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RESEARCH ARTICLE

An Experimental Approach of Smart Engine Coolant Thermal Management Strategy During Engine Warmup Period

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HIGHLIGHTS

> Coolant flow rate control strategy was investigated with the help of an electric water pump on the engine cooling system at variable engine speed conditions.
> The effects of fuel energy fractions on each other were investigated by energy balance analysis.
> The percentage of conversion of fuel energy into useful work, engine warm-up time, brake specific fuel consumption and CO emission production were improved thanks to the application of a 50% level of coolant flow rate at different loads.

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ABSTRACT

Various thermal energy management strategies and energy efficiency researches are carried out in internal combustion engines. For this purpose, in this study, a kind of thermal energy management strategy regarding the application of coolant flow rate control in the engine cooling system has been investigated. In the thermal energy management strategy approach, the experiments were carried out on a gasoline-fueled spark-ignition test engine under different engine load and continuously variable speed conditions. An electric pump is integrated into the system, which can be switched on as needed to ensure a controlled flow of coolant. A total of four different configurations were tried on the engine cooling system, including a mechanical pump and an electric water pump integrated.6.2% and 5.34% improved. In addition, energy balance and engine performance characteristics analyzes were made during the warm-up phase of the engine and it was seen that the EPICS strategy had positive effects on engine efficiency and CO exhaust emissions. With the EPICS strategy, the specific fuel consumption under different loads improved by 7.72% and 5.2%, respectively. Thus, the benefits of the coolant control strategy applied in variable speed operating conditions are examined in detail.

Contents

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1. Introduction

Examining thermal energy management systems is of great importance in terms of energy efficiency. In this respect, it is very important to improve thermal efficiency for internal combustion engines, which is one of the subtle areas in energy efficiency. In fact, mitigating the thermal losses and friction losses from the engine components is a central approach for increasing the efficiency of the engine. In this case, the temperatures of the engine block and other cooling system components must be adjusted during the warm-up period of the engine. As such, the coolant temperature, which is a very effective parameter in the cooling system, has an important role in improving thermal efficiency and reducing thermal losses and friction losses [1].

The thermal state of the engine should be thoroughly checked to reduce the amount of fuel consumption, especially when the engine is running under part load conditions. The design, positioning and control of cooling system components directly affect fuel consumption and efficiency. Therefore, different engine cooling system strategies are compared with various methods developed to calculate the theoretical minimum fuel consumption [2]. However, the effects of heat fluxes and temperature adjustments in the engine cooling system. It can be studied by developing numerical models with the help of coolant flow rate control [3].

In the thermal energy management approach, an electric pump is used in the cooling circuit to control the coolant flow rate. In addition, there are various applications where the thermostat is used in the engine lubrication line circuit. The use of an electric water pump in the cooling system is an effective method on engine temperature control. With this method, the engine temperature can be controlled effectively and it can make significant contributions to the improvement of exhaust emission values and hence engine life [4]. It is important to evaluate the performance and reflex of wax type thermostats, which are effective cooling system components, in different engine operating conditions. With the thermostat used in the cooling circuit, effective results can be obtained on engine cooling efficiency and engine performance by providing benefits on engine warm-up time and fuel consumption [5].

The beneficial effects of the use of components such as controllable thermostat, electric water pump and electric fan in the engine cooling system circuit on engine fuel consumption and pollutant emissions can be modeled through experimental data. The model outputs have shown that an improvement of 1.1% can be obtained in fuel economy, 5.3% and 6.1% in hydrocarbon and carbon monoxide emissions, respectively, can be achieved in a driving cycle such as New European Driving Cycle. These results prove that the use of controllable components on the cooling system can significantly contribute to engine efficiency [6]. Structural changes or applications, such as optimum coolant flow control, have direct effect on thermal energy losses in engines and on shortening engine warm-up time. Providing a coolant flow rate of 30% in the cooling circuit of an engine operating under part engine load conditions minimizes friction and heat transfer loss and increases thermal efficiency [7]. However, the temperature of engine coolant and engine components directly trigger engine knocking tendency. Therefore, the relationship between unburned fuel temperature and the dynamic response of the engine walls with load and coolant flow should not be overlooked. The risk of knocking can be reduced by controlling the engine metal temperature under constant and transient load conditions and determining appropriate cooling control strategies, accompanied by coolant control. Thus, in the event that a knock-free engine starts to knock, the starting time of the engine can be delayed by at least one minute by applying a controlled coolant flow rate. With this application, engine torque and effective efficiency can be improved by at least 3% [8].

Electric water pump has been used before in the cooling system circuit of engines [9]. While the engine coolant control provides different energy gains in the entire engine system, the electric water pump, also acts as a system controller with regards to the engine endurance, pollutant emissions, fuel consumption and overall performance of the vehicle. Thanks to this important task, the problems related to the mechanical pump can be remediated by using an electronically controlled electric pump and control components in the classical cooling system [10–12]. Actually, by reducing the rotational speed of a mechanical pump, which circulates the fluid in the engine cooling system at a higher flow rate than required for cooling, by 65%, significant benefits were achieved in fuel consumption and engine warm-up time [13]. By obtaining variable coolant flow rates with an electric pump operated at various rotational speeds, their effects on fuel economy and reliability were investigated under different engine loads, and engine's thermal balance experiments were carried out to design and build smart cooling system control strategies [14].

The effects of coolant and intake air temperature on thermal efficiency have been tried to be understood through one-dimensional engine simulations using software, such as GT-Suite. The brake thermal efficiency was estimated quantitatively by varying the coolant temperature under different engine speed and loads. Controlling the coolant and intake air temperature was found to improve thermal efficiency [15].

In the conventional cooling system, where the coolant flow rate is proportional to the engine speed, overcooling or insufficient cooling conditions are experienced. In this case, the optimum coolant flow rate can be applied to shorten the engine warm-up period, taking into account the risk of boiling. With the implementation of the cooling plan, significant controls were established on the flow of the coolant, and low pumping power as well as a smaller, optimum radiator size were achieved. As a result of this approach, improvements in fuel consumption, hydrocarbon emission rate and pumping power by 2.1%, 8.6% and 44.3%, respectively were recorded [16].

An important method in the approach of energy management strategy in engines is the analysis of the fuel energy entering the engine control volume. With this analysis, the conversion rate of fuel energy into useful work (engine efficiency), exhaust gases, coolant and unaccounted lost energy rate are determined. Energy balance and sometimes exergy analyzes are used to define more precisely the effects of cooling system components. It is understood that these analyzes were done to examine the effects of fossil fuels with nanoparticle additions, of various alternative fuels and their mixtures on engine performance, engine thermal efficiency and exhaust emissions [17–19]. In different studies, the intake air conditions can be determined with the help of mathematical models based on thermodynamic concepts [20]. The engine performance and environmental effects of the fuel type and exhaust gases were analyzed numerically and experimentally [21].

Thermal energy management is essentially an effective approach to improve fuel economy and reduce exhaust emissions. Further optimization of thermal energy management in internal combustion engines requires a detailed analysis of the energy flow in each of its components. As a result of making the necessary improvements with these analyzes, it was stated that the engine efficiency would be more than 38% at different loads and rotational speeds. In addition, it has been reported that the energy fraction lost to the coolant can remain above 50% at low powers and below 30% at high powers. These results showed that the optimization of coolant and lubricating oil offers significant benefits in terms of emissions and fuel consumption [22].

Studies on energy balance in engines reported in the literature were carried out mostly under constant engine speed and load conditions [23], which facilitates engine tests and measurements. However, internal combustion engines mostly operate in unpredictable variable speed and load conditions. Therefore, it is very important to examine the energy efficiency and performance analysis of an engine operating under these conditions. Experimental studies of this kind with variable conditions are limited in different operating conditions.

In this study, the test engine was operated continuously at a variable speed of 2000-3000 rpm at 15 second intervals and at 25% to 40% engine loads during the warm-up period. In the first part of the experiments, the coolant flow rate in the cooling circuit was provided proportional to the engine speed by the mechanical pump. In the second part of the experiments, electric pump was used in the cooling circuit, the cooling liquid obtained by the mechanical pump was circulated with a controlled 50% reduction in flow rate. After real-time experimental measurements, energy balance and engine performance analyzes were made.

2. Experimental System and Components

Experiments have been carried out on a spark-ignition Ford MVH418 engine installed on a hydraulic dynamometer bench. The experimental system, electronic modules, measurement devices, and mechanical components is shown schematically in Figure 1 and the technical specifications of the test engine are given in Table 1.

K type thermocouples (0.5 °C accuracy) were used on the experimental system to measure the temperatures of engine inlet air, coolant, ambient air, exhaust manifold, exhaust gas and cooling water. A fuel measure device and an intake air flow sensor were used on the experimental system to calculate the fuel consumption and intake air flow rate entering the cylinders. The technical specifications of the fuel flow meter and an intake air flow sensor are given in Table 2.

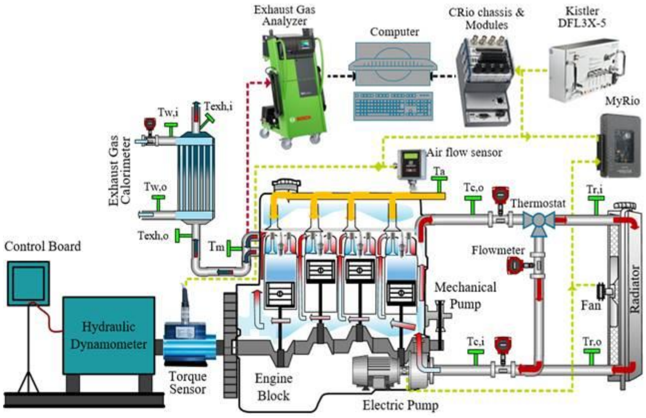


Figure 1 A schematic view of the experimental system

Table 1 The technical specifications of engine.

Descriptions	Value
Engine model	Ford MVH418
Engine type	Spark ignition, fuel injected
Stroke number	4 stroke
Cooling system	Water cooled engine
Cylinder type	In line – 4 (DOHC-16V)
Firing order	1-3-4-2
Stroke/ Diameter	88 mm / 80.6 mm
Total displacement	1796 cm^3
Compression ratio	10:1
Maximum power	93 kW @ 6250 rpm
Maximum torque	157 Nm @ 4500 rpm
Idle speed	900 ± 50 rpm
Fuel	Gasoline
Fuel injection system	BOSCH KE-Jetronik

A Bosch BEA-250 exhaust emission device was used for measuring %volume CO emission (0.001 accuracy). The engine is connected to a hydraulic dynamometer to apply various brake loads at the engine. The Kistler-4503A torque sensor (±0.1 Nm accuracy) was placed on the engine output shaft for torque measurements. The torque sensor signals are obtained by processing over the FPGA-based chassis connected to the AD-Combo analog input module. All experimental measurements were obtained in the Labview environment. Labview software works with same logic as the FPGA data flow structure. Therefore, all modules used in the measurement of variable parameters were integrated on the CompactRIO chassis, which can be programmed with Labview and contains FPGA integrated circuits. A myRIO device was used to control the engine coolant flow rate with an electric centrifugal water pump with a capacity of 6000 L/h.

The flow rate of the coolant was measured with a turbine type flowmeter (3% accuracy) operating according to the hall effect principle. In the experiments, the throttle was electronically controlled so as to allow the engine work stably at the desired speed conditions. Therefore, a throttle control program was added to the Labview program prepared for recording experimental measurements. The throttle opening ratio is entered into the program manually by the user. The engine controller starts to follow the operating map of the variable speed test according to the variable throttle opening position entered in the program. The experiments for each test are conducted at variable engine speed (2000-3000 rpm) and two different engine loads (25% and 40%).

Table 2 Technical specifications of the fuel flow meter and intake air flow sensor.

Fuel flow meter	
Fuel flow meter type	Cysts DFL3X-5bar
Maximum pump flow	120 L / h @ 5 bar
Measurement accuracy	± %0.5 @ 1–50 L/h
Connection interface	CAN/USB/RS-232C
Intake air flow sensor	
Air flow sensor	Sierra 628S
Gas type	Ambient air
Power	18-30 VDC
Output signal	0-5 VDC linear
Display type	LCD digital display
Response time	200 ms
Accuracy	±1%

3. Methodology of Configurations

In this study, two different configurations (a) and (b) were constructed: (a) mechanical pump integrated cooling system (MPICS) configuration and (b) Electric water pump integrated cooling system (EPICS) configuration which was designed to reduce the coolant flow rate by 50%. The MPICS configuration can be described as a conventional engine cooling system. In this system, the mechanical pump, which takes its movement from the crankshaft with the help of a belt pulley, circulates the coolant in the engine cooling circuit. Depending on the engine speed, the mechanical pump circulates the coolant at an excessively high flow rate, especially during the engine warm-up period, and causes the engine warm-up time to prolong. Since the engine coolant has such an effective role on engine performance, fuel consumption and energy efficiency, an exemplary application has been made for engine coolant flow rate control. In this context, firstly MPICS Variable speed engine experiments were carried out on the configuration. In these experiments, the engine was operated at 2000 rpm for 15 s and the engine coolant flow rate was 13 L/min with a mechanical pump. During the following 15 seconds, the throttle opening ratio was increased and the engine was operated at 3000 rpm and the engine coolant flow rate with the mechanical pump was measured as 21 L/min. In order to change the engine speed in a controlled way, an electronic control system program prepared in Labview software was used and the engine throttle opening ratio was controlled electronically. Thanks to the designed controller, the engine was operated continuously at 2000 rpm for 15 seconds from the first start and then at 3000 rpm for 15 seconds, and this cycle was repeated until the engine reached stable operating conditions. Then, in the MPICS configuration, the belt pulley components were removed and the mechanical pump was separated from the system, replacing it with an electric water pump. In the MPICS experiment, the coolant flow rate values measured at 2000 and 3000 rpm revolutions were reduced by 50% in the EPICS experiment and thus circulated in the cooling circuit. Engine coolant flow rate in EPICS experiments were 6.5 L/min at 2000 rpm and 10.5 L/min at 3000 rpm. All of the experimental tests were started when the test setup was at an ambient temperature of 16-17˚C. Here, in contrast to the studies carried out under constant operating conditions, variable effects on engine performance, energy distribution, engine efficiency and CO exhaust emissions were observed during the entire warm-up period when the engine speed was variable.

4. Energy Distribution of Internal Combustion Engines

The energy terms, input and output to/from the control volume of an internal combustion engine are schematically shown in Figure 2. The instantaneous energy balance analysis during warm-up period in internal combustion engines can be performed with Eq. (1).

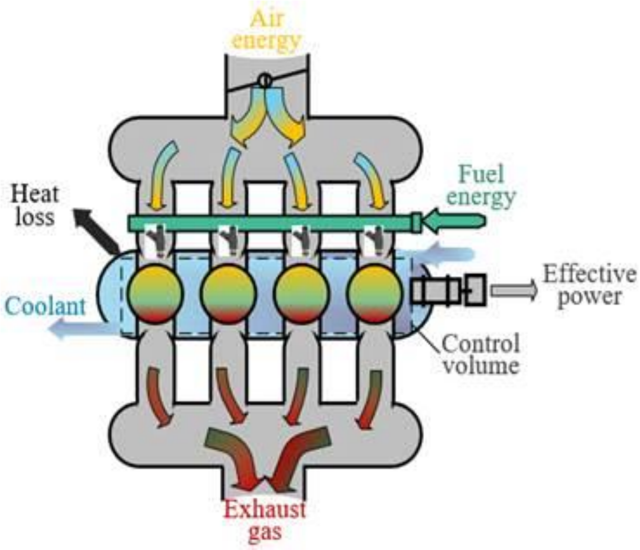


Figure 2 Input and output energy terms for control volume of the engine.

$$\dot{E}n_f + \dot{E}n_a = P_e + \dot{E}n_c + \dot{E}n_{exh} + \dot{E}n_{loss} \quad (1)$$

The energy of the fuel "$\dot{E}n_f$ (kW)" and the energy of inlet air "$\dot{E}n_a$ (kW)" are calculated using Eqs. (2) and (3), respectively. A part of the fuel energy is converted into useful work by the engine crankshaft. This useful work is known as effective power "P_e (kW)" and is calculated by Eq. (4).

$$\dot{E}n_f = \dot{m}_f H_u \quad (2)$$

$$\dot{E}n_a = \dot{m}_a h_a \quad (3)$$

$$P_e = \frac{Tn}{9549.3} \quad (4)$$

A part of the thermal energy produced as a result of the combustion of the fuel-air mixture is expelled to the environment from the cylinders through the exhaust gases. An exhaust gas calorimeter is integrated into the experimental setup to calculate the thermal energy lost through the exhaust gases. Exhaust gas calorimeter consists of three parts: the thermal energy lost via the exhaust manifold, the thermal energy stored in the calorimeter cooling water, and the heat energy expelled to the environment from the calorimeter. The exhaust gas energy loss rate "$\dot{E}n_{exh}$ (kW)" is determined by Eq. (5).

$$\dot{E}n_{exh} = \dot{m}_{exh}C_{exh}(T_m - T_{exh,i}) + \dot{m}_{exh}C_{exh}(T_{exh,i} - T_{exh,o}) + \dot{m}_{exh}C_{exh}(T_{exh,o} - T_a) \quad (5)$$

The thermal energy stored in the shell and tube exhaust gas calorimeter cooling water is expressed by Eq. (6).

$$\dot{m}_{exh}C_{exh}(T_{exh,i} - T_{exh,o}) = \dot{m}_w C_w(T_{w,o} - T_{w,i}) \quad (6)$$

Eqs. (5) and (6) are rearranged, exhaust gas energy loss rate is computed in Eq. (7).

$$\dot{E}n_{exh} = \left[\frac{\dot{m}_w C_w(T_{w,o} - T_{w,i})}{(T_{exh,i} - T_{exh,o})} \cdot (T_m - T_a)\right] \quad (7)$$

The heat loss rate to the coolant "$\dot{E}n_c$ (kW)" is calculated with Eq. (8),

$$\dot{E}n_c = \dot{m}_c C_c(T_{c,o} - T_{c,i}) \quad (8)$$

The energy efficiency of the system "η (%)", is determined with Eq. (9),

$$\eta = \left(\frac{P_e}{\dot{E}n_f + \dot{E}n_a}\right) \cdot 100 \quad (9)$$

A part of the energy loss is transferred from the engine block to the environment, a part of it is lost to friction and the rest is expelled unburn out of the exhaust as a result of incomplete combustion. All these, as a whole, are defined as the unaccounted energy loss rate "$\dot{E}n_{loss}$ (kW)" in the energy balance analysis. $\dot{E}n_{loss}$ is expressed as shown in Eq. (10).

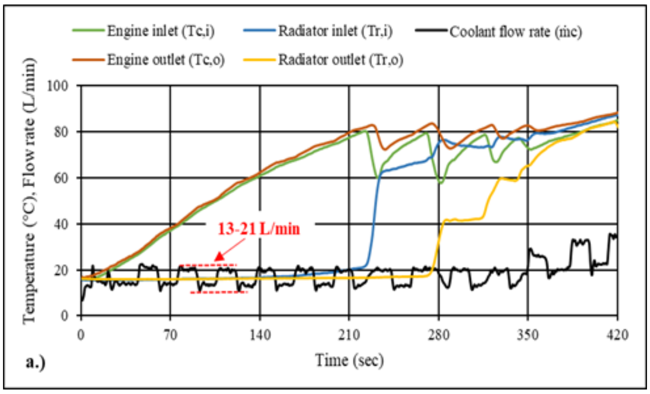
$$\dot{E}n_{loss} = \dot{E}n_f - P_e - \dot{E}n_c - \dot{E}n_{exh} \quad (10)$$

During the warm-up period, the engine block, coolant, and lubricating oil start store thermal energy. Therefore, the temperature of engine parts and coolant changes over time in parallel. The instantaneous energy balance for the engine cooling circuit in the control volume is expressed as shown in Eq. (11) [2].

$$\dot{E}n_{wall,c} - \dot{E}n_{eb} + \dot{m}_c C_c(T_{c,i} - T_{c,o}) = m_c C_c \frac{dT_{c,o}}{dt} \quad (11)$$

5. Results and Discussion

Under variable engine speed (2000-3000 rpm) and 25% engine load, instantaneous changes of the temperature and flow rate of the engine coolant taken from different points in the mechanical pump integrated cooling system (a-MPICS) and electric pump integrated cooling system (b-EPICS) are shown in Figure 3. Here, the graphs regarding the coolant temperature and flow rate change in the experiments with 40% load are not given graphically, but they are discussed in detail below together with the results of the experiments with 25% load.



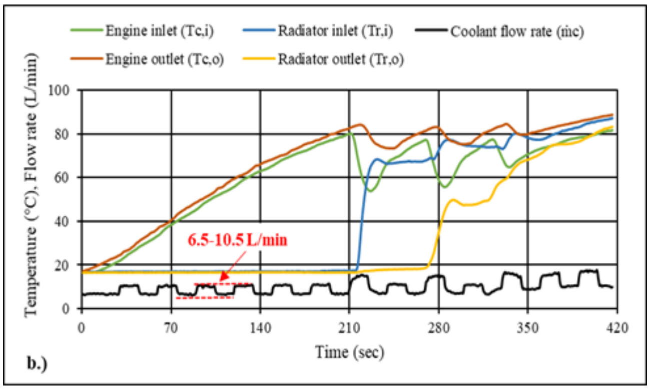


Figure 3 Instantaneous variations of coolant temperature and coolant flow rate at 2000-3000 rpm / 25% load, [a.) MPICS - b.) EPICS].

As seen in Figure 3, in the MPICS configuration under low engine load the coolant was circulated in the cooling circuit at 13 L/min, the lower limit, and 21 L/min, the upper limit, at 15s intervals. The instantaneous change of engine inlet-outlet and radiator inlet-outlet temperatures starting from the start of the engine and during the warm-up period is shown in Figure 3. It is clearly seen from Figure 3 that the engine warm-up time was measured as 226s with the MPICS configuration at 25% engine load. Then, when the coolant temperature reached around 83°C, the thermostat opened and the engine coolant was directed to the radiator line. Thus, after a few short thermostat on-off cycles, the thermostat stayed open and the engine was tried to be kept within the stable operating temperature range.

Coolant flow rate values obtained from MPICS experiments were given to the controller set up in Labview, and the cooling water flow rate was reduced by 50% with the electric pump, and in the EPICS configuration, the engine coolant flow rate was circulated in the cooling circuit at 15s intervals between the lower limit 6.5 L/min and the upper limit 10.5 L/min. In this case, the engine warm-up time at low load (25%) was calculated as 212s. The engine warm-up period, which is very important in 2000-3000 rpm conditions, where urban usage is intense and most of the engine operating time is realized, especially in cold climate regions, is controlled by the coolant. It showed a significant improvement of 6.2%. Engine warm-up times of all experiments carried out at 2000-3000 rpm / 25% and 40% load conditions are given in Figure 4 in detail.

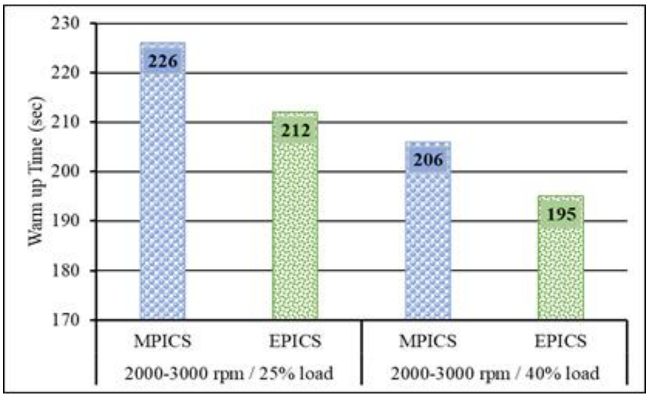


Figure 4 Variations of warm up time of all experiments.

Thanks to the EPICS configuration under 40% load conditions, the warm up period has decreased from 206s to 195s. Thus, an improvement of 5.34% was achieved in the engine warm-up time. When attention is paid, the thermostat started to open in a shorter time under 40% load conditions. This shortening can be attributed to the high increase in energy entering the thermodynamic system together with the coolant flow rate and high engine load.

With the strategy of reducing the coolant flow rate by 50%, positive effects were seen on the average brake specific fuel consumption (BSFC) obtained during the engine warm-up period. The BSFC values of EPICS and MPICS configurations are shown in Figure 5. With the strategy, the BSFC value in EPICS configuration at 25% load conditions decreased from 579.8 g/kWh to 535 g/kWh. In this case, the BSFC value decreased by 7.72% during the warm-up period of the engine compared to the MPICS strategy thanks to the coolant flow control strategy. Similarly, while the specific fuel consumption was 423 g/kWh in the 40% loaded MPICS experiments, this value decreased to 401 g/kWh with the EPICS strategy. Thus, the BSFC improved by 5.2% during the warm-up period. As a result, significant improvements were achieved with the strategies applied for both load conditions.

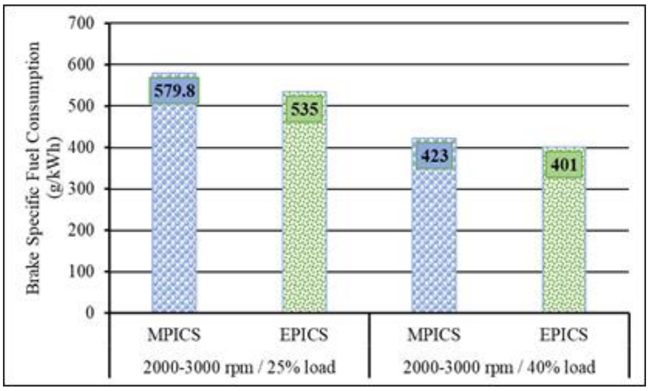


Figure 5 The BSFC values during the warm up period.

Depending on the length of the engine warm-up period, the contents of the exhaust emission components are significantly affected as a result of incomplete combustion products formed in greater amounts in this process. In this study, the effects of the coolant control strategy on CO, an important exhaust emission component, during the engine warm-up period were investigated and the emission rates are given in Table 3. While the average CO emission was 0.726% by volume under 25% load conditions, it decreased to 0.698% by volume with the EPICS strategy, and an improvement of 3.74% was achieved. Similarly, the effectiveness of the EPICS strategy was observed in the experiments performed under 40% load conditions, and the CO emission level decreased from 0.831%vol to 0.766vol%, providing an improvement of 7.82%.

Table 3 The CO emission production rate during the warm up period.

Average CO emission production rate (% Volume)			
2000-3000 rpm	25% load	MPICS	0.726
	40% load	EPICS	0.698
	25% load	MPICS	0.831
	40% load	EPICS	0.766

In Figure 6, instantaneous energy balance changes obtained from MPICS and EPICS configuration experiments at 2000-3000 rpm / 25% load test condition are given. When Figure 6 is examined, the conversion rate of fuel energy obtained from the engine output shaft into useful work during the warm-up period (P_e) started to increase in both configurations. Energy lost to coolant ($\dot{Q}_{coolant}$) and exhaust gases ($\dot{Q}_{exhaust}$) over the percentage of useful work rates began to increase gradually. In Figure 6, after 226s in the MPICS strategy, the thermostat started to open and the rate of increase in the energy lost to the coolant began to increase. The change of the main energy fractions resulting from the combustion of fuel energy with 100% unit energy is shown instantaneously in Figure 6. In the EPICS configuration, where the coolant is reduced by 50%, the energy dissipation rates are relatively lower. As the variable engine speed causes sudden changes on some parameters such as engine torque, coolant flow and fuel consumption, energy distribution changes exhibited a fluctuating behavior as expected.

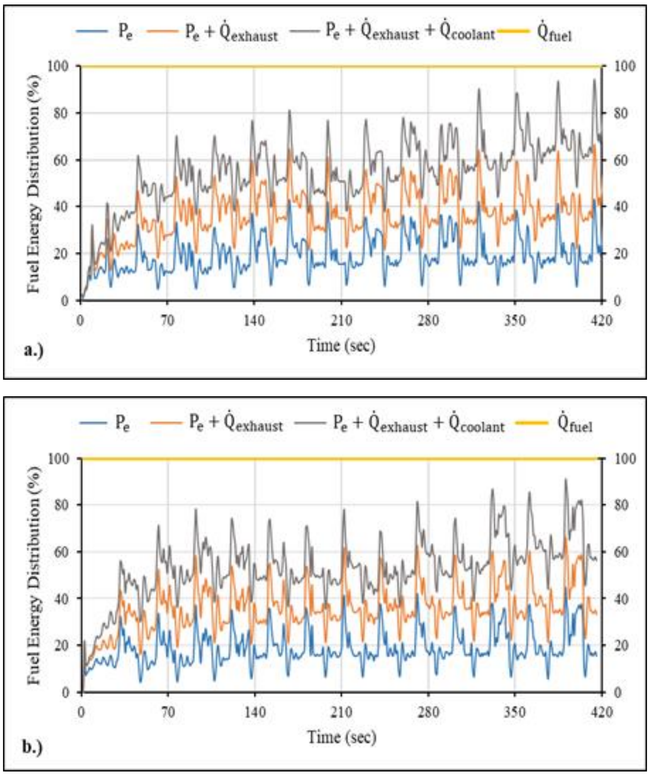


Figure 6 Instantaneous energy distributions of fuel energy at 2000-3000 rpm / 25% load, [a.) MPICS - b.) EPICS].

The percentages of thermal energy balance versus coolant flow rate for various operating conditions are shown in Figure 7. Fuel energy is the sum of effective power, the energy lost through the exhaust gases, the energy lost to the coolant and the unaccounted energy loss. While the effective power, which is the conversion rate of fuel energy to useful work, was 17.70% during the engine warm-up period with the EPICS strategy at 25% load, this value became 17.49% in the MPICS strategy. Thus, when the results of the energy balance analysis performed within the scope of the EPICS configuration are examined, the effective efficiency has improved by 1.2% compared to the MPICS configuration. 14.8% of fuel energy in MPICS strategy at 25% load (P_e) separated as the ratio of energy lost to the coolant. This value was 15.95% with the EPICS strategy and increased by 7.2%. With the EPICS strategy, the percentage of energy lost through exhaust gases decreased by 4.78% and the percentage of thermal losses decreased by 1.11%.

On the other hand, when the results of the energy balance analysis performed within the scope of EPICS configuration under 40% load test condition are examined, the rate of conversion of fuel energy into useful work, in other words, the effective efficiency has improved by 3.2% compared to the MPICS configuration. Here, with the EPICS configuration, the percentage of fuel energy lost through exhaust gases increased by 2.2%, the percentage of energy lost to the coolant increased by 1.35% and the other unaccounted percentage of lost energy decreased by 2.6%. The unaccounted energy loss rate showed a more significant change at 40% load conditions and corresponded to 47.33% of the fuel energy. As a result, while the energy lost due to the limited time for heat transfer at high engine load and the percentage of energy loss through exhaust gases decreased, the percentage of useful work and energy lost to the coolant increased.

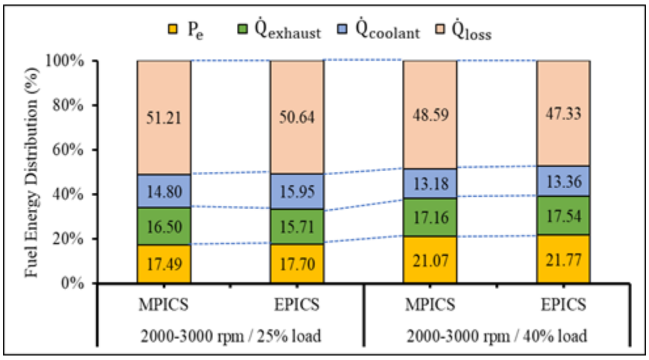


Figure 7 A summary of the energy balance during the warm up period at all experimental configurations.

6. Conclusion

In this study, the events occurring during the warm up period of a spark-ignition engine under transient conditions were investigated. Coolant flow rate control application was carried out in the engine cooling circuit with MPICS and EPICS configurations designed at 2000-3000 rpm and at 25% and 40% engine load conditions. The general results obtained for the experimental study were as listed below:

1. Thanks to the EPICS strategy during the warm-up period, the engine block reached its stable operating temperature in a shorter time and without the risk of boiling, by circulating the coolant in the cooling circuit with 50% reduced flow rate. Thus, the engine warm-up time is shortened at all engine loads.
2. By controlling the coolant flow rate with the electric pump during the warm-up period, the BSFC improved by 7.72% and 5.2%, respectively, under low and high engine load conditions.
3. Since the increase in engine load increases the velocity of energy entering the thermodynamic system during the warm-up period, CO emission values increased under 40% loaded conditions. However, during the engine warm-up period, improvement effects were observed on the CO emission production rate with the 50% level coolant flow rate strategy.
4. Energy balance analyzes were made and it was seen that the coolant flow rate is a very effective parameter in the warm-up period. Friction and heat transfer rate lost from the engine block during the warm-up period cause low engine efficiency and high energy loss. As the steady state was approached, the engine efficiency started to increase and the lost energy ratio started to decrease gradually. In low and high load experiments where 50% coolant flow rate was applied, the total unaccounted lost energy rates decreased as a result of the shortening of the warm-up time.
5. During the warm-up period, there was no significant change in the effective efficiency of the engine due to thermodynamic irreversibility in the combustion process, but the percentage of conversion of fuel energy into useful work with 50% coolant flow rate in the EPICS configuration strategy was 1.2% and 3.2% at low and high load, respectively. In addition, the increase in engine load contributed to the increase in the percentage of improvement in thermal efficiency.

As a result, different studies with new designs can be carried out under different operating conditions within the scope of coolant control strategies.

Declaration of Conflict of Interest

The authors declare no conflict of interest.

Nomenclature

CO	Carbon monoxide (% vol.)
EPICS	Electric Water Pump Integrated Cooling System
FPGA	Field-Programmable Gate Arrays
MPICS	Mechanical Pump Integrated Cooling System
C_c	Specific heat capacity of coolant (kj/kgk)
C_{exh}	Specific heat capacity of exhaust gases (kj/kgk)
C_w	Specific heat capacity of calorimeter cooling water (kj/kgk)
$\dot{E}n_a$	Inlet air energy (kw)
$\dot{E}n_c$	Energy lost by coolant (kw)
$\dot{E}n_{eb}$	Energy transfer rate from the coolant to the engine block (kw)
$\dot{E}n_{exh}$	Energy lost by exhaust gases (kw)
$\dot{E}n_f$	Fuel energy (kw)
$\dot{E}n_{loss}$	Unaccounted energy loss rate (kw)
$\dot{E}n_{wall,c}$	Cylinder wall to coolant energy transfer Rate (kw)

h_a	Enthalpy of air (kj/kg)
H_u	Low heating value of fuel (kj/kg)
m_c	Total mass of engine coolant (kg)
n	Engine speed (rpm)
$\dot{m}_a$	Mass flow rate of air (kg/s)
$\dot{m}_c$	Coolant mass flow rate (kg/s)
$\dot{m}_{exh}$	Exhaust gas mass flow rate (kg/s)
$\dot{m}_f$	Fuel consumption (kg/s)
$\dot{m}_w$	Mass flow rate of calorimeter water (kg/s)
P_e	Effective power (kw)
rpm	Revolutions per min
T	Torque (nm)
T_a	Ambient temperature (k)
$T_{c,i}$	Coolant temperature at engine inlet (k)
$T_{c,o}$	Coolant temperature at engine outlet (k)
$T_{exh,i}$	Exhaust gas temperature at calorimeter inlet (k)
$T_{exh,o}$	Exhaust gas temperature at calorimeter outlet (k)
T_m	Exhaust manifold temperature (k)
$T_{r,i}$	Coolant temperature at radiator inlet (k)
$T_{r,o}$	Coolant temperature at radiator outlet (k)
$T_{w,i}$	Cooling water temperature at calorimeter inlet (k)
$T_{w,o}$	Cooling water temperature at calorimeter outlet (k)
η	Energy efficiency of the system (%)

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