

Effect of Active Cooling Control on Internal Combustion Engine Exhaust Emissions and Instantaneous Performance Enhancement

Abstract: Energy flows and energy efficiencies in the operation of a modern automobile are development expressed and corresponding on reduction of pollutant emission from spark ignition (SI) engines is desirable due to reduce the highly impact on the green environment, the trend toward high engines efficiency has led to new implementation of more intelligent engine thermal management (ETM) systems. However, the present work aims to clear the effect of different cooling systems and operating parameters of the SI engine on their engine performance and emission characteristics. An active coolant control (ACC) as well as of engine ETM has been realized using a variable-speed electrical pump, and a variable-position smart valve which controls the fluid flow rate distribution between the radiator and its bypass line to better regulate heat rejection. Experimental engine cooling systems have been operating assembled two different cooling systems (conventional or ACC). The Nissan SI engine was used as the experimental set-up and equipped with exhaust gas analyzer, Thermocouple sensors and magnetic pickup transducer to measure the exhaust emissions, coolant temperature and engine rotational speed respectively. The measurements were conducted at different engine speed and load torque. The results indicate that most of the carbon monoxide (CO), carbon dioxide (CO₂), and unburned total hydrocarbons (THC) emission appears at higher load as well as near the coolant temperature. Nitrogen oxides (NO_x) emission was observed to increases because higher coolant temperature contributed to raising combustion temperature, the results show that the arrangement of ACC flow rate and operation in idle state is very effective to reduce exhaust emissions. The assessment results of the experimental tests revealed that warm up period with ACC was decreased by 31%, In addition, CO₂, CO, THC, and NO_x, of SI engine with ACC were reduced by 11%, 13.5%, 8.4%, 6.2%, and 26.0%, respectively.

Keywords: *Active cooling control; Exhaust emission; Thermal management; engine warm up; engine performance coolant temperature.*

1. Introduction

One effective way to alleviate the air pollution situation in a city is to issue regulations on maximum allowable limits of vehicle Internal Combustion Engine (ICE) exhaust gas release, the effect of engine operating parameters like mixture quality, Engine thermal management (ETM) and ignition systems were studied intensively. For spark ignition (SI) engine a reasonable solution for reducing emissions is by controlling some advanced cooling systems, in such way engine performance is kept unaltered. A combustion engine consists of different subsystems that work together to achieve efficient combustion. Improving the efficiency or reducing the losses in one of these subsystems can reduce the fuel consumption. One such subsystem is the cooling system.

The heat transfer rate in an engine is dependent on the coolant temperature and the engine size, among other variables. Complex interactions exist between various operational parameters. For example, as the temperature of the engine coolant decreases, the heat transfer to the coolant will increase, and the combustion temperature will decrease [1]. The engine-cooling system can be improved significantly with the inclusion of advanced design and operating features, allowing the engine-cooling system to operate efficiently and effectively, indirectly improving fuel economy and lowering emission output. With greater emphasis placed on improving fuel economy are lowering emissions output from modern IC engines, engine downsizing & raising power density are the favored options. Through this route, modern engines can attain similar power outputs to larger convectional engines with reduced frictional losses [2].

Dynamic models for the thermal system, smart thermostat valve, coolant pump, and radiator fan have been described in [3:5], where the cooling system configuration has been tested by replacing the engine with an electronic immersion heater and by operating the actuators via a non-linear tracking controller, A servo motor valve and a pump were installed into the cooling system to regulate the coolant flow with engine control unit. Enhancement of gasoline and

diesel engine thermal management systems through a smart thermostat valve and a variable speed water pump has been discussed in [6], where a non-linear tracking controller has been proposed to ensure that engine temperature follows the desired trajectory. A non-linear controller has also been designed for transient temperature tracking in an experimental system involving a variable position smart valve, a variable speed electrical water pump, a variable speed electrical radiator fan, an engine block, and various sensors, in this system a steam-based heat exchanger mimics the heat generated by the engine's combustion process. The power consumption associated with four different simulated solutions has been compared in a comprehensive assessment of different configurations for a heavy-duty diesel engine [7, 8]. A reduction of coolant flow is mentioned by various authors as an important point for the improvement of engine efficiency. Some authors have proposed to throttle the conventional belt-driven cooling pump. It is evident that this method is less efficient than the substitution of the conventional pump with an electrical one [9, 10]. Nevertheless the importance of a better control of the flow partitioning between the bypass duct and the radiator has been largely ignored. ETM studies have drawn attention as a high-potential field to improve efficiency and reduce emissions of internal combustion engines. As a primary route for wasted heat energy, the cooling system particularly affects the thermal efficiency through the combustion temperature. However, the conventional cooling system has structural problems that are directly dependent on engine speed. In addition, the coolant flow can only travel in one route: from block to head. Because the flow rate of the coolant is determined by the thermal limitations of specific components requiring the largest cooling loads, this conventional configuration led to the problem of over cooling problem in the specific components because of the difference in thermal limitations. Several ETM studies have been conducted on coolant systems with electric devices, which increase the advantages of thermal management [11, 13]. An automotive company launched the split cooling system in a small gasoline engine to control each component temperature independently [14].

The exhaust gas emission and fuel consumption are highly related to driving modes: idle, steady-state cruise, acceleration and deceleration. It can be noted that the pollutant emission is relatively sensitive to the quality and accuracy of description to the driving speeds. Thus, a certain level of detail in the speed description is necessary for an accurate evaluation of emission. The emission from vehicles equipped with SI engines contributes in raising the pollutants level of atmosphere and consequently the global environment. This pushes the scientific society and authorized institutions to review the current legislations to reduce the emission [15, 16]. Most of previous work focused on studying the engines performance and emission in labs and not in their real environment. The present work is oriented towards the evaluation of emission from vehicle engines at real road test condition. The measurements were carried out at different vehicle speeds and gearshifts. The concentrations of CO₂, CO, THC and NO_x as well as engine speed are recorded for each vehicle speed and gearshift.

The paper is organized as follows. Section 2 presents the structure of the cooling system development model based on a variable-speed electrical fluid pump, and a variable-position smart valve. Section 3 describes the experimental apparatus and instrumentation. Section 4 presents the simulation and experimental results, the actual measurements of coolant temperature and exhaust emissions of an engine tests, these measurements were conducted at different engine operating conditions. Section 4 conclusions.

2. Cooling System Model

2.1 Principles of cooling system active thermal management

The typical automotive cooling system has two main thermal components: engine and radiator. The coolant flow through the engine loop transports excess combustion heat to the radiator loop which dissipates this heat. Controlling and directing coolant flow between these two loops is the main function of most thermostat valves.

Traditional cooling system, the common cooling system has three key components working to regulate engine temperature: thermostat, water pump, and radiator fan (refer to Figure 1a). In operation, when the engine is cold, the thermostat is closed and coolant is forced to flow through an internal engine bypass (usually a water passage parallel to the engine water jackets). Once the coolant reaches the desired operating temperature, the thermostat begins to open and allow coolant to flow through the radiator where excess heat can be rejected. Coolant flowing through the radiator is further cooled by the radiator fan pulling air across the radiator. When the coolant has dropped below the thermostat temperature rating, the valve closes (via spring force) directing the coolant again through the bypass. Conventional thermostats are wax based; their operation depends on the material properties of the wax in the thermostat housing and the coolant temperature surrounding it [16]. Traditional water pumps and

radiator fans are generally mechanically driven by the engine's crankshaft. Specifically, the water pump is driven as an accessory load while the radiator fan is often connected directly to the crankshaft with a clutch. Advanced cooling system, a three-way valve controls coolant flow through the bypass and radiator loops as shown in **Figure 1b**, the coolant flow can be completely blocked from entering the radiator or bypass which aids in engine warm-up time. In addition, valve geometries can become complicated when designing a three-way valve that proportionally controls coolant flow while minimizing the pressure drop. The term "thermal management" describes the efficient control of thermal energy flows in the vehicle in accordance with the specific requirements and the prevailing operating and load conditions. As a result, engine emissions can be reduced; also the thermal state, friction torque and mechanical engine efficiency can be improved. This leads to lower fuel consumption, longer engine life and upswing in thermal comfort. The thermal response of advanced cooling system can be represented by a suite of lumped parameter mathematical descriptions. The automotive cooling elements included a 1600cc petrol engine. The engine's thermal management system features a three-way smart valve, variable speed electric coolant pump, and a variable speed electric radiator fan. Finally, the charge-air-cooler was a simple loop to reduce air temperature after compression with an integrated coolant pump [17].

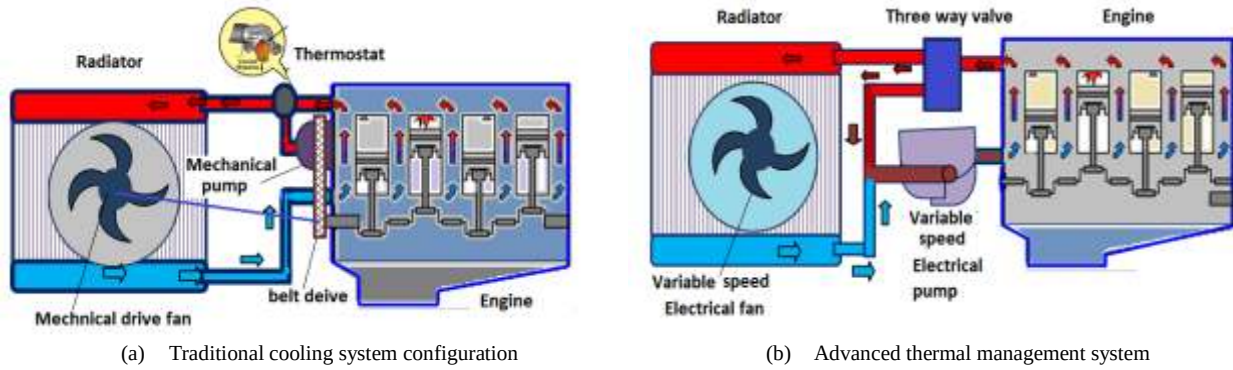


Figure 1: Traditional and advanced spark ignition engine thermal management system

2.2 Cooling-system thermal descriptions and operating strategy

Complete engine behaviour simulations were run at different thermal levels, ranging from cold to fully warm up. In the boost program the different thermal levels were simulated by taking different values for the liner mean temperature, which for the present engine ranges from 95°C (fully warmed up), a simplified circuit of cooling as shown in **Figure 2**. The heat rejection from the engine to coolant was calculated at different operation conditions by using the following equation [18]:

$$Q_{cooling} = \dot{m}_c C_p (T_{e2} - T_{e1}) \quad (1)$$

$$\dot{m}_c = \rho_c \dot{V}_c$$

Where: $\dot{m}_c$ coolant mass flow rate, C_p specific heat capacity of the coolant, T_{e1} temperature at engine inlet T_{e2} temperature at engine outlet, ρ_c coolant fluid density $\dot{V}_c$ coolant volume flow rate, the coolant valve lets more coolant through the radiator, thus increasing $\dot{m}_r$ and simultaneously reducing the coolant mass flow $\dot{m}_{py}$ in the by-pass circuit. The thermal dynamics for the engine and radiator nodes, $T_e(t)$, $T_r(t)$, the overall coolant mass flow rate is

$$\dot{m}_c = \dot{m}_r + \dot{m}_{by} \quad (2)$$

The enthalpy balance for the mass of coolant in the radiator is:

$$\dot{m}_r C_p \frac{dT_{r2}}{dt} = \dot{m}_r C_p [T_{e2} - T_{r2}] - \dot{Q}_r \quad (3)$$

The Effectiveness of the radiator is given below:

$$\varepsilon = \frac{Q_{actual}}{Q_{Max}} = \frac{\dot{m}_c C_p (T_{e2} - T_{e1})}{\dot{m}_a C_a (T_{e2} - T_a)} \quad (4)$$

The mass flow of coolant from the by-pass circuit and the mass flow of coolant from the radiator mix at the pump inlet. Since all components except the engine and the radiator are assumed to be adiabatic, the coolant temperature in the pump does not change, and the same temperature is found both at the pump outlet and at the engine inlet.

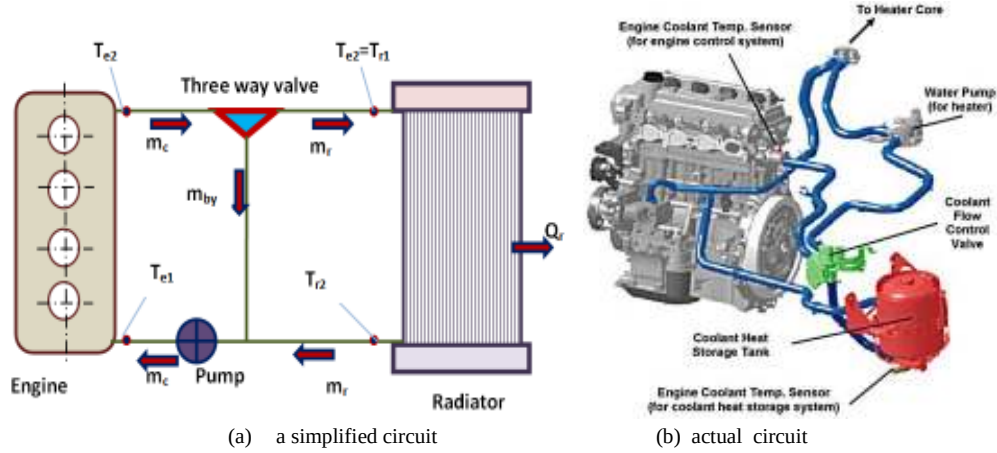


Figure 2: active cooling system

2.3 Electrical coolant pump

The application of electric coolant pump enables the reduction of the power consumption by optimizing coolant flow control. In addition, an electric coolant pump has higher efficiency than a mechanical pump. One of the reasons is that a mechanical pump receives driving power through belts or gears. Its configuration such as drive shaft and auxiliary elements restrict to increase hydraulic efficiency. However, an electric pump is directly driven by electric brushless DC motor (BLDC), which can eliminate external elements such as belt pulley and offer the more flexibility of pump configuration design to enhance the efficiency. The speed control Input unit provides motor speed input to the control system. The actual motor speed is fed back to the closed-loop controller. The PI controller is used as the closed-loop control algorithm to track the actual motor speed and also apply the speed control input. Based on speed control input and present and past errors (proportional and integral values), the closed-loop control either increases or decreases the pulse width modulation (PWM) duty cycle, which in turn controls the speed of the motor.

Pump performance maps were used in order to calculate pump power consumption. Pump performance maps consist of pressure rise, flow rate, pump speeds, and efficiencies. Based on these map data, the pump power consumption was calculated using the following equation.

$$P_{pump} = Q_p * \eta * \Delta P \quad (5)$$

Where P : pump power consumption, η : pump efficiency, Q_p : volumetric flow rate, Δp : pressure rise across the pump. A computer-controlled electric motor operates the high capacity centrifugal water pump.

The motor's armature current $i_{ap}(t)$ can be described as:

$$\frac{di_{ap}}{dt} = \frac{1}{L_{ap}} (V_p - R_{ap} i_{ap} - K_{bp} \omega_p) \quad (6)$$

where the motor's angular velocity $\omega_p(t)$ can be computed as

$$\frac{d\omega_p}{dt} = \frac{1}{J_p} (-b_p \omega_p + R_f V_o^2 \omega_p + K_{mp} i_{ap}) \quad (7)$$

The coolant mass flow rate for a centrifugal water pump depends on the coolant density, shaft speed, system geometry, and pump configuration, the mass flow rate may be computed as:

$$\dot{m}_c = \rho_c (2\pi * r_p * b * v) \quad (8)$$

$$v = \omega_p r_p \tan \beta = \frac{2\pi n_p}{60} r_p \tan \beta$$

It is assumed that the inlet radiator velocity v is equal to the inlet fluid velocity, and that the flow enters normal to the impeller.

2.4 Variable-Position Smart Valve

A DC servo-motor has been actuated in both directions to operate the multiposition smart thermostat valve. The compact motor, with integrated external potentiometer for position feedback, is attached to a worm-gear assembly

that is connected to the valve's piston. The governing equation for the motor's armature current $i_{av}(t)$ can be written as:

$$\frac{di_{av}}{dt} = \frac{1}{L_{av}} \left(V_v - R_{av} i_{av} - K_{bv} \frac{d\theta_v}{dt} \right) \quad (9)$$

2.5 Variables-Speed Radiator Fan

A cross-flow heat exchanger and a dc servo-motor-driven fan form the radiator assembly. The electric motor directly drives a multi-blade fan that pulls the surrounding air through the radiator assembly. The air mass flow rate going through the radiator is affected directly by the fan's rotational speed ω_f so that:

$$\frac{d\omega_f}{dt} = \frac{1}{J_f} (-b_f \omega_f + K_{mf} i_{af} - \rho_a A_f R_f V_{af}^2) \quad (10)$$

The fan motor's armature current $i_{af}(t)$ can be described as

$$\frac{di_{af}}{dt} = \frac{1}{L_{af}} (V_f - R_{af} i_{af} - K_{bf} \omega_f) \quad (11)$$

A voltage-divider circuit has been inserted into the experimental system to measure the current drawn by the fan and estimate the power consumed. [18]

3. Experimental Apparatus

3.1 Description of experimental tests

The revised cooling system used in this study is shown in **Figure 3a**. It consisted of a cooler substituted for a radiator in real vehicles, an electric water pump, and a variable-position smart valve replaced with the wax-type thermostat. The engine for this experiment was the 1.6L SI engine coupled with hydraulic dynamometer. The specifications of the Nissan engine are summarized in **Table 1**. The setup is also provided with necessary instruments for measurements of engine torque. Thermocouple sensors were used to measure the coolant temperature $T_c(t)$, and $T_r(t)$, and the temperature data were sent to a control module circuit to determine a variable-position smart valve opening ratio and the temperature sensors are connected to the DAQ controller. The electrical water pump was operated electrically by connecting it with a DC motor. To select the capacity of DC motor, the driving torque of the water pump was measured. This motor had a uniform driving torque for all pump speed range. The (K8004) sends the pump signal through a driver circuit which amplifies the signal to a recognizable level for the pump. The outputs of the microcontroller (K8004) will be modulated using PWM. PWM will allow for greater control of the speed and operation time of the water pump. And USB experiment interface board (K8055) has 5 digital inputs and 8 outputs and the operation controls the rate of a variable-position smart valve opening and closing of to be 10 KHz, the maximum supply voltage is limited to 24V.

The future modifications to the oil and coolant circuits allowed control with different fuels should be explained of the following parameters:

- The engine-out coolant flow control.
- The coolant flow in the exhaust gas recirculation EGR cooler circuit.
- The coolant flow to the oil cooler.
- The oil pressures using the variable-flow oil pump (VFOP).
- The use of coolant- or oil-cooled EGR.
- A change in the other common calibration parameters, notably the injection timing.

Table 1. engine specification.

S/No.	1	2	3	4	5	6	7
Parameters	Type	Cylinder array	Swept volume	valves per cylinder	Compression ratio	Maximum power	Maximum torque
Values	Nissan/gasoline	In line 4 cylinder	1600 (cm ³)	Four	10:1	92 HP@ 5000 rpm	165 Nm @ 3200 rpm

Figure 3b presents an overview of the laboratory dynamometer type DT400-1 and engine is supported in experimental work. **Figure 3c** shows Interfaces of hydraulic dynamometer type DT with controller packet; the braking power of the dynamometer is controlled through the water filling level by a motor-operated control valve installed in the cooling water outlet at the bottom. The lowest torque of the dynamometer, the so-called idling

characteristics, results from minimal water filling. This characteristic is not zero if there is no water in the dynamometer because in this case air whirls instead of water whirls develop, with minimal water filling neither air nor water whirls can develop completely which causes the lowest torque. This so-called “idling power” is highest at maximal speed of the dynamometer and correspondingly lower at lower speeds. The specifications of the engine Dynamometer are summarized in **Table 2**.

Table 2. Dynamometer specifications

Dyno. type	HORIBA /DT400-1
Max. torque	1300 N.m
Max. speed	10000 rpm
Max. power	400 kw
Inertia	0.08 kg/m ²
Weight	200 kg

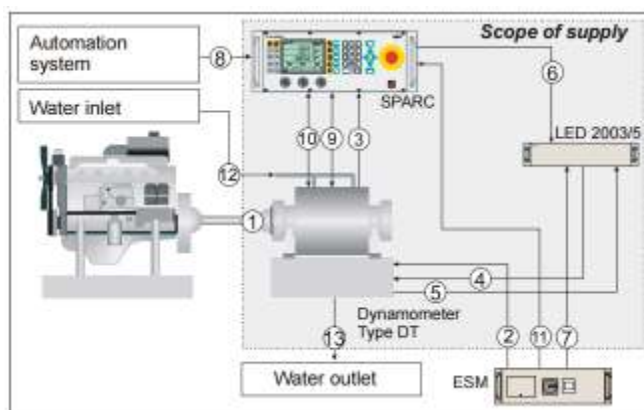
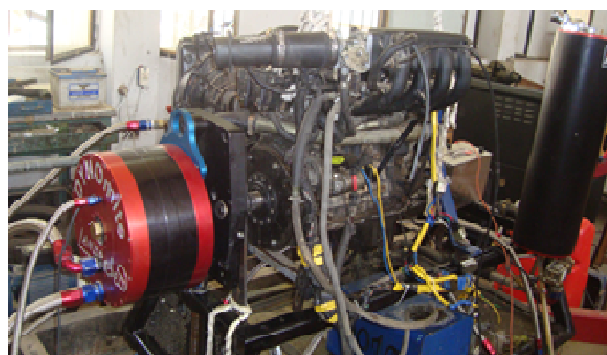
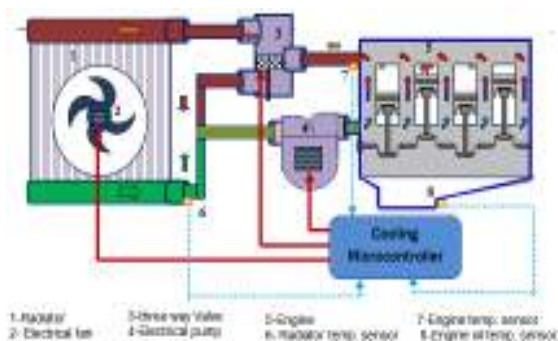


Figure 3: Schematics of revised cooling system schematic and dynamometer controller packet.

3.2 Instrumentation

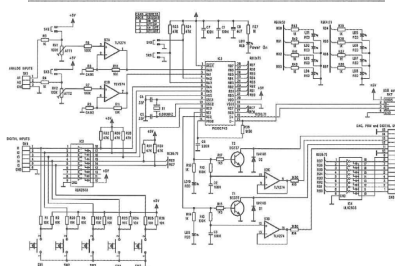
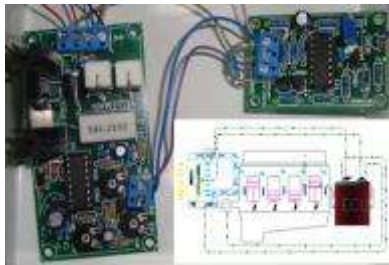
Portable version of infrared gas analyzer is used during the experimental tests. A HOMANS gas analyzer is equipped with gas sampling probe to collect the exhaust gas from the muffler. The gas is then filtered and dried before entering the analyzer, inductive speed transducer is used also to measure the engine speed in rpm. The measuring range and accuracy for measuring instruments are presented also. **Figure 4** presents an overview of the laboratory engine dynamometer and the instrumentation system, while the installation of instruments, during the exhaust emissions test continuously proportioned samples of the dilute exhaust mixture and the dilution air are collected and using sample engine exhaust for analysis. A gas analyzer is utilized to measure diluted exhaust concentrations of carbon monoxide (CO), Nitrogen oxides (NOx), total unburnt hydrocarbons (THC), and carbon

dioxide (CO₂). Gas analyser were used to measure exhaust THC, CO, CO₂, and NO_x concentrations, and the analysers have the measuring accuracy of 1ppm for THC, 0.01 per cent for CO, 0.01 per cent for CO₂, and 2ppm for NO_x.

A thermocouple is a sensor for measuring temperature. This sensor consists of two dissimilar metal wires, joined at one end, and connected to a thermocouple thermometer or other thermocouple-capable device at the other end. When properly configured, thermocouples can provide temperature measurements over wide range of temperatures. Table 3 shows the specifications of K-type thermocouple sensor. It is based on using the AD8495 thermocouple amplifier, which is designed specifically to measure K-type thermocouples. This analog solution is optimized for minimum design time: It has a straightforward signal chain and requires no software coding.

Table 3: Thermocouple sensor specifications

Thermocouple Type	Measurement Junction Temperature Range	Reference Junction Temperature Range	Accuracy at 25°C	Power Consumption	Voltage Change Seebeck Coeff.
K	-25°C to +400°C	0°C to 50°C	±3°C (A grade) ±1°C (C grade)	1.25 mW	41 (µV/°C)



(a) Electric water pump and microcontroller

(b) a variable-position smart valve

(c) layout of the gas analyzer

Figure 4: The electrical water pump with driving controller, Electronic control valve with driving controller and the measuring equipment's

4. Results and Discussions

4.1 Simulation results

A numerical simulation has been used to evaluate the performance of coolant systems by using Matlab/Simulink software package. The initial simulation conditions were set to be as follows $T_c(0)=20\text{ }^{\circ}\text{C}$ and the reference temperatures were selected to be $T_{ref} = 100 + \sin(0.06 t)$. The engine temperatures response with mechanical and electrical pumps is presented in Figure 5, the coolant temperature reached up to 95 °C in case of conventional cooling system. However, the maximum coolant temperature attained with the electric pump system is lower than the conventional pump. This result shows that the cooling system with the electric pump could operate with the downsized radiator and still satisfy the temperature limit established for the conventional system, while the flow rate with time and temperature response is shown in Figure 6.

4.2 Experimental Results

This section will present the results obtained from the tests carried on the ACC and conventional coolant systems in the laboratory.

4.2.1 Influence of fluid mass flow rate on the warm-up period

The rate of engine warm up is comparable in the two systems, though there is a slight advantage to be had with the ACC system. Reducing the minimum pump flow set by the control strategy could reduce warm-up time further, to reduce the warm-up period; zero-flow rate strategy was examined by closing the a variable-position smart valve and stopping the electrical water pump operation during the warm-up period. **Figure 7** shows an increase of the coolant temperature as a function of time. The zero-flow strategy helped the coolant temperature to increase quickly, thereby, shortening the warm-up period. It was observed that the warm-up time was shortened from 620 to 894 second, approximately by 31% with the strategy. **Figure 8** depicts the effect of coolant system type on coolant temperature. **Figure 8a** shows the different flow rate in ACC system, and fixed flow rate ate in conventional system with constant idle speed at 800rpm as shown in **Figure 8b**, but from these figures it can be seen that the coolant temperature with ACC system is equal to the desired temperature.

4.2.2 Effect of a variable-position smart valve lift on coolant temperature and flow rate

The characteristics of the two cases of controlling the cooling capacity were made, by a variable-position smart valve discharge and change in coolant pump speed. **Figure 9a** depicts the effect of a variable-position smart valve lift on coolant temperature, the coolant temperature decrease when the valve lifts increase, and flow rate increase when the valve lifts increase. The effect of adjustment performance of a variable-position smart valve lift on the intensity of the heat transfer was preceded by determining the intensity of the liquid flowing out of the engine block and flowing through the radiator as a function of the opening of a variable-position smart valve lift and fluid flow as a function of speed water pump. As a measure of valve lift was assumed from the fully open position to the closed valve position. Changes in flow rate as a function rotational speed from (600 to 3000rpm) of the water pump shown in **Figure 9b**. **Figure 10** depicts the repeated test to approximately transited state engine temperature from 58 to 95 C°, the thermal response of the engine and a radiator coolant temperature is displayed. The engine and radiator coolant temperatures were determined versus time at constant values of torque (12, 25, 45 and 75 Nm).

4.2.3 Influence of coolant temperature on the engine performance

Figure 11a shows the effect of coolant temperature on engine SFC and engine torque, the SFC was reduced when the coolant temperature increases (95:105 C°) to approximately 4.5%, and the optimal engine output torque up to (90:108 C°) coolant temperature. **Figure 11b** shows the effects of coolant temperature on exhaust emissions and fuel consumption. The most significant fact was that emissions of Total hydrocarbon (THC) and Carbon monoxide (CO) were reduced when the coolant temperature was higher than 88°C. Compared with data at 88C°, THC and CO at 105 C° were reduced by 12% and 4.5%, respectively. However, NO_x emission was observed to increase because higher coolant temperature contributed to raising combustion temperature.

4.2.3 Influence of active coolant temperature on the exhaust gas emission

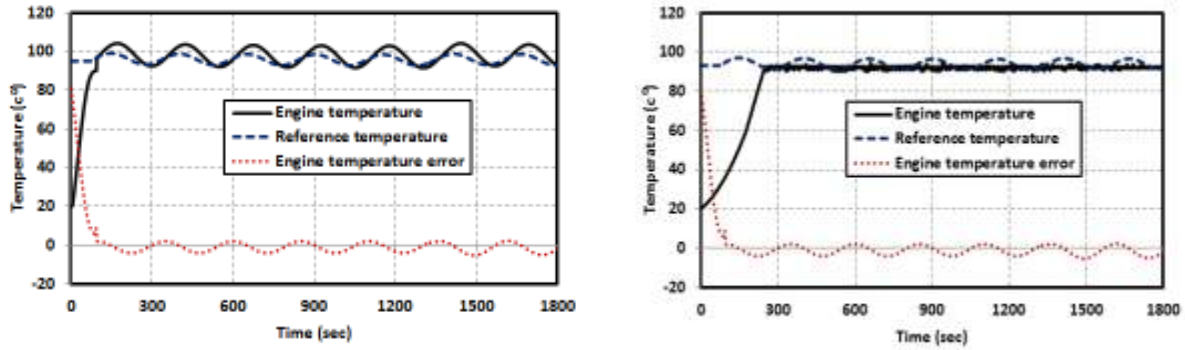
An active control of the coolant flow rate with an a variable-position smart valve enables to control the optimal engine temperature, the emission figures 14 to 17 shows with two constant values of dynamometer torque (12, and 75 Nm). **Figure 12** illustrates the trends of variation of NO_x against engine temperature, engine speed and engine load. The NO_x increases when the engine load and speed increases at engine temperature increases. **Figures 13, 14 and 15** show the effect of active cooling temperature and engine load on total THC, CO and CO₂ respectively, the emissions (THC, CO and CO₂) indicates that better to engine operate temperature over to 90C°.

Figure 16 shows the effect of coolant flow rate on exhaust emissions. To ensure engine safety, we kept coolant temperature constant at 95°C. The coolant flow rate was decreased down from 90% to 0% of active coolant flow rate. Compared with THC, CO₂ and CO emission obtained from the ACC system, THC CO₂ and CO emissions were reduced by 46.5ppm, 8.7 ppm, and 54% respectively with decreasing the coolant flow rate when 0%. However, NO_x emission was found to increase by 37.6 ppm using this strategy as it was expected. The decreases in THC and CO emissions as well as the increase in NO_x emission correspond to approximately 42 %. It is believed that the low coolant flow rate made the cylinder temperature high. Consequently, the high metal temperature enhanced fuel evaporation and increased combustion temperature of a petrol engine, while the CO emission was reduced by 58 % with decreasing the coolant flow rate when 0%. **Figure 17** depicts the compared exhaust emission between conventional and ACC system, the decreases in exhaust emission with ACC system, the emissions (NO_x, THC, CO

and CO₂) decreases with ACC system by 6.2%, 8.4%, 13.5%, and 11%, respectively. The air index (λ) is considered to represent the fuel consumption, and is defined as:

$$\lambda = \frac{(A / F)_{actual}}{(A / F)_{stoichio}} \quad (10)$$

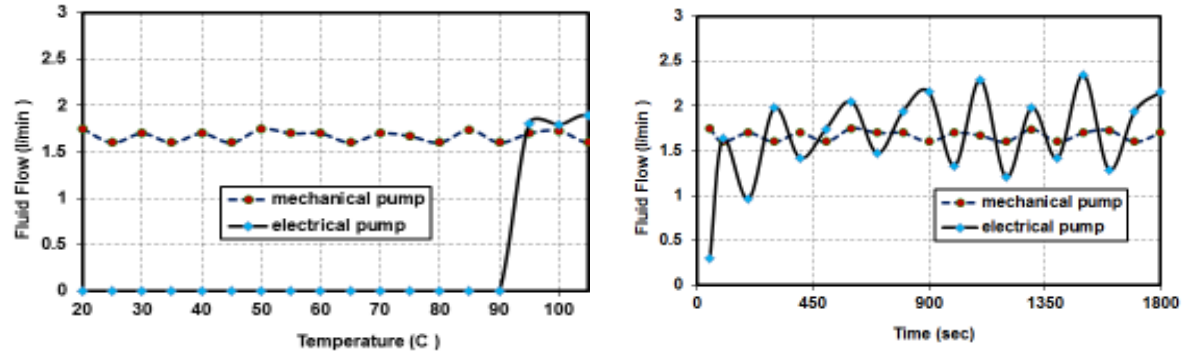
The stander ranges of λ are 1.2-0.85 between lean and rich for gasoline engine operations. It is clearly seen that the recording of the vehicle air index (λ) is almost constant. The compared air index between conventional and ACC system values vary 0.92 and 0.98 respectively, this values indicts the fuel consumption at ACC consequently better fuel consumption can be obtained.



(a) Mechanical pump

(a) Electrical pump

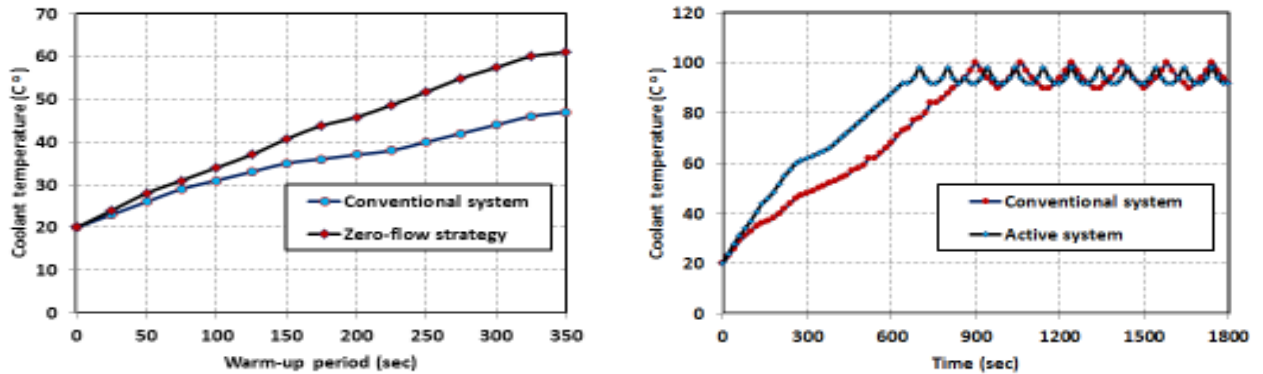
Figure 5: The coolant temperature at mechanical and electric coolant pumps



(b) Fluid flow with temperature

(c) Fluid flow with time

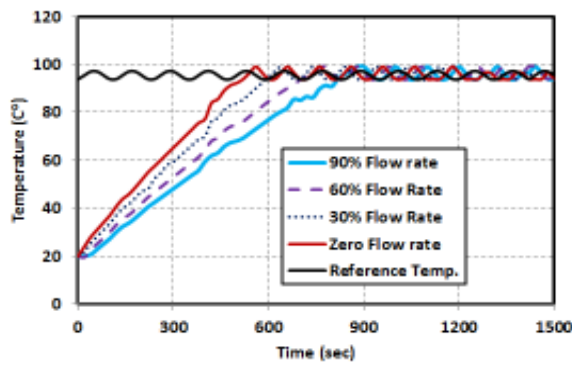
Figure 6: The coolant fluid flow at mechanical and electric coolant pumps at engine speed 2500rpm



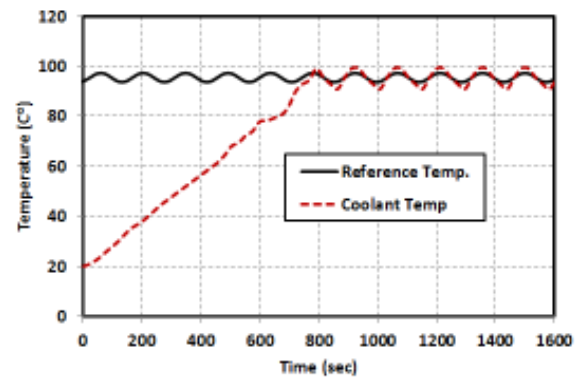
(a) Effect of the zero-flow strategy in warm-up period

(b) Coolant temperature tracking

Figure 7: Coolant temperature in different system



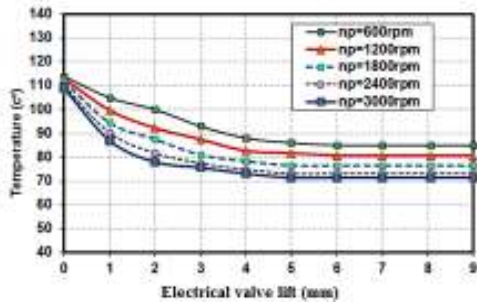
(a) Active cooling System



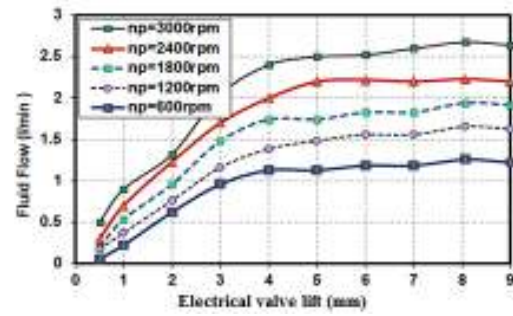
(b) Conventional cooling System

Figure 8: Effect of coolant system type on coolant temperature

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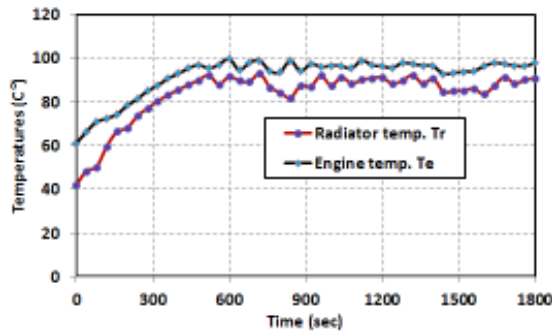


(a) Coolant temperature with different pump speed

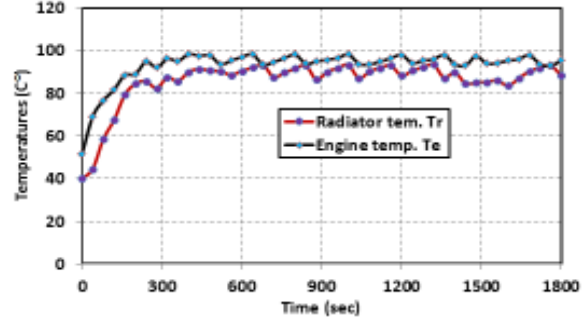


(a) Coolant flow rate with different pump speed

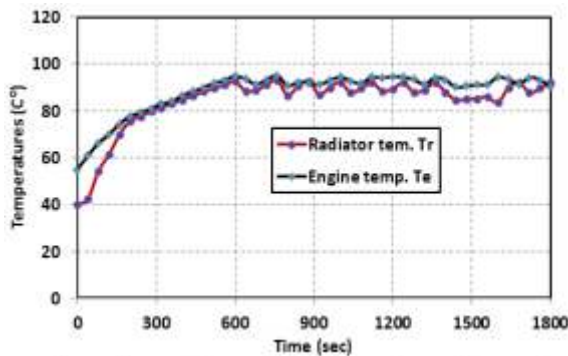
Figure 9: Effect of an electrical valve lift on coolant temperature and flow rate



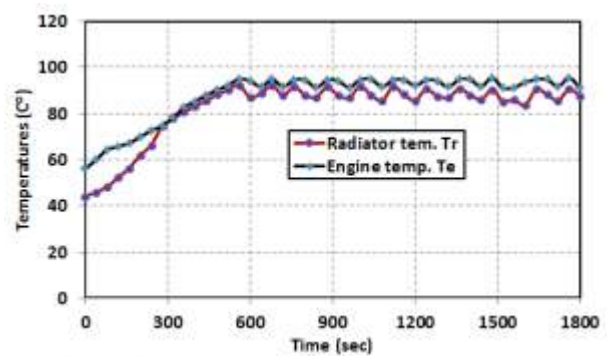
(a) Load torque=12 Nm



(b) Load torque=25Nm



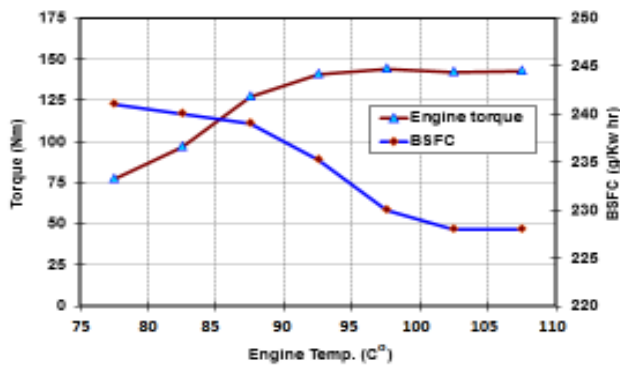
(a) Load torque=45Nm



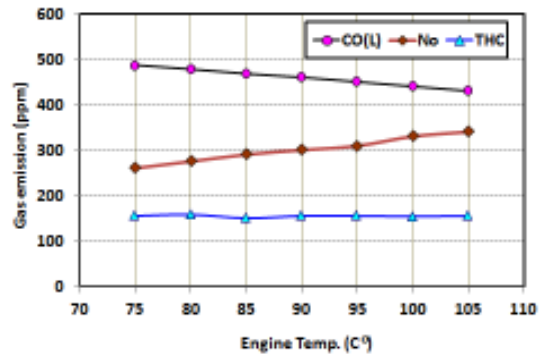
(b) Load torque=75Nm

Figure 10: Effect of engine load on radiator and engine temperature

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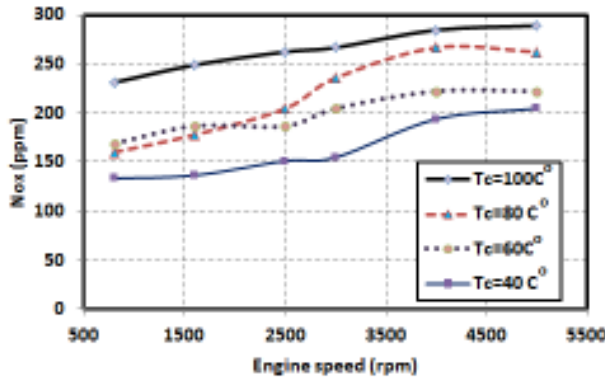


(a) Engine torque and SFC

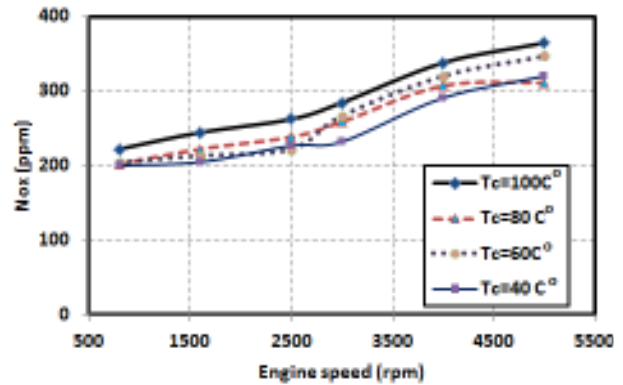


(b) Emission characteristics

Figure 11: Effect of coolant temperature on engine torque, SFC and emission characteristics

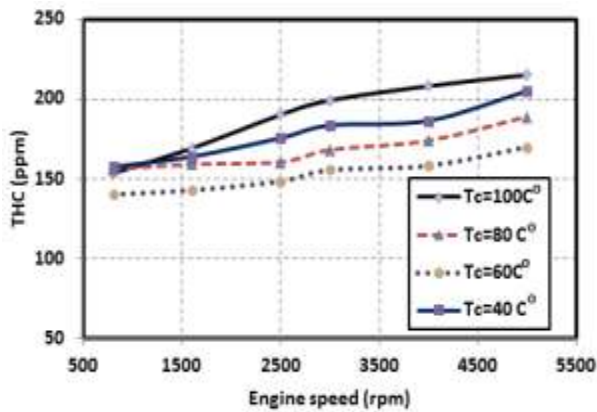


(a) Load torque=12Nm

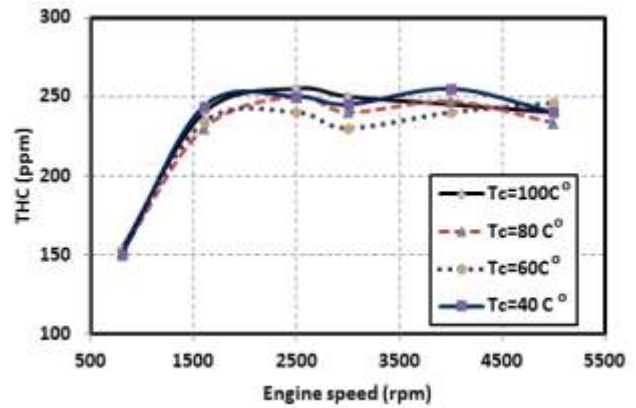


(b) Load torque=75Nm

Figure 12: Effect of active cooling temperature and engine load on nitrogen oxides (NOx)

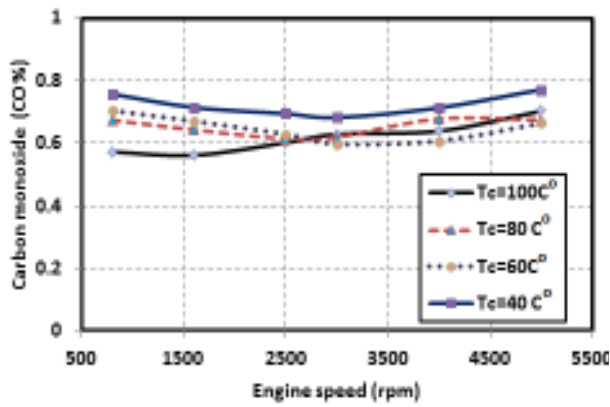


(a) Load torque=12Nm

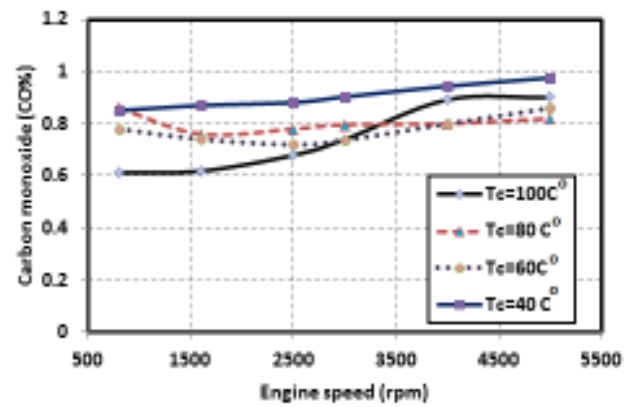


(b) Load torque=75Nm

Figure 13: Effect of active cooling temperature and engine load on total hydrocarbon(THC)

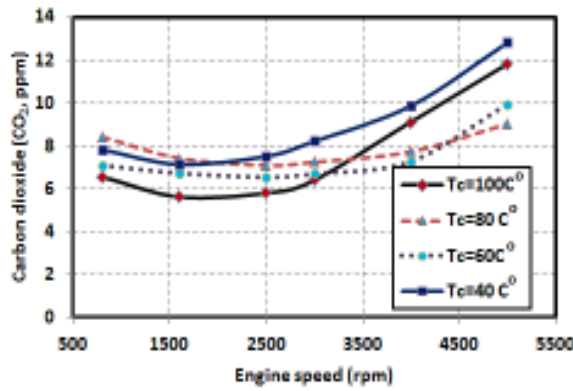


(a) Load torque=12Nm

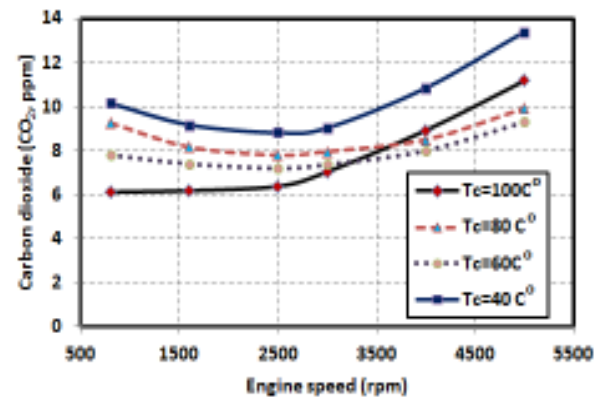


(b) Load torque=75Nm

Figure 14: Effect of active cooling temperature and engine load on carbon monoxide (CO)



(a) Load torque=12Nm



(b) Load torque=75Nm

Figure 15: Effect of active cooling temperature and engine load on carbon dioxide (CO₂)

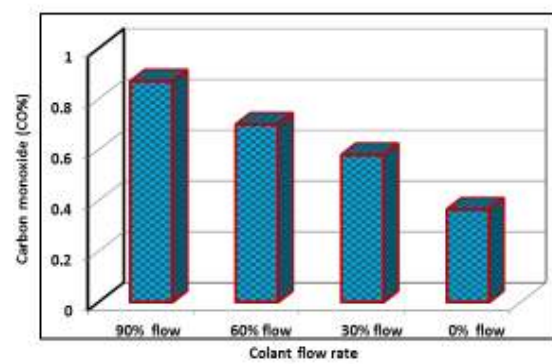
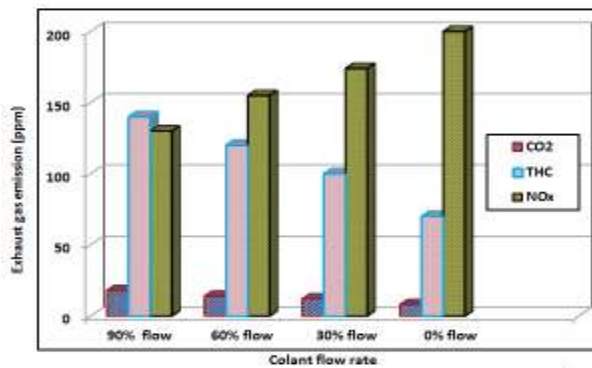


Figure 16: Comparison between coolant flow rate and average exhaust emission

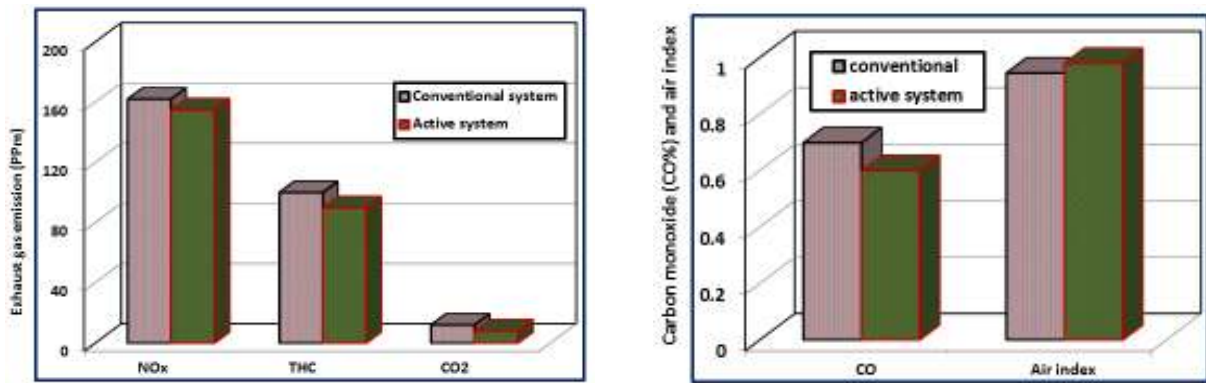


Figure 17: Comparison between coolant system type and average exhaust emission and air index

5. Conclusions

In this paper, we present engine ACC system and investigate the system configuration and experimental evaluation of ICE performance and emissions. The main conclusions from the work carried out are summarised in the following points:

1. The new type of engine cooling system was developed to improve the thermal efficiency and exhaust emissions of the internal combustion engine. This advanced engine cooling system consists of microcontroller to control the quantity of coolant fluid flow rate into the engine cylinder block. The ETM of the internal combustion engine is increased remarkably by using this type of engine ACC system. The laboratory apparatus and experimental methodology capability established in this work could be utilised for evaluating the ACC performance of the SI engine coolant system. Moreover, SI engine was instrumented and operated with ACC and equipped with engine dynamometer, exhaust gas analyzer to measure the exhaust emissions.
2. An electric water pump and a variable-position smart valve were adapted for ACC strategy, controlling the flow path also shortened the warm-up period. Significant reduction of the warm-up period was achieved by closing a variable-position smart valve between the engine and the coolant pump, the effectiveness of ACC in the condition of idle state is much better than that for transient temperature state based on the engine dynamometer and the experimental tests revealed that warm up period from ACC was decreased by 31%.
3. The coolant flow rate was decreased down from 90% to 0% of active coolant flow rate, compared with THC, CO₂ and CO emission obtained from the ACC system, THC CO₂ and CO emissions were reduced by 46.5ppm, 8.7 ppm, and 54% respectively with decreasing the coolant flow rate when 0%. However, NOx emission was found to increase by 37.6 ppm using this strategy as it was expected. The decreases in THC, CO₂ and CO emissions as well as the increase in NOx emission correspond to approximately 42 %. It is believed that the low coolant flow rate made the cylinder temperature high.
4. Compared between conventional and ACC coolant system, at partial load conditions of engine dynamometer, the results indicate that most of the CO, CO₂, and THC emission appears at higher load as well as near the coolant temperature. NOx emission was observed to increases because higher coolant temperature contributed to raising combustion temperature. The assessment results of the experimental tests revealed that CO₂, CO, THC, and NOx, of SI engine with ACC were reduced by 11%, 13.5%, 8.4%, 6.2%, and 26.0%, respectively.

References

- [1] **D. J. Allen and M. P. Lasecki**, "Thermal Management Evolution and Controlled Coolant Flow", SAE Paper 2001-01-1732, 2001.
- [2] **Robert D. Chalgren Jr.**, Thermal Comfort and Engine Warm-Up Optimization of a Low-Flow Advanced Thermal Management System", SAE Paper 2004-01-0047.
- [3] **John R. Wagner, Venkat Srinivasan and Darren M. Dawson**, "Smart Thermostat and Coolant Pump Control for Engine Thermal Management Systems", SAE technical paper 2003-01-0272, 2003.
- [4] **Wagner, J. R., Paradis, L., Marotta, E. E., and Dawson, D.**, "Enhanced Automotive Engine Cooling systems – A Mechatronics Approach", International Journal of Vehicle Design, vol. 28, no. 1/2/3, pp. 214-240, 2002.
- [5] **P. Setlur, J.R. Wagner, D.M. Dawson and E. Marotta** "An Advanced Engine Thermal Management System: Nonlinear Control and Test" IEEE/ASME Transactions on Mechatronics, Vol. 10, Issue 2, 2005.

- [6] **P. Setlur, J. Wagner, D. Dawson and J. Chen**, “Nonlinear Controller for Automotive Thermal Management Systems” Proceedings of the American Control Conference, Denver, Colorado, June 4-6, 2003.
- [7] **M. H. Salah, T. H. Mitchell, J. R. Wagner and D. M. Dawson**, “Nonlinear-Control Strategy for Advanced Vehicle Thermal-Management Systems, IEEE Transactions on Vehicular Technology, Vol. 57, No. 1, 2008.
- [8] **N. Staunton, V. Pickert and R. Maughan**, “Assessment of Advanced Thermal Management Systems for Micro- Hybrid Trucks and Heavy Duty Diesel Vehicles” IEEE Vehicle Power and Propulsion Conference (VPPC), Harbin, China, 3-5, 2008.
- [9] **N. S. Ap, N. C. Golm** “New Concept of Engine Cooling System (Newcool) ”, SAE International Congress, SAE Paper 911775, Detroit, 1991.
- [10] **J. Kern, P. Ambros** “Concepts for a Controlled Optimized Vehicle Engine Cooling System, SAE International Congress, SAE Paper 971816, Detroit, 1997.
- [11] **A. Choukroun, M. Chanfreau** “Automatic control of electronic actuators for an optimized engine cooling thermal management” in: SAE Vehicle Thermal Management Systems Conference & Exposition, Nashville, Tennessee, USA, 2001.
- [12] **J.R. Wagner, V. Srinivasan, D.M. Dawson, E.E. Marotta**, “Smart Thermostat, Coolant Pump, Control for engine thermal management systems” in: SAE World Congress & Exhibition, Detroit, Michigan, USA, 2003.
- [13] **D.H. Lim, S.C. Kim, M.S. Kim**, “Thermal analysis of an electric water pump for internal combustion engine vehicles”, Int. J. Aut. Technol. 14 (2013) 579-585, 2013.
- [14] **R. Ernst, R. Friedfeldt, S. Lamb, D. Lloyd-Thomas, P. Philips, R. Russell, T. Zenner**, “The new 3 cylinder 1.0L gasoline direct injection turbo engine from Ford”, in: 20th Aachen Colloquium Automobile and Engine Technology, Aachen, Germany, 2011.
- [15] **Kim, M-S**. "Dynamometer Tests of Brake Shoes under Wet Conditions for the High Speed Trains" International Journal of System Applications, Engineering and Technology, Issue 2, Volume 5, 2011.
- [16] **Vicente F, Marina K M, Leonidas N, Stefan H and Panagiota D** “Road Vehicle Emission Factors Development: A Review” Journal of Atmospheric Environment, Vol.70 (2013), pp. 84-97, 2013.
- [17] **Choukroun, A., and Chanfreau, M.**, “Automatic Control of Electric Actuators for an Optimized Engine Cooling Thermal Management,” SAE paper no. 2001-01-1758, 2001.
- [18] **Mohammad H. S., Tom H. M, John R. W, and Darren M. D**, “Nonlinear-Control Strategy for Advanced Vehicle Thermal-Management Systems” IEEE Transactions on Vehicular Technology, Vol. 57, NO. 1, pp. 127-137, 2008.

Appendix I

<i>Notation:-</i>	<i>Abbreviations:-</i>
$\dot{m}_c$:Coolant mass flow rate	ACC :Active coolant control
C_p :Specific heat capacity of the coolant	A/F :Air fuel ratio
T_{e1} :emperature at engine inlet	BLDC :Brushless Direct Current
T_{e2} :Temperature at engine outlet	CO :Carbon monoxide
ρ_c :Coolant fluid density	CO ₂ :Carbon dioxide
$\dot{V}_c$:Coolant volume flow rate	ETM :Engine thermal management
$\dot{m}_r$:The coolant mass flow	ICE :Internal Combustion Engine
$\dot{m}_{py}$:The coolant mass flow in the by-pass circuit.	NOx :Nitrogen oxides
P : Pump power consumption	PWM :pulse width modulation
η : Pump efficiency	SFC :Specific fuel consumption
Q_p : Volumetric flow rate	SI :Spark ignition
Δp : Pressure rise across the pump	THC :Total hydrocarbons
T_a :Cooling air temperature	λ :Air index
T_{e1} :Inlet temperature at engine	
T_{e1} :Inlet temperature at engine	
T_{e2} :Outlet temperature at engine	
T_m :Mean actual engine temperature	
V_{ep} : Engine coolant pump applied voltage.	
V_{rf} : Radiator fan applied voltage.	