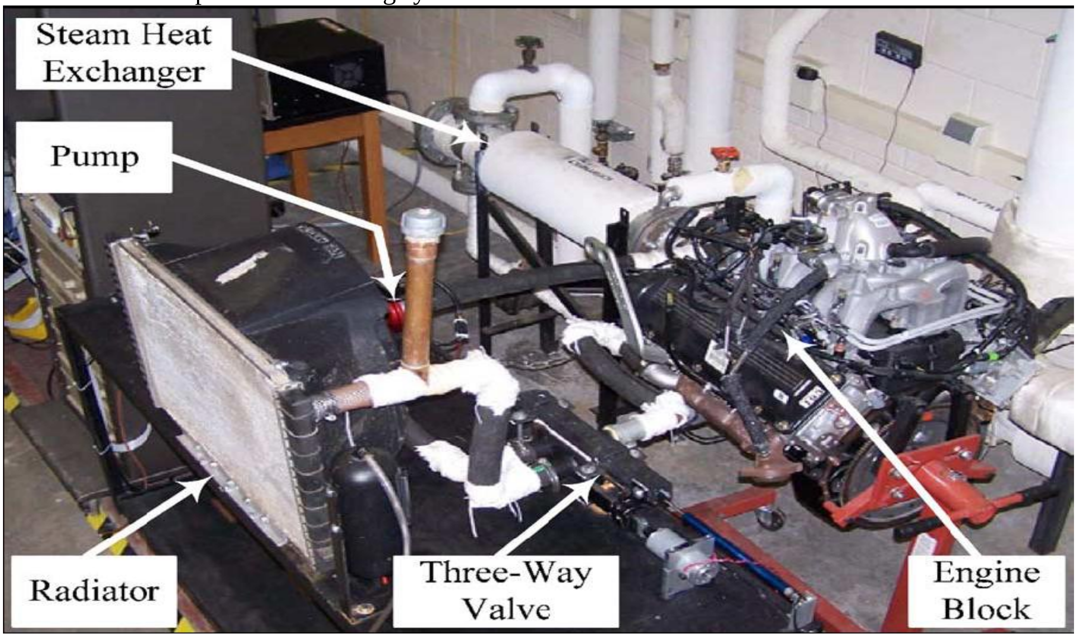

Figure 5.2 Experimental scale cooling control system hardware for real-time algorithm studies
Source: (Parker and Treybal,1961)

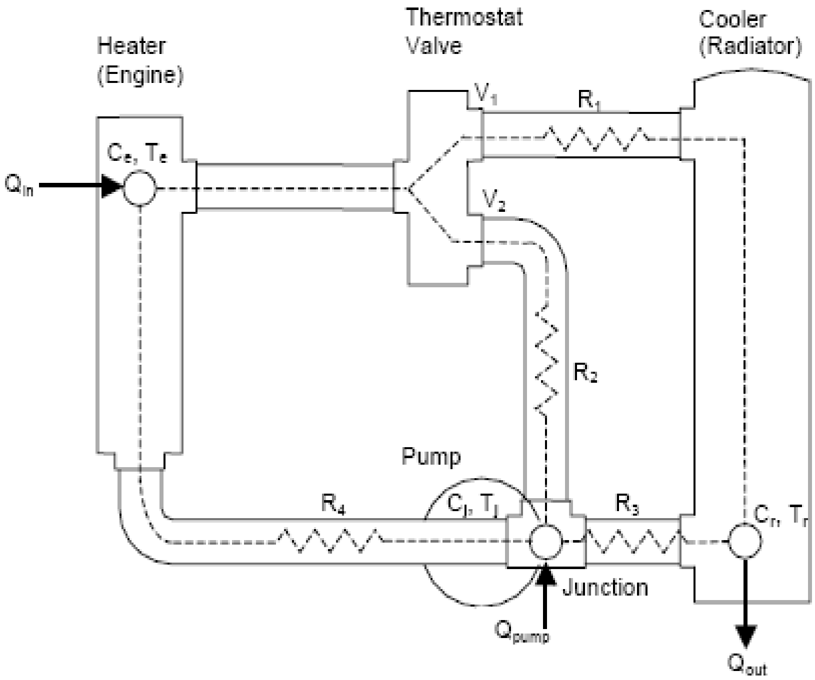

Figure 5.3: Experimental cooling system with flow direction
Source: (Nottage, H.B., 1941**)**

VI. CONCLUSION

The energy released in the combustion chamber of an internal combustion engine is dissipated in three different ways. About 35 % of the fuel energy is converted to useful crankshaft work, and about 30 % energy is expelled with the exhaust. This leaves about one-third of the total energy that must be transmitted from the enclosed cylinder through the cylinder walls and head to the surrounding atmosphere. The heat generated in the combustion chamber which is transferred to the engine body is cooled down to a reasonable temperature (that permits the continuous operation of the engine) basically for three reasons; firstly to promote a high volumetric efficiency, secondly to ensure proper combustion and thirdly to ensure mechanical operation and reliability. Non continuous cooling of the engine why in operation leads to overheating of the engine. This can affect the mechanical performances of an engine.

This has led to the development of mathematical model for cooling of spark ignition internal combustion engines. The mathematical model was simulated to estimate the thermal system response for the user specified input heat profile, Q_{in}. The maximum heat input supplied to the system is approximately 1800 watts. This magnitude is substantially below the energy released during the actual engine combustion process. The engine, radiator, and junction node temperatures plotted against time for a series of symmetrical 10.0°C disturbances in the temperature set point. For the selected controller gains, the engine temperature displays small steady-state departures from the set point temperature of approximately 0.9°C. The junction temperature profile is very similar to the engine temperature as expected. The radiator temperature changes slightly which reflects the large thermal exhaust capabilities for this node.

Thus, the output seems to stay at the same level. Fuel consumption has decreased up to 20%, engine efficiency increased from $\eta = 0.36$ in case of classic cooling up to $\eta = 0.39$ in case of differentiated cooling. Better values of the measured pollutants were recorded on almost all speed and load interval.

VII. RECOMMENDATION

When there is non-continuous cooling of the engine while in operation, overheating of the engine is one of the resultant effects. The poor performances of the engine such as loss of strength and loss of lubrication can also be experienced. It is recommended that proper cooling mechanism should always be setup in any internal combustion engine to avoid all the aforementioned problems. Also, appropriate models should be set up for different types internal combustion engines.

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