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CONTROL STRATEGY OF ELECTRIC COOLANT PUMPS FOR FUEL ECONOMY IMPROVEMENT

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ABSTRACT−The engine cooling system for a medium duty V6, 4.5 L diesel engine was modeled with a commercial code, GT-Cool in order to investigate the effect of controllable electric pump on the cooling performance and the fuel economy. The simulation results of the cooling system model with mechanical coolant pump were validated with experimental data. Two different types of electric pumps were implemented into the cooling system model and PID control for electric pump operation was incorporated into the simulation study. Based on the simulation result with electric pump, conventional thermostat hysteresis was modified to reduce pump operation for additional improvement of fuel economy, and then the benefit of electric pumps with modified thermostat hysteresis on fuel economy was demonstrated with the simulation. The predicted result indicates that the cooling system with electric pump and modified thermostat hysteresis can reduce pump power consumption by more than 99 % during the FTP 74 driving cycle.

KEY WORDS : Engine cooling system, GT-Cool, Fuel economy, Controllable electric pump, Modified thermostat hysteresis

1. INTRODUCTION

As the available advanced mechatronics technology for automotive applications increases, significant portion of vehicle components are replaced with electrified parts in order to improve the engine performance and the vehicle safety. However, the automotive cooling systems are still similar to those over 50 years ago (Visnic *et al.*, 2001). For instance, mechanical coolant pump is driven by the engine with belts or gears and the pump speed is in proportion to only engine speed. Therefore coolant flow rate is determined by the engine speed, which is not an optimal control in most cases. In consequence, conventional engine-driven coolant pump could cause unnecessary parasitic losses. For this reason, the need for higher fuel economy through the reduction of parasitic losses of mechanical pump has forced to introduce controllable electric pump to cooling system (Hnatczuk *et al.*, 2000).

The feasibility of electric pump for advanced cooling system has been discussed for several years. Problems implementing the electric pump have involved a lack of electric power and pump designs. Recent industry movement towards higher voltage systems and advanced hybrid powertrains will make it feasible to add electric pump on advanced cooling system for improving fuel economy (Allen *et al.*, 2001).

The research results on advanced cooling systems have been published in several papers over the last several years, which were focused on control scheme based on simulation study. Xu *et al.* (1984) developed, simulated, and tested a control scheme for various cooling components of a heavy duty diesel. In this work, a computer controlled fan clutch, variable speed coolant pump, and a modified thermostat were introduced to control scheme. Melzer *et al.* (1999) proposed the potential of a demand-responsive cooling system and specified the requirements of the cooling system and its components. They then built the advanced cooling system to show the fuel economy benefit of the system experimentally. Hnatczuk *et al.* (2000) introduced servo-motor driven coolant pumps in order to improve thermal efficiency, reduce engine parasitic losses, and control engine temperature or emissions. Cortona *et al.* (2000) developed the thermal behavior model of cooling circuit and the control method for electric pump and electrical valve. A servo motor valve and pump were installed into the cooling system to regulate the coolant flow with engine control unit by Wagner *et al.* (2002, 2003). In their works the coolant flow rig tests were performed to study these dual actuators. They used an electric heater, not a real engine, to simulate the heat rejection from the engine.

Previous researches were aimed at the development of

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cooling system model and control scheme for electric pump and valve. However, they controlled servo-motor driven electric valve by monitoring coolant mass flow rate. The electric valve causes additional power consumption and cost. Moreover, it is difficult to measure and control coolant mass flow rate in practical use. In addition, they were conducted to validate model with rig tests or research engine tests on the simulated simple engine condition.

Thus, the objective of this work is to conduct simulation studies in order to evaluate the impact of the controllable electric pump on the cooling performance and the fuel economy for vehicle driving schedule without electric valve. At first the conventional cooling system model for a V6 diesel engine vehicle was configured using commercial program, GT-Cool and then it was validated with experimental data. The cooling system model was modified to implement two different types of electric pumps into this system, and then conventional thermostat hysteresis was modified to reduce the electric pump operation based on the simulation result with electric pump. Finally, the effect of electric pumps with modified thermostat hysteresis on fuel economy improvement was investigated during the FTP 74 driving cycle.

2. MODELING

V6, 4.5 L diesel engine with its cooling system was used to configure the cooling system model, which is shown in Figure 1.

The cooling system model was built with the commercial program, GT-Cool from Gamma Technologies. GT-Cool is based on one-dimensional fluid dynamics, representing the flow and heat transfer in the pipe and the other components of a cooling system, and it features a module-based code that provides flexible model building facility.

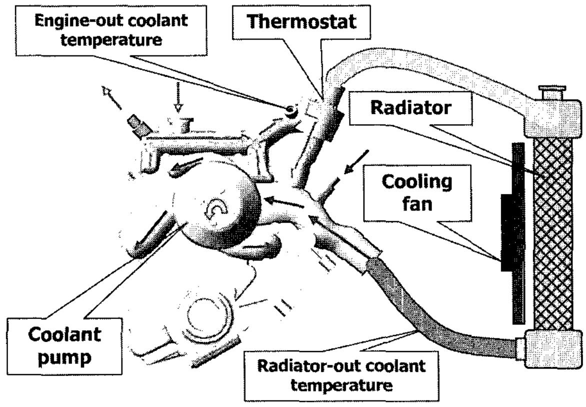


Figure 1. Schematic diagram of V6, 4.5 L diesel engine with its cooling system.

Complete V6 diesel engine cooling system model is presented in Figure 2. Various modules in GT-Cool library were configured in order to simulate cooling components. Components and modeling approaches for V6 diesel engine and each cooling component are as follows.

2.1. Engine

The engine is modeled with a look-up table type module.

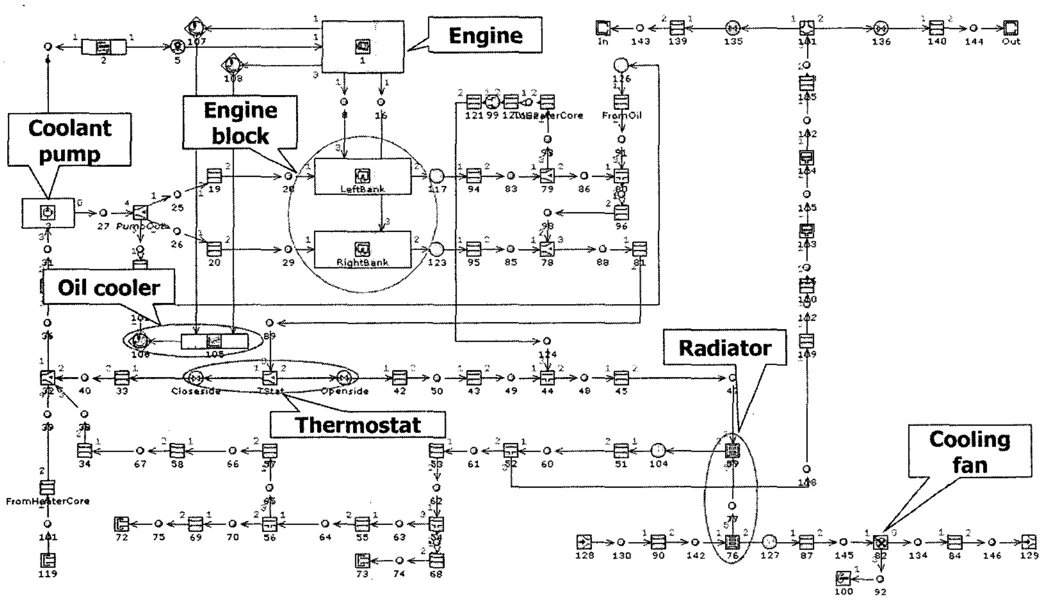


Figure 2. Complete V6 cooling system model.

Engine heat rejection rate and BSFC data as a function of engine speed and load (BMEP) are provided by a user input file. In this work, the engine performance maps were obtained from the engine dynamometer test. In order to simulate a driving schedule, engine operating points including engine speed and load can be specified as a function of time by another user input file.

2.2. Engine Block

Lumped thermal mass method was used to calculate the heat transfer to the environment. In this model, the engine block was assumed to have uniform wall temperature that was calculated from the balance of heat rejection by the engine, heat transfer to the coolant, and heat transfer to the ambient air.

2.3. Coolant Pump

Pump performance maps were used in order to calculate pump power consumption. The map data of mechanical and electric pumps were provided by International Truck and Engine Co. and Engineered Machined Products Inc. (EMP), respectively. Pump performance maps consist of pressure rise, flow rate, pump speeds, and efficiencies. Based on these data, the pump power consumption was calculated using the following equation.

$$P = \eta \cdot Q \cdot \Delta P \tag{1}$$

where

P : pump power consumption (kW)

η : pump efficiency

Q : volumetric flow rate (m^3/s)

ΔP : pressure rise across the pump (kPa)

2.4. Oil Cooler

Heat addition model was used to simulate oil cooler, which means that the heat from oil is added directly to the coolant. Heat addition rate from the oil cooler to coolant depends on engine speed and load. The heat addition rate in full load condition was measured at each engine speed, and the heat addition rate in part load condition was calculated by interpolating the full load data.

2.5. Radiator

Two modules, as shown in Figure 2, were used to model heat exchange between two fluid circuits, i.e. coolant and air. The heat transfer coefficients were calculated from the separate Nusselt number.

$$Nu = a\mathrm{Re}^b\mathrm{Pr}^{1/3} \tag{2}$$

where $Nu = \left(\frac{hL}{k}\right)$, $\mathrm{Re} = \left(\frac{\rho UL}{\mu}\right)$, $\mathrm{Pr} = \left(\frac{\mu C_P}{k}\right)$

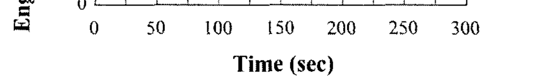
The constants a and b were extracted from steady-state experimental performance data.

3. MODEL VALIDATION

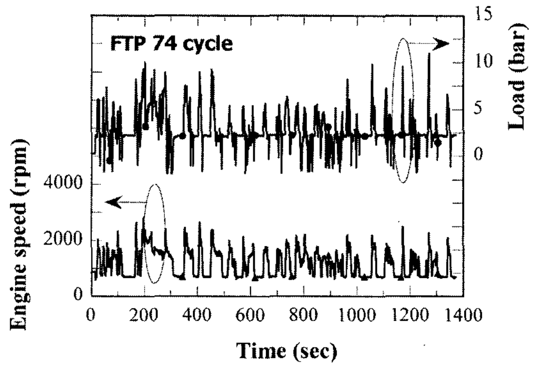
Experimental data of V6, 4.5 L diesel engine with conventional mechanical pump provided by Ford Motor Co. was used to validate the cooling system model. The heat rejection rate from engine block to coolant was measured, which was used as the input data for the engine model. The steady state experiments were conducted in order to specify the heat transfer coefficient of the radiator. Coolant temperatures were measured at three points, such as before and after the thermostat and the radiator exit.

The experimental conditions for the validation were the warm-up condition and the FTP 74 driving cycle, which are shown in Figure 3. The cooling fan was operated at a constant speed of 3000 rpm in both cases.

The cooling system model was validated with engine experimental data taken during the warm-up period shown in Figure 3 (a). Figure 4 shows the comparison of predicted and measured coolant temperature during the warm-up period. As shown in this figure, the predicted coolant temperatures are very well matched with the experimental data. As the second step, coolant temper-



(a) Warm-up condition



(b) FTP 74 driving cycle

Figure 3. Driving conditions for simulation.

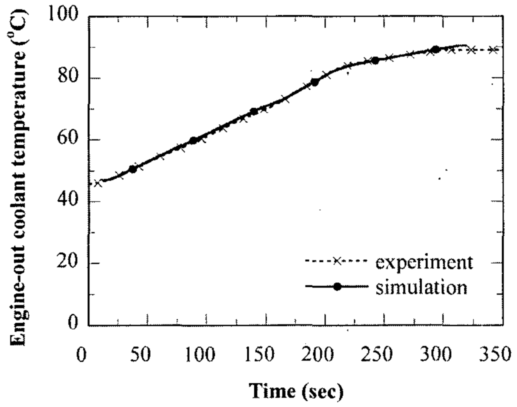


Figure 4. Comparison of predicted coolant temperature with experimental data during the warm-up period.

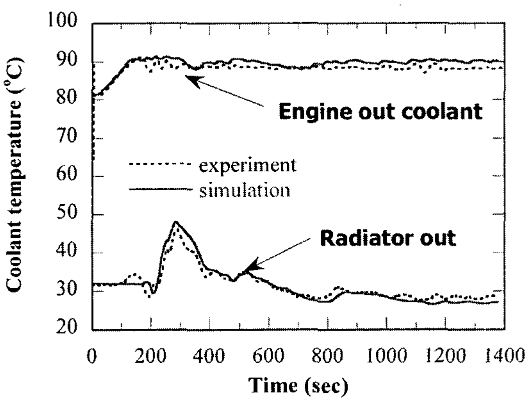


Figure 5. Comparison of predictions and measurements during the FTP 74 driving cycle.

ature variation during the FTP 74 driving cycle was calculated and compared with the experimental data, which is shown in Figure 5. The simulation result shows good agreement with the experimental data. This result indicates that this cooling system model is capable of simulating coolant temperature behavior during transient engine condition and evaluating the cooling system performance.

Figure 6 shows the predicted coolant mass flow rate to the pump and radiator. As can be seen in Figure 6, coolant flow to the pump inlet is much higher than to the radiator. Under normal driving conditions at moderate ambient temperature, which was represented by the FTP 74 driving cycle, most of the coolant is recirculated inside the engine and coolant flow to the radiator is limited by the thermostat to prevent over-cooling until the engine coolant temperature becomes higher than the thermostat opening temperature. It is because the cooling system is usually designed to prevent over-heating under extremely severe conditions. This phenomenon indicates that the

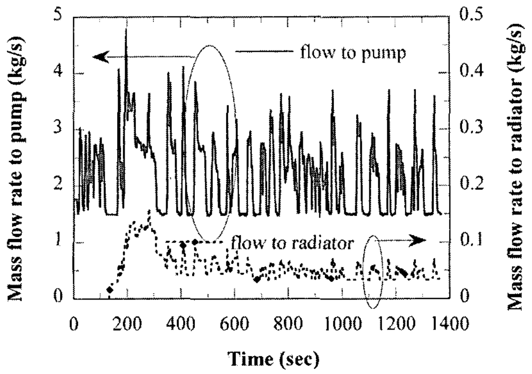


Figure 6. Predicted coolant mass flows during the FTP 74 driving cycle.

majority of the power consumed by the pump is wasted to recirculate the coolant inside the engine instead of cooling the engine. Hence, a new cooling device, such as controllable electric pump, has the significant potential to save the power consumption of coolant pump.

4. ELECTRIC COOLANT PUMP

The application of electric coolant pump enables to reduce the power consumption by optimizing coolant flow control. Basically, 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 restricts the hydraulic efficiency. However, an electric pump is directly driven by electric servo-motor, which can eliminate external elements such as belt pulley and offer the flexibility of pump configuration design to enhance the hydraulic efficiency (Allen *et al.*, 2001).

Two different types of electric pump data were used to investigate the effect of electric pump on power saving. Figure 7 shows the comparison of total efficiency at the maximum pump speed between mechanical and electric pumps. Detail number was left out due to confidentiality issue. The advantages of EMP electric pump can be easily found in this figure. Overall efficiency of both electric pumps is much higher than that of mechanical pump over all flow ranges. The cooling system model was modified to simulate electric pump. The PID controller was used to control the coolant pump speed for an active control of pump speed.

Figure 8 shows the FTP 74 driving cycle simulating result using EMP electric coolant pumps. The minimum pump speed was set at 200 rpm arbitrarily for both electric pump cases. The engine-out coolant temperature from the experimental data, was used as the target signal for PID control. The simulation result has good

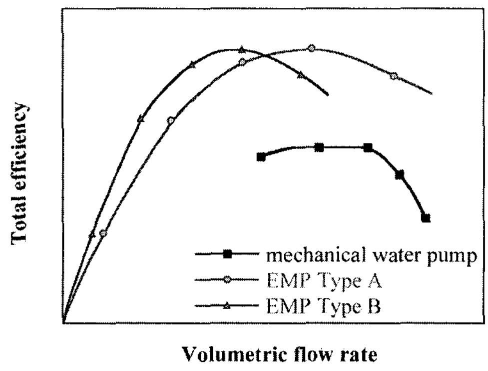


Figure 7. Comparison of mechanical and electric pump performances.

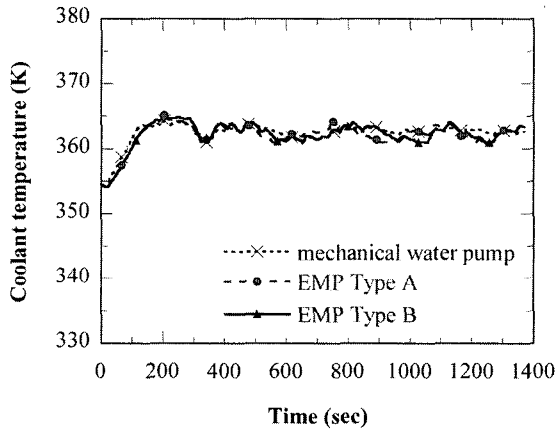


Figure 8. Comparison of coolant temperature at the exit of engine block between mechanical and electric pumps during the FTP 74 driving cycle.

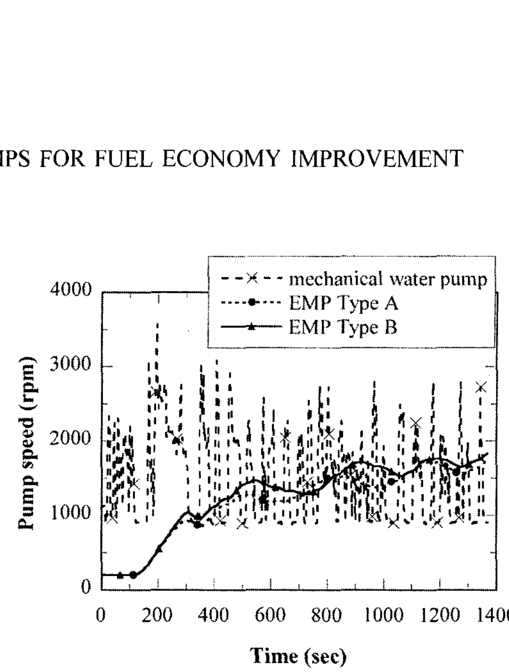


Figure 9. Comparison of pump speed between mechanical and electric pumps during the FTP 74 driving cycle.

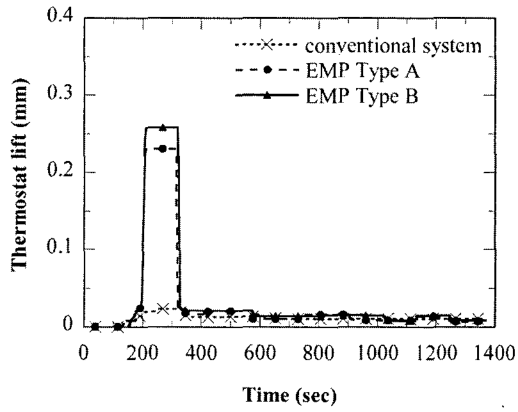


Figure 10. Comparison of thermostat lift behavior between conventional and electric cooling systems during the FTP 74 driving cycle.

agreement with target signal. Up to 200 seconds the coolant temperature gradually increases, which is warm-up period, and then it converges to about 365 K.

Comparisons of coolant pump speed during the FTP 74 driving cycle is presented in Figure 9. The mechanical pump was operated in proportion to engine speed regardless of coolant temperature, which is the reason of over-power consumption. In cases of electric pumps, they were operated at the minimum speed during warm-up period, and then pump speed gradually increases to match the target signal.

However, the electric pump speed increased to match the target signal in spite of convergence of engine-out coolant temperature as shown in Figure 8. This is caused by very small thermostat lift. The FTP 74 driving cycle consists of various part load conditions, which are mild conditions for cooling system. Hence, as shown in Figure 10, the maximum thermostat lift was less than 0.3 mm, which is about 1.7% of the total lift. The total thermostat lift is 12 mm. Because of small thermostat lift, most of coolant flow cannot go through the radiator and the radiator is not effectively used to cool down the engine. For this reason, the PID controller makes the pump speed increase to cool down the engine.

Figure 11 shows the coolant mass flow to pump and radiator during the FTP 74 driving cycle. Even though coolant flow by electric pumps is smaller than mechanical pump case, most of the coolant is still recirculated inside the engine, which shows the possibility of additional power saving.

The coolant flow determined by initial electric pump speed is recirculated inside engine without flowing toward radiator in early stage of engine operation. As the engine is operating, coolant temperature increases and thermostat is gradually expanded in order to make the coolant flow to radiator. After thermostat opening, the coolant flow is divided into two directions which are the recirculating flow in the engine and the flow to radiator. In other words, the coolant flow to pump is the sum of the

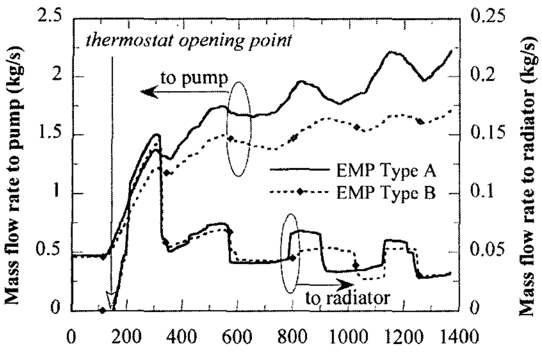


Figure 11. Predicted coolant mass flow with electric pumps during the FTP 74 driving cycle.

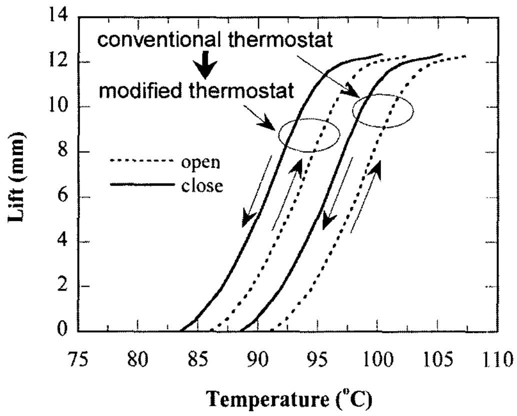


Figure 12. Modification of thermostat hysteresis.

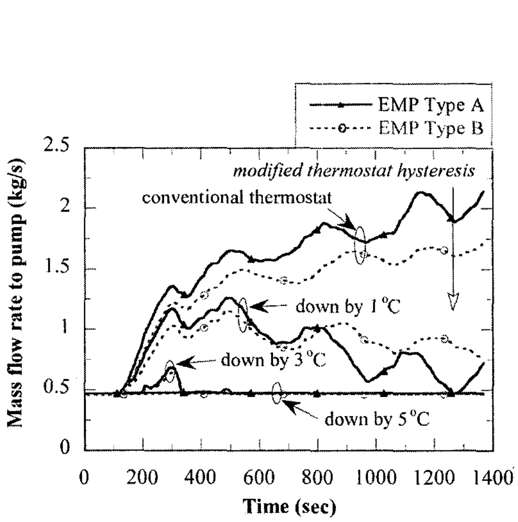


Figure 13. Predicted coolant mass flow to electric pumps with the modified thermostant hysteresis.

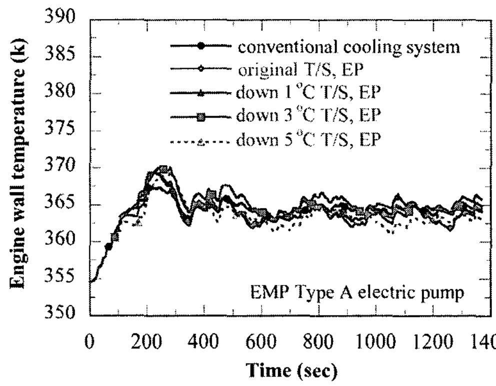


Figure 14. The effect of thermostat hysteresis on the engine wall temperature.

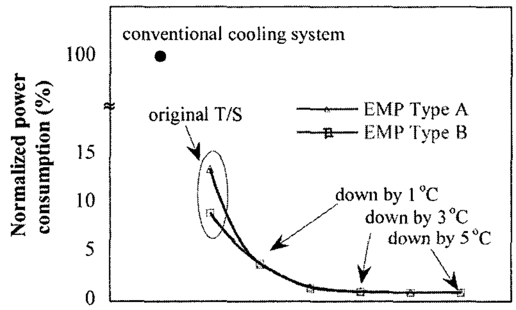


Figure 15. Comparison of pump power consumption during the FTP 74 driving cycle.

recirculationg flow and the flow to radiator. Therefore, if the radiator is effectively used by modifying thermostat hysteresis, the reciuculating coolant flow can be reduced and additional power saving can be achieved.

The effect of thermostat hysteresis on coolant flow rate to pump was investigated during the FTP 74 driving cycle. As can be seen in Figure 12, opening/closing temperature of thermostat was modified without any change of hysteresis curve shape. The coolant mass flow to pump was predicted with the modified thermostat hysteresis which was moved down by 1, 3, and 5°C respectively, which is shown in Figure 13. The coolant mass flow to pump is reduced until the thermostat hysteresis is moved down by 5°C, and then it maintains the initial coolant flow during the FTP 74 driving cycle.

Figrue 14 shows the effect of thermostat hysteresis on engine wall temperature with EMP Type A electric pump operation. Even though the thermostat is expanded at lower temperature than original thermostat case, the engine block wall temperature rarely falls because the heat transfer to coolant is lower with the smaller coolant mass flow. This result indicates that the initial coolant flow rate is enough to cool down the engine with the modified thermostat, and it is possible to improve the fuel economy through the minimum pump operation.

Figure 15 shows the pump power consumption during the FTP 74 driving cycle. Compared with mechanical pump, more than 86% power was saved by using electric pumps with the original thermostat due to higher efficiency of electric pumps. However, about more than 99% power reduction was achieved by the use of modified thermostat.

In order to utilize the electric pump for improving the fuel economy, the minimum pump speed, which was set at 200 rpm in this PID control, should be carefully determined. Otherwise, insufficient coolant flow rate due to low minimum pump speed setting for electric pumps could cause local hot spot on the engine block or head. Therefore, the minimum pump speed or coolant flow rate has to be investigated to prevent hot spot for the application of electric pump.

5. CONCLUSIONS

The cooling system model for V6, 4.5 L diesel engine was configured with a commercial program, GT-Cool. The simulation results were compared with the experimental data in order to verify this cooling system model, and then the effect of electric pump on the fuel economy was investigated. Finally, the additional benefit of electric pump with the modified thermostat was demonstrated with the simulation. The main conclusions of this research are summarized as follows;

(1) The cooling system model was successfully used to simulate the cooling system performance.

(2) Compared with mechanical coolant pump, power consumption can be reduced by about 86% during the FTP 74 driving cycle with two different types of electric pumps.

(3) Thermostat hysteresis has to be modified to take full advantage of the electric pump operation for more power saving.

(4) About 99% power reduction can be achieved with electric pump and modified thermostat hysteresis during the FTP 74 driving cycle.

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