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Numerical Modeling and Simulation of the Vehicle Cooling System for a Heavy Duty Series Hybrid Electric Vehicle

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ABSTRACT

The cooling system of Series Hybrid Electric Vehicles (SHEVs) is more complicated than that of conventional vehicles due to additional components and various cooling requirements of different components. In this study, a numerical model of the cooling system for a SHEV is developed to investigate the thermal responses and power consumptions of the cooling system. The model is created for a virtual heavy duty tracked SHEV. The powertrain system of the vehicle is also modeled with Vehicle-Engine SIMulation (VESIM) previously developed by the Automotive Research Center at the University of Michigan. VESIM is used for the simulation of powertrain system behaviors under three severe driving conditions and during a realistic driving cycle. The output data from VESIM are fed into the cooling system simulation to provide the operating conditions of powertrain components. The cooling system model includes various component models for three main fluid circuits of coolant, cooling air, and engine oil. The model predicts the thermal responses of all cooling system components and the temperatures of the engine and electric components. Using the cooling system models, the thermal response and power consumption of the cooling system over a realistic driving cycle is estimated and the factors that affect the performance and the power consumption of the cooling system are identified.

INTRODUCTION

The Series Hybrid Electric Vehicle (SHEV) is considered as an alternative platform for heavy duty military ground vehicles. It offers several advantages over conventional propulsion systems such as improved fuel economy, better acceleration performance, low acoustic signature, and exportable electric power. However, SHEVs need additional components such as a generator, driving motors, a battery pack, and a power bus, all of which make the cooling system more complicated. Thus a more strategic approach is required when designing a cooling system for heavy duty SHEV. SHEVs require a dedicated cooling system for the hybrid components due to the different cooling requirements of the components. The extra cooling system increases the cost and weight of the vehicle and affects its fuel economy. In addition, compared to the cooling system for commercial vehicles, the cooling system for combat vehicles cannot take advantage of the ram air effect because the system must be shielded from potential enemy attacks. All these factors make the design of a cooling system for SHEV combat vehicles challenging.

Traci and Acebal [1] demonstrated that a numerical approach could be used for thermal management system design of HEVs. They simulated a cooling system of an all-electric combat vehicle which uses a diesel engine as a prime power source and stores the power in a central energy storage system. The energy storage system consists of a flywheel and a large battery and is used for propulsion and weapon systems. They conducted parametric studies of the ambient temperature effect on the fan power consumption and the coolant temperature effect on the system size.

Park and Jaura [2] used a commercial software package to analyze the underhood thermal behavior of an HEV cooling system and studied the effect of the additional hardware on the performance of cooling system. They also investigated the effect of an electronic module cooler on the conventional cooling system.

These previous works, however, focused on parametric studies and they did not take the power consumption of cooling systems into consideration, which affects vehicle’s fuel economy. In this study, a numerical method is used to address the thermal responses and the power consumption of the cooling system based on the realistic driving cycle. Thermodynamics-based component models are developed and they are integrated into the cooling system for a virtual 20-ton tracked vehicle. The performance and power consumption of the cooling system for the SHEV are evaluated for extreme conditions and a realistic driving cycle.

COOLING SYSTEM SIMULATION

A cooling system has many sub-components such as coolant pumps, fans, radiators, thermostats, and heat sources. In this study, each component is modeled to predict thermal response and power consumption. The component models are integrated into a cooling system designed for a SHEV using Simulink® to investigate the performance and the power consumption of the cooling system based on vehicle driving conditions.

COOLING SYSTEM CONFIGURATION

The cooling system of a heavy duty SHEV needs more cooling circuit and component than that of a conventional vehicle because additional powertrain components require more cooling system components. In this study, a SHEV cooling system is configured by adding a separate cooling circuit for the electric components to a conventional cooling system of a diesel engine as shown in Fig. 1. All the electric components are integrated into one cooling circuit and an electric pump is used for the circulation of coolant. The battery pack is assumed to be cooled by the compartment Air Conditioning (AC) system due to its low operating temperature and the heat from the battery pack is considered to be dissipated through the AC condenser. This configuration is advantageous in terms of simplicity and minimal number of additional cooling components.

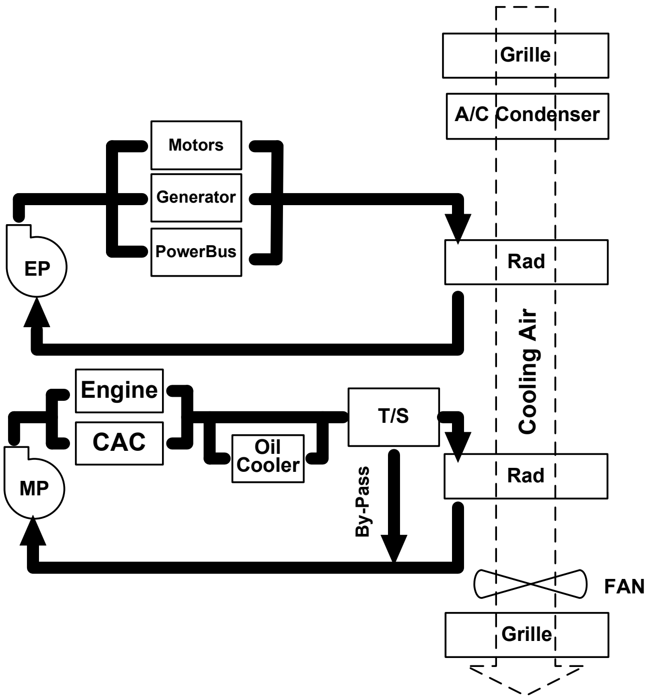


Figure 1. Schematic of SHEV cooling system (Rad: Radiator, EP: Electric Pump, MP: Mechanical Pump, T/S: Thermostat, CAC: Charge Air Cooler).

COMPONENT MODELING APPROACH

A cooling system model consists of various component models. Each component was carefully modeled with different fidelity depending on its influence and sensitivity to the cooling performance. The components can be catagorazied by their function in the cooling system: heat source, heat sink, and media delivery components. Each component has several sub-models for heat transfer, pressure drop, flow rate, and heat generation.

Heat Source Component Model

In heavy duty SHEVs, an internal combustion engine, a generator, drive motors, and a power bus are the heat source components. Lumped thermal mass model is used for the temperature calculation of all heat source components. In the model, the average temperature of the component is calculated from the balance of heat generation by the component, heat transfer to the coolant, and heat transfer to the ambient. Thus, the component temperature change is calculated from

$$\frac{dT_{comp}}{dt} = \frac{Q - q_{int} - q_{ext}}{\rho C_p} \tag{1}$$

$$q_{int} = (hA)_{int}\left(T_{comp} - T_{cool}\right) \tag{2}$$

The heat transfer to the ambient includes heat transfer by natural convection and by radiation.

$$q_{ext} = (hA)_{ext}\left(T_{comp} - T_{ext}\right) + \sigma A_{ext}(T_{comp}^4 - T_{ext}^4) \tag{3}$$

Heat generation of the engine is modeled with a look-up table type module. Engine heat rejection rate and Brake Specific Fuel Consumption data as a function of engine speed and Brake Mean Effective Pressure (BMEP) are provided by a user input file. In this work, the engine heat rejection rate and performance maps are obtained from the engine dynamometer test.

Heat generations by the generator and motor are calculated based on their efficiency. The efficiency lookup tables are adopted from the ADVISOR library [3]. The heat generation from the generator and motor is calculated with

$$Q_{gen} = \tau_{gen} \times \omega_{gen}\left(1 - \eta_{gen}\right) \tag{4}$$

$$Q_{mot} = \tau_{mot} \times \omega_{mot}\left(\frac{1}{\eta_{mot}} - 1\right) \tag{5}$$

The motor also works as an electric generator during braking mode which will be described in the vehicle simulation section. The heat generation from the motor during regenerative braking is calculated with

$$Q_{mot} = \left|\tau_{mot} \times \omega_{mot}\right|\left(1 - \eta_{mot}\right) \tag{6}$$

The heat generated by the power bus is calculated based on the power delivered by the power bus and the efficiency of the power bus. The power delivered by the power bus is determined based on the power management mode which will be described in the vehicle simulation section. In normal mode, all of the power from power sources is supplied to motors, thus, the power consumed by motors is the total power delivered by the power bus. Therefore, the heat generated by the power bus is:

$$Q_{pb} = (1-\eta_{pb})\left(\frac{\tau_{mot}\times\omega_{mot}}{\eta_{mot}}\right) \quad (7)$$

In recharging mode, the power supplied by the Power Generating Unit (PGU) which includes the engine and generator is consumed both by the motors and battery. Thus, the heat generated by the power bus is calculated based on the summation of the battery power and motor power delivered by the power bus.

$$Q_{pb} = (1-\eta_{pb})\left(\left|(VI + VI(1-\eta_{pb})\right| + \frac{\tau_{mot}\times\omega_{mot}}{\eta_{mot}}\right) \quad (8)$$

In braking mode, the power generated by the motor using the braking force is the total power delivered by the power bus and the heat generated by the power bus is:

$$Q_{pb} = (1-\eta_{pb})(\eta_{mot}\times\tau_{mot}\times\omega_{mot}) \quad (9)$$

Pressure drop across each heat source component is also calculated and used for the calculation of the coolant flow rate and power consumption of the coolant pump. For the calculation of coolant pressure drop across the engine, experimental correlation is used. The coolant pressure drop across the electric component is calculated by assuming that the coolant path in the component is a smooth pipe and the pressure drops across the components are calculated as [4]

$$\text{Laminar : } \Delta p = \frac{128\mu L\dot{V}}{\pi d^4} \quad (10)$$

$$\text{Turbulent : } \Delta p = 0.241 L\rho^{\frac{3}{4}}\mu^{\frac{1}{4}}d^{-4.75}\dot{V}^{1.75} \quad (11)$$

Heat Sink Component Model

Thermal resistance concept based 2-Dimensional Finite Difference Method (2-D FDM) developed by Jung and Assanis [5] is used for the modeling of the radiator. They employed 2-D FDM with staggered grid system to develop a numerical model with the predictive capability for various design parameters of the heat exchanger. The same model is also used for the charge air cooler. The only difference between a charge air cooler and a radiator is that heat is transferred from the compressed charge air to the coolant in a charge air cooler while heat is transferred from the coolant to the cooling air in a radiator. For the condenser of air conditioning device for passenger compartment, heat addition model is used. The heat rejection rate from the condenser is assumed to be constant.

Oil cooler model consists of heat source model, heat exchanger model, and oil pump model because another cooling circuit for oil circulation is needed. The heat from the engine is added to the engine oil and the heat added to the oil is transfered to the coolant through a heat exchanger. Thus, additional heat exchanger and additional pump are needed for the heat transfer between the oil and the coolant. A performance data based model is employed for the oil pump and Effectiveness-NTU method [6] is employed for the oil heat exchanger model. NTU for the heat exchanger is calculated from

$$NTU \equiv \frac{UA}{C_{\min}} \quad (12)$$

The effectiveness of the heat exchanger is calculated as

$$\varepsilon \equiv \frac{1-\exp(NTU(C_r-1))}{1-C_r\exp(NTU(C_r-1))} \quad (13)$$

Finally, the heat transferred in the oil cooler is:

$$q = \varepsilon C_{\min}(T_{h,i} - T_{c,i}) \quad (14)$$

Media Delivery Component Model

The function of media delivery component in a cooling system is controlling the fluid which carries the heat to maintain the heat source component under its control target temperature. As shown in Table 1, each heat source component has its own control target temperature, which is the maximum allowable temperature that should be maintained by the cooling system. Media delivery component includes coolant pump, cooling fan, and thermostat.

Coolant pump model - Coolant pump model calculates the coolant flow rate based on the total pressure drop across the cooling system components and the pump speed. To calculate the coolant flow rate, performance map which consists of flow rate, pressure rise and pump speed is used. The mechanical pump is driven by the engine and the electric pump is driven by an electric motor. The electric pump can be used to control the component temperature by adjusting the electric motor speed which drives the pump. But, mechanical pump cannot be used to control the temperature because the pump speed is dependent on the engine speed. Thus, to prevent the overcooling of the coolant, a thermostat is necessary for the cooling circuit where a mechanical pump is used.

Thermostat model - A thermostat is a three way valve which controls the coolant temperature by channeling the coolant to the radiator or to the by-pass circuit. The valve opening sizes to the radiator and to the by-pass circuit are determined by the temperature and hysteresis characteristics of the thermostat which is shown in Fig. 2. The thermostat temperature is calculated with the lumped thermal mass model. The coolant flow rates to the by-pass circuit and to the radiator circuit are determined at the point where the pressure drops through each circuit are equal to each other. Figure 3 shows the concept of the flow rate calculation model of the thermostat model.

Table 1. Control target temperatures of powertrain components.

Component	Control target temperature (°C)
Engine	120
Motor	95
Generator	95
Oil cooler	125
Power bus	70
Battery	45

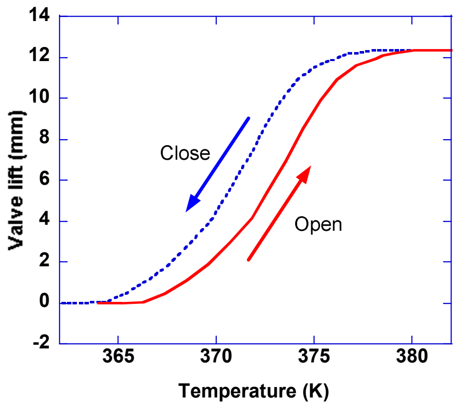


Figure 2. Valve lift curve of thermostat with respect to the thermostat temperature with hysteresis characteristics.

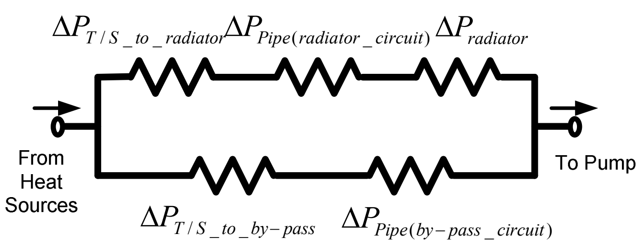


Figure 3. Flow rate calculation of thermostat model based on system resistance concept.

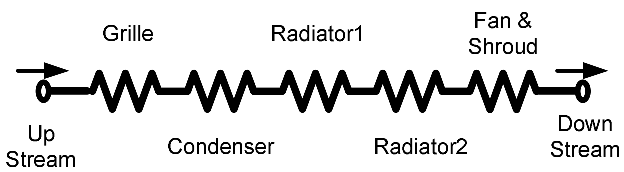


Figure 4. Air duct system based on system resistance concept.

Cooling fan model - Cooling fan model is similar to the pump model because cooling fan model calculates the cooling air flow rate based on the total pressure drop across grilles and heat exchangers and the fan operating speed. Figure 4 shows the concept of cooling fan model. To calculate the cooling air flow rate, performance map which consists of flow rate, pressure rise and fan speed is used.

COOLING SYSTEM SIZING

Cooling system design has two constraints that need to be satisfied: cooling performance and packaging. The cooling system size is limited by the packaging space in the vehicle while the system should be capable of removing heat generated by heat sources within available space. Since there is a trade-off relationship between the two constraints a cooling system should be carefully designed to satisfy both constraints.

Radiator and pump sizes are the main design variables that determine the cooling capacity. A scaling method is developed for the initial estimation of radiator and pump sizes. The initial sizes of radiator and pump in each cooling circuit are estimated by scaling sizes of the components from a referenced cooling system of a conventional vehicle based on the heat rejection rate, as is described in the Appendix. Once the scale factors are found, the estimated radiator size is checked whether it is within the packaging constraint. The radiator frontal size is limited considering the specifications of the existing vehicles at the same class. Thus, if the radiator size is out of the limit, the radiator size is reduced to the limit and the pump size is rescaled to compensate the smaller radiator size. After the draft design is completed, the pump and radiator are resized until all component temperatures can be controlled lower than their control target temperatures under the severe condition.

SERIES HYBRID ELECTRIC VEHICLE SIMULATION

The cooling system simulation requires component operating conditions as a function of time to simulate the thermal response of the cooling system when the vehicle is driven over a driving cycle. In this study, a vehicle model is used to provide the operating condition data of the SHEV powertrain components to the cooling system simulation. The output data of the vehicle simulation are used as input data to the cooling system simulation.

VEHICLE CONFIGURATION

Compared to conventional vehicle powertrains, those of SHEVs are characterized by additional electric components and complicated power management modes. The schematic of a SHEV propulsion system modeled in this study is illustrated in Fig. 5. The schematic shows the main powertrain components of the SHEV propulsion system and the arrows in the schematic indicate the direction of power flow. The SHEV propulsion system is composed of an internal combustion engine, a generator, a power bus, a battery pack, and two drive motors. In conventional vehicles, the internal combustion engine is the prime power source, thus the mechanical power from the engine is transferred to driving shafts through a transmission. However, in SHEVs, the engine power is converted to the electricity with the generator and it is stored in the battery or directly used by the drive motors depending on the power management mode. The electricity is managed to power the drive motors or to charge the battery by the power bus which includes the inverter and the voltage-boosting converter. The drive motors are powered by the electricity from the generator or the battery. Detailed power management strategy is presented in the following.

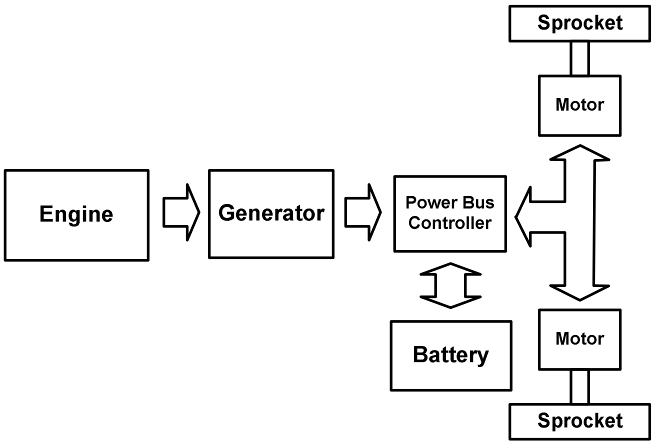


Figure 5. Schematic of series hybrid vehicle propulsion system.

POWER MANAGEMENT STRATEGY

The power management process starts by interpreting the driver pedal signal as a power request. Depending on the power request and the battery State of Charge (SOC), the power management strategy is divided into three control modes: Braking mode, Normal mode, and Recharging mode. An active brake pedal position is interpreted as a negative power request and the braking mode is engaged. Regenerative braking is activated to absorb braking power within the limits of the battery and motor. Friction braking is activated when the braking power request exceeds the regenerative braking capacity. If the power request is positive with an active acceleration pedal position, either normal mode or recharging mode is used according to a charge-sustaining policy. The charge-sustaining strategy assures that the battery SOC stays within the preset lower and upper bounds for efficient operation and prevention of battery depletion or damage. In a normal propulsive driving condition, the PGU is shut off and the battery supplies the requested power to the driving motors. However, once the power request exceeds what the battery can generate, the PGU is turned on to supply the additional power. Whenever the SOC drops below the lower limit, the controller will switch to recharging mode until the SOC reaches the upper limit and then normal mode will resume. In recharging mode, the PGU provides additional power to charge the battery in addition to powering the driving motors. If the total power request is greater than the maximum PGU power, the battery assists the PGU to power the driving motors. Once the PGU power request is determined, the engine is operated at the point that provides the best combined efficiency of the PGU for the required engine power level [7].

VEHICLE SPECIFICATIONS

Based on the configuration of vehicle components and the power management modes, a vehicle model with a SHEV propulsion system is created employing Vehicle-Engine SIMulation (VESIM) which has been developed at the Automotive Research Center (ARC) at the University of Michigan [7]. The model is used to acquire the operating conditions of the SHEV powertrain components. The specifications of the virtual SHEV simulated in this study are summarized in Table 2. They are selected for a 20-ton heavy duty tracked vehicle. A turbocharged diesel engine is chosen for better efficiency against the spark ignition engine and for lower cost against the gas turbine. The rated engine power is determined based on the power (kW) to weight (ton) ratio of 15. Generator and motor capacities are determined to fully convert the power from the engine to electricity and propulsion. Two AC induction type electric motors are used to drive two separate tracks of the vehicle. Lead-acid battery is used to store and supply electric power. The maximum vehicle speed is limited by the track dynamics and durability and is assumed to be governed at 72 km/h (45 miles/h), which is the typical maximum speed of the compatible tracked vehicles.

Table 2. Specifications of series hybrid electric vehicle.

Component	Type	Specification
Vehicle	Tracked Vehicle with Series-Hybrid Electric Powertrain	20 tons
Engine	Turbocharged Diesel Engine	300 kW
Generator	Permanent Magnetic	300 kW
Motor	AC Induction	2 × 150 kW
Battery	lead-acid	18Ah/120 modules
Maximum speed	(Governed)	72 km/h

DRIVING CONDITIONS AND CYCLE

The capacity of a cooling system should be enough to remove all the waste heat generated by the hardware under an extreme operating condition. Thus, an extreme condition simulation is necessary for cooling system design. To find the extreme condition for the cooling system, three conditions are evaluated. The three conditions are grade load, maximum speed, and off-road conditions which are summarized in Table. 3. These conditions are simulated and the most severe condition is used for sizing the cooling system components.

One of the goals of cooling system design is minimizing power consumption of the cooling system as the vehicle carries out a typical mission. This mission consists of combined mode of urban and cross-country driving and the specific road and speed profiles are shown in Fig. 7. The first part of the driving cycle represents urban driving with a flat road profile and frequent accelerations and decelerations. In the second part, which represents cross-country driving, the driver attempts to maintain constant speed on roads with uneven road profile. This combined driving cycle is used to evaluate the parasitic power consumption by the cooling system.

Table 3. Vehicle driving conditions.

Condition	Grade load	Max. speed	Off-road
Vehicle speed	48 km/h	72 km/h	48 km/h
Road profile	7% (uphill)	flat	Fig. 6
Ambient temp.	40ºC	40ºC	40ºC

RESULTS AND DISCUSSION

VEHICLE SIMULATION RESULTS

Figures 8 and 9 show an example of vehicle simulation results. Figure 8 shows the operating conditions of powertrain components under the grade load condition. As can be seen, the vehicle switches between the normal and recharging modes. The motor speed follows the speed of the vehicle but other components are controlled by the hybrid vehicle controller based on the power management mode. The vehicle simulation results under three driving conditions are used to find out which of the three driving conditions is most severe to the cooling system. Figure 9 shows the operating conditions of the powertrain components over the combined driving cycle. As can be seen, the engine is at idle for significant period of the driving cycle compared with the result of grade load condition because the required power for driving is lower than that under grade load condition. The motor speed follows the vehicle speed of the combined driving cycle and the battery SOC is controlled within the control limits (0.6~0.7).

HEAT GENERATIONS FROM SHEV POWERTRAIN COMPONENTS AND COMPONENT SIZING

Heat generation rates from all the heat source components under three driving conditions are compared with each other to find the most severe condition for the cooling system. Figure 10 shows the histories of heat generation rates from all heat source components for three conditions. Using the vehicle simulation results as input data, cooling system

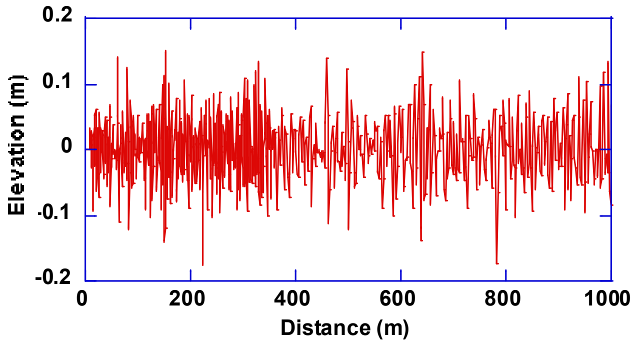
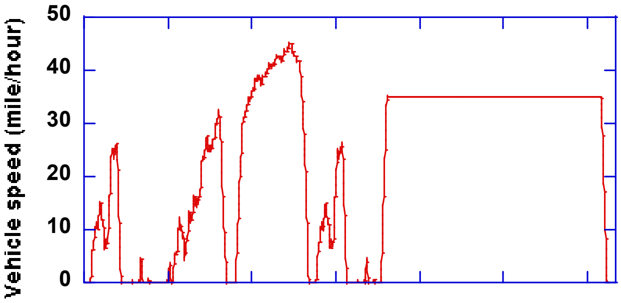
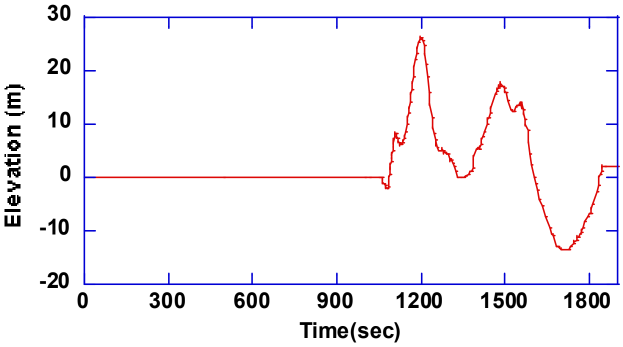


Figure 6. Off-road profile.

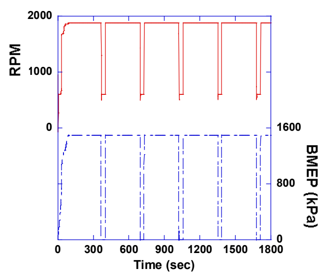


Velocity Profile

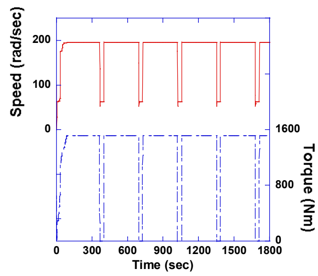


Road Profile

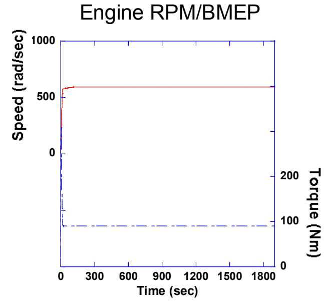
Figure 7. Combined driving cycle.



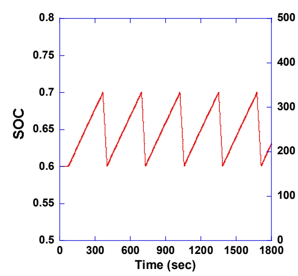
Engine RPM/BMEP



Generator Speed/Torque



Motor Speed/Torque



Battery SOC

Figure 8. Operating conditions of powertrain components under grade load condition.

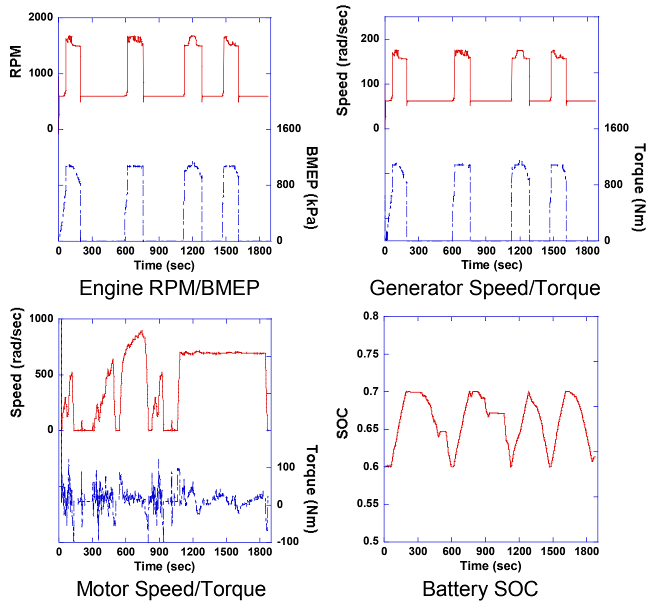


Figure 9. Operating conditions of powertrain components over combined driving cycle.

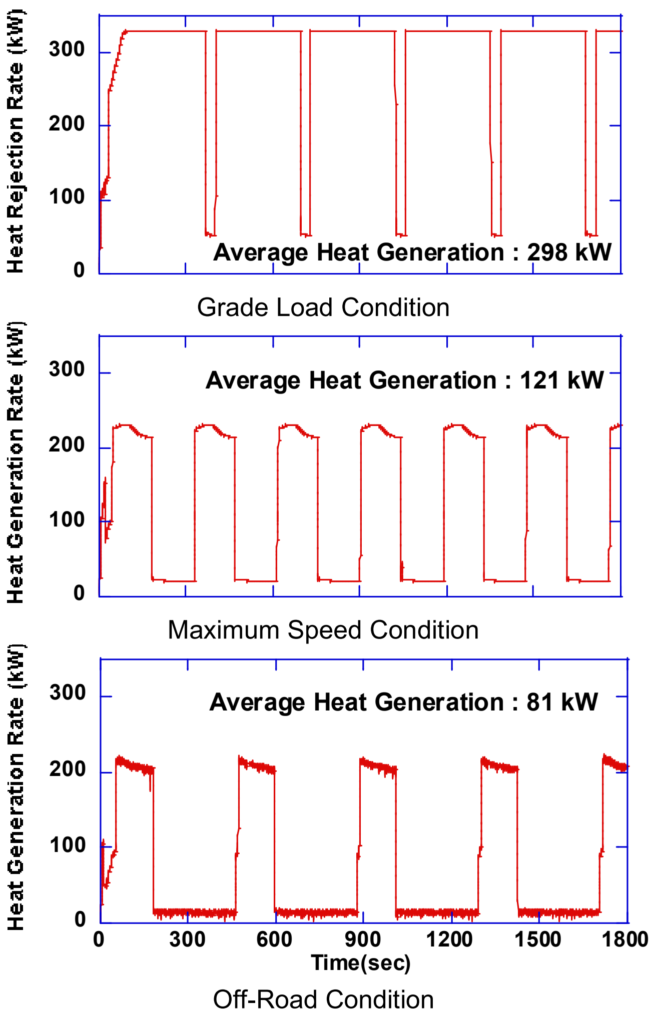


Figure 10. Heat generation rates under three driving conditions.

simulations are conducted for 1800 seconds. As shown in Fig. 10, the heat rejection rate shows periodical behavior because the power management mode changes between normal mode and recharging mode. The average heat generation as well as the peak heat generation under grade load condition is much larger than those under the other two conditions. This means that grade load condition is the most severe among them.

Table 4 presents the peak value of the heat generation from each of the SHEV powertrain component under grade load condition. Considerable heat is generated from the electric components. For example, the generator produces 65kW at its peak under the grade load condition which is 34% of the peak heat rejection by the engine. Since all the heat generation from each component should be rejected at the radiator under severe condition, the peak heat generation rate is used for the first estimation of the radiator and pump sizes by using the scaling method.

Table 4. Peak heat generation rates from powertrain components under grade load driving condition.

Component	Peak Heat Generation Rate (kW)
Engine	190
Motor	27
Generator	65
Charge air cooler	13
Oil cooler	40
Power bus	6
Battery	12

Table 5. Final cooling system component specifications.

Component	Specification
Coolant Pump for Engine Module	770 liters/min, 4644rpm
Coolant Pump for Electric Components	1100 liters/min, 4450rpm
Radiator for Engine Module	0.6m(height) x 1.2m(width) x 0.075m(thickness)
Radiator for Electric Components	0.6m(height) x 1.2m(width) x 0.1m(thickness)
Cooling Fan	24.6m^3/s, 3000rpm

As described earlier, the final sizes of the pump and radiator are determined for the grade load condition. The sizes of the pump and radiator are scaled so that the cooling circuit can control the component temperature under the cooling target temperature of the component. The temperature distribution in a heat source including the engine, generator, motors and power bus should be minimized by the cooling system because large temperature distribution can deteriorates the durability of the heat source component. Thus, the coolant temperature change across the heat source component should be limited to minimize the temperature distribution. The coolant temperature change can be controlled by changing the pump and radiator sizes. Larger radiator increases the coolant temperature change, while larger pump decreases the coolant temperature change. Thus, the pump and radiator sizes are tuned for the coolant temperature change not to exceed 10 ºC. The final cooling system component sizes determined for the grade load condition are presented in Table 5.

Figure 11 shows the temperature histories of the electric components and the coolant temperature over grade load condition. The temperature of the coolant entering the heat source component is marked by “Coolant In” and the temperature of the coolant exiting from the heat source component is marked by “Coolant Out”. As can be seen, the component temperatures start from ambient temperature (40ºC) and reach their control target temperatures, and then controlled under their control target temperature by the cooling system.

The coolant temperature in the electric component cooling circuit is limited by the control target temperature of the power bus because it is lower than those of others in the same cooling circuit. Thus, in Fig. 11, it is observed that the “Coolant In” temperature is controlled lower than 340 K which is under the control target temperature of the power bus (343 K). Accordingly, temperatures of generator and motor are lower than their control target temperature (368 K).

PERFORMANCE AND POWER CONSUMPTION OF COOLING SYSTEM OVER A REPRESENTATIVE DRIVING CYCLE

The power consumption of a cooling system is one of the important criteria designing the cooling system because the parasitic loss by cooling system affects the fuel economy of the vehicle. Figure 12 presents the parasitic power consumption by the cooling components with respect to the power output of the engine. Since the engine power and the cooling system power consumption change frequently over the driving conditions, both of the powers are averaged over the period and the percentage of cooling system power consumption relative to the power supplied by the engine is calculated. It is notable that the cooling system consumes from 7.7% to 11.6 % of the engine power in this heavy duty SHEV vehicle.

Figure 12 also shows the power consumptions of the fan and pumps under different driving conditions. As can be seen, the fan consumes more power than the pumps. In case of grad load condition, the fan consumes 81% of the power consumed by the cooling system. Another noticeable result is that the mechanical pump consumes much more power than the electric pump. This is mainly due to the fact that the mechanical pump should deliver much more coolant than the electric pump due to larger heat rejection from the engine compared to the heat rejection from the electric components. Another reason

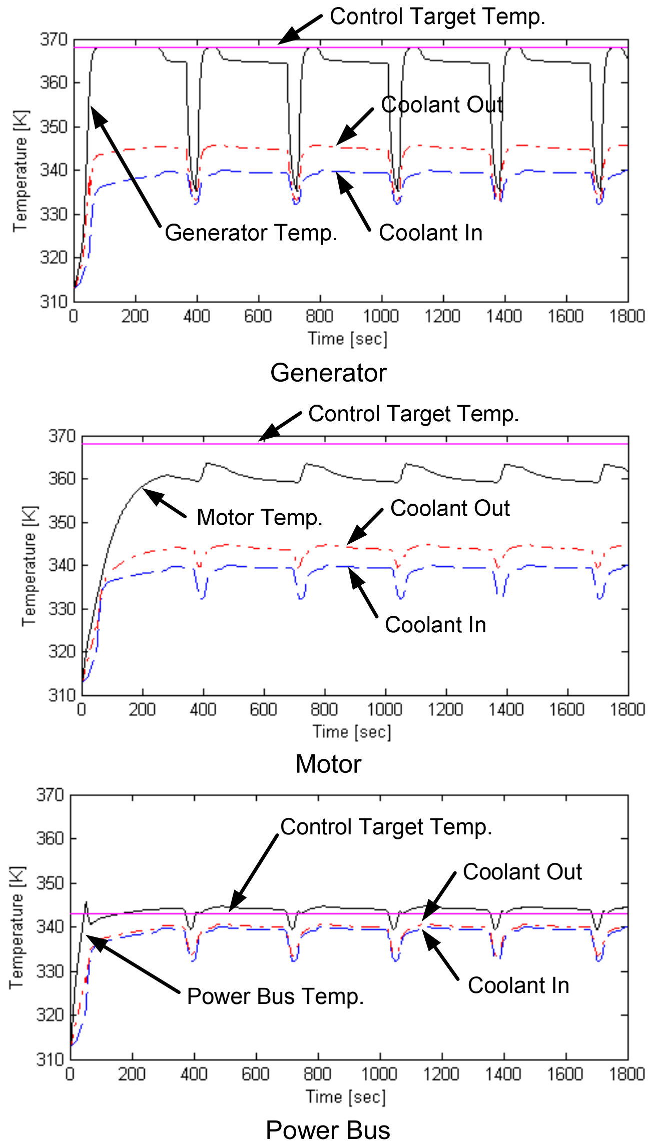


Figure 11. Temperature histories of electric components under the grade load condition.

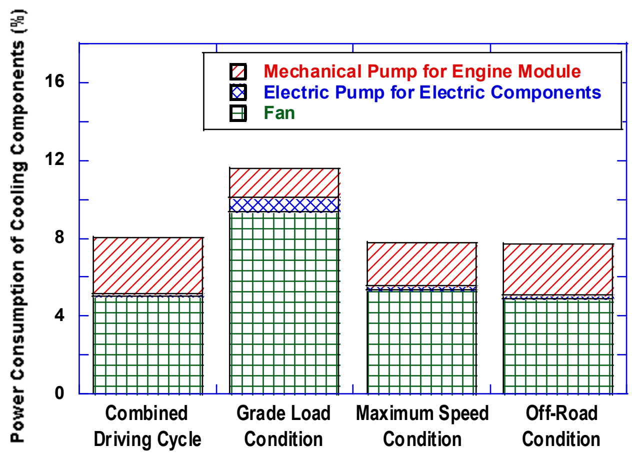


Figure 12. Parasitic power consumption of cooling components.

is that the mechanical pump speed cannot be controlled based on the engine temperature because it is directly connected to the engine. Since the pump operates depending on the engine speed regardless of the engine temperature a thermostat is used to control the coolant temperature by channeling the coolant to the radiator or the by-pass to prevent overcooling. In contrast, the electric pump speed is controlled by an electric motor based on the electric component temperature and consumes power only when needed. In SHEV it is relatively easy to convert mechanical pump-thermostat system to electric pump system with readily available power supply and this is expected to reduce the power consumption by the pump with better controllability.

Figure 13 shows the temperature histories of the electric components over the combined driving cycle. All the electric component temperatures are controlled well under their control target temperatures over the combined driving cycle. However, the electric component temperatures fluctuate frequently. In Fig. 13, it can be observed that the fluctuation is aggravated because the electric components share one cooling circuit. For example, at 450 second, the power bus temperature reaches its control target temperature, thus the coolant pump and fan are activated to cool down the power bus. As a result, the generator is cooled down together because they share the same cooling circuit. This causes the generator temperature deviate more from its control target temperature. This result suggests that more study on the configuration of SHEV cooling system is needed to design better system configuration.

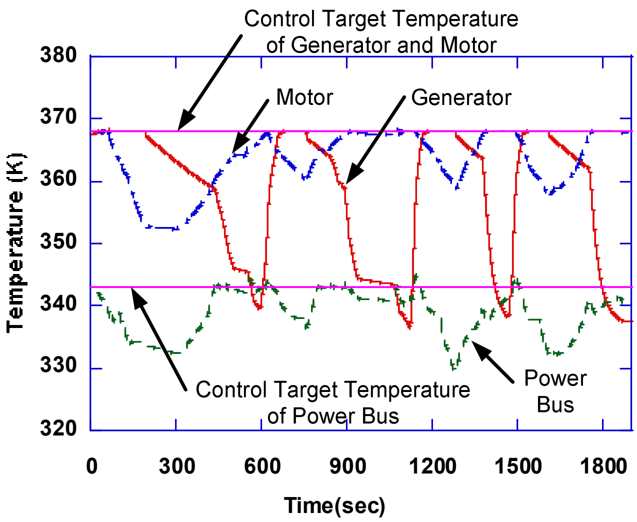


Figure 13. Temperature histories of the electric components over the combined driving cycle.

CONCLUSION

A cooling system model for a heavy duty SHEV has been developed for the studies of the cooling system design and performance evaluation. Cooling system simulation is conducted for three driving conditions and a representative driving cycle and the main conclusions of this study are summarized as follows:

1. In heavy duty applications of SHEV, a dedicated cooling circuit is required for electric powertrain components due to considerable heat generated from the electric components.
2. In the cooling system simulated in this study, the coolant temperature of the electric component cooling circuit is limited by the control target temperature of the power bus because it is lower than those of others that share the same cooling circuit. Accordingly, temperatures of generator and motor are lower than their control target temperature.
3. The cooling system consumes from 7.7% to 11.6 % of the engine power depending on the driving condition. The fan consumes more power than the pumps. Another noticeable result is that the mechanical pump consumes much more power than the electric pump. However, the power consumption of cooling system can be reduced by replacing the mechanical pump-thermostat system with an electric pump with better controllability.
4. Cooling system of SHEVs should be carefully configured because of various cooling requirements of powertrain components, power management strategy, parasitic power consumption, and the effect of driving conditions.
5. It is demonstrated that the numerical cooling system model provides a useful tool for the design and parametric assessment of a cooling system for an advanced concept vehicle.

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DEFINITIONS, ACRONYMS, ABBREVIATIONS

A : area

C_p : specific heat

C : fluid heat capacity rate

C_r : the ratio of minimum to maximum fluid heat capacity rate (C_{min}/C_{max})

d : diameter

h : convection heat transfer coefficient

I : electric current

L : length

$\dot{m}$: mass flow rate

NTU : number of transfer units

p : pressure

q : heat transfer rate

Q : heat generation

T : temperature

U : overall heat transfer coefficient

V : voltage

$\dot{V}$: velocity

Greek

α : scaling factor

η : efficiency

ω : angular velocity

τ : torque

ρ :density

π : ratio of the circumference of a circle to the diameter

ε : effectiveness

σ : Stefan-Boltzmann constant

μ : dynamic viscosity

Subscripts and Superscripts

cap : capacity

$comp$: component

$cool$: coolant

ext : external

gen : generator

int : internal

min : minimum

max : maximum

mot : motor

pb : power bus

$pump$: pump

rad : radiator

ref : reference

APPENDIX: COOLING SYSTEM SCALING

The heat rejection from a radiator is proportional to the coolant flow rate and it is also proportional to the product of radiator frontal area and the temperature difference between the ambient air and the coolant. Thus, this can be summarized as

$$q_{rad} \propto \dot{m}_{cool} \tag{A1}$$

$$q_{rad} \propto A_{rad}\Delta T \tag{A2}$$

where ΔT is the temperature difference between the coolant and the ambient air. Therefore,

$$q_{rad} \propto \dot{m}_{cool} A_{rad} \Delta T \tag{A3}$$

Thus, the product of coolant flow rate and radiator area should be proportional to the heat rejection from the radiator and inversely proportional to the temperature difference:

$$\dot{m}_{cool} A_{rad} \propto \frac{q_{rad}}{\Delta T} \tag{A4}$$

In Eq. (A4), the heat rejection from the radiator (q_{rad}) can be replaced by the heat generated by the component

(q_{comp}) in the cooling circuit because the heat generated by the component should be rejected at the radiator.

$$\dot{m}_{cool} A_{rad} \propto \frac{q_{comp}}{\Delta T} \tag{A5}$$

Assuming that the coolant flow rate is proportional to the pump capacity and calculating the heat generation from each component model, the pump capacity and radiator size are scaled based on the following scale ratio.

$$\dot{m}_{pump\_cap,ref} A_{rad,ref} : \dot{m}_{pump\_cap} A_{rad} = \left(\frac{q_{comp}}{\Delta T}\right)_{ref} : \left(\frac{q_{comp}}{\Delta T}\right) \tag{A6}$$

To get the first estimation of the pump and radiator size, the scale factors for the pump and the radiator are assumed to be the same,

$$\dot{m}_{pump\_cap} = \alpha \dot{m}_{pump\_cap,ref} \quad , \quad A_{rad} = \alpha A_{rad,ref} \tag{A7}$$

Plugging them into Eq. (A6), the scaling factor can be found as

$$\alpha^2 = \left(\frac{q_{comp}}{\Delta T}\right) \Bigg/ \left(\frac{q_{comp}}{\Delta T}\right)_{ref} \tag{A8}$$