

An Engine Thermal Management System Design for Military Ground Vehicle - Simultaneous Fan, Pump and Valve Control

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ABSTRACT

The pursuit of greater fuel economy in internal combustion engines requires the optimization of all subsystems including thermal management. The reduction of cooling power required by the electromechanical coolant pump, radiator fan(s), and thermal valve demands real time control strategies. To maintain the engine temperature within prescribed limits for different operating conditions, the continual estimation of the heat removal needs and the synergistic operation of the cooling system components must be accomplished. The reductions in thermal management power consumption can be achieved by avoiding unnecessary overcooling efforts which are often accommodated by extreme thermostat valve positions. In this paper, an optimal nonlinear controller for a military M-ATV engine cooling system will be presented. The prescribed engine coolant temperature will be tracked while minimizing the pump, fan(s), and valve power usage. A case study investigates the proposed control strategy's performance in comparison to other methods for temperature tracking and energy conservation. The optimal nonlinear controller offered satisfactory coolant temperature tracking with an average error of 0.35°C and at least 13% reduction in total cooling power.

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1. INTRODUCTION

Modern ground vehicles apply a variety of electronic sensors, on-board controller systems, and electric driven actuators to regulate the powertrain's operation for improved fuel economy and reduced tailpipe emissions. Advanced control algorithms have been introduced for precise fuel injection, spark delivery, air flow management, and transmission shifting to name a few process to satisfy federal regulations while meeting consumer demands. Applying model-based control strategies for accurate thermal management system operation is a promising approach as vehicle cooling has not yet received thoughtful widespread attention from the automotive engineering community.

In an engine cooling system, the actuators (pump, fans) operate at full rotational speed to reach their maximum heat removal capabilities when the engine load and ambient temperature are high. In this instance, the thermostat valve will be fully open. In contrast, the actuator speeds can be reduced and thermostat valve predominately closed when the thermal load and ambient temperatures are low which promotes passive convective cooling. However, in between these extremes, the thermal management system's operation needs to be optimized for variable heat rejection to achieve temperature

tracking with minimal power usage. A medium thermal load under a moderate surrounding temperature provides a space for advanced thermal management controller designs as shown in [Fig. 1](#).

An efficient thermal management system design is essential for powertrain reliability, fuel economy, and performance [1]. In a study by Park and Jung [2], various powertrain cooling system architectures and the accompanying effect on the power consumption has been investigated. The upgrade of mechanical actuators in cooling systems by real time controlled electro-mechanical components facilitates improved efficiency under most operating conditions. A variety of control strategies, applied to advanced vehicle powertrain thermal management systems, have been studied [3][4]. The integration of an electric pump and smart valve for faster engine coolant warm up time and reduced temperature fluctuation has been reported in [5]. Cho *et al.* [6] explored the benefit of a controllable electric pump for truck engine cooling which offers a significant reduction in power consumption and possible heat exchanger downsizing. Page *et al.* [7] investigated a classical PID controller for the cooling system featuring with multiple electric radiator fans and heat controlled thermostats. A suite of nonlinear control strategies was developed by Salah *et al.* [8] to adjust the smart valve position and the coolant flow rate to track the desired engine temperature. A differential flatness nonlinear controller has demonstrated improved temperature trajectory tracking performance [9].

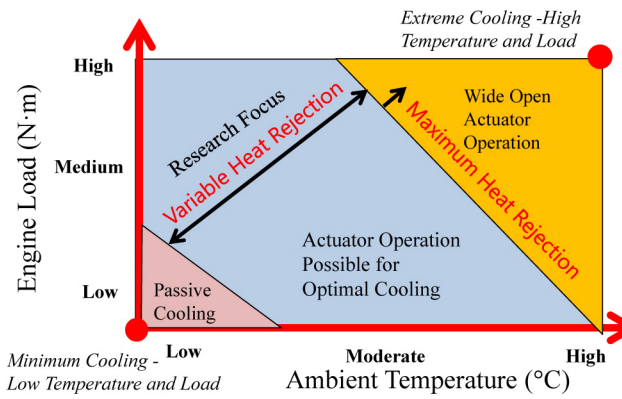


Figure 1. Engine Cooling System Optimization Control Space for Power Cost Reduction: Variable Heat Rejection for Medium Thermal Load under Moderate Ambient Temperature

The radiator fan(s) has been identified to consume that greater power within engine cooling systems when compared to the other components (pump, smart valve) [10]. Recently, attention has focused on optimizing the fan(s) speed to minimize power usage while rejecting sufficient heat. Wang *et al.* [11] conducted a detailed experimental analysis of multiple electric radiator fan configurations. A rule of thumb and optimization control strategy for total fan array power minimization based on thermal load was proposed [12]. Within the area of hybrid electric vehicles (HEV), Tao *et al.* [13] developed a model predictive controller to regulate the compressor speed in vapor compression systems to track cooling air temperature and stabilize core battery temperatures within battery packs. A high fidelity electric motor thermal model was implemented in a heavy-duty HEV cooling system controller which tracked the motor's inner peak temperature [14].



Figure 2. Notional MRAP All-Terrain Vehicle (M-ATV)

This research study will examine an engine thermal management system for military ground vehicles, specifically an M-ATV, as shown in Fig. 2. The proposed control strategy will synchronously regulate the radiator fan, coolant pump, and smart valve operations to track the prescribed reference temperature with consideration power consumption. The remainder of this paper is arranged as follows. In Section 2, a lumped parameter thermal model is presented that describes the engine cooling system's thermal behavior. In addition, a suite of actuator models will be developed to estimate the power consumption. In Section 3, an optimal nonlinear controller, with two supplemental controllers, will be designed. A case study with

numerical results will be presented and discussed to explore the performance and power consumption for different control strategies in Section 4. Section 5 concludes this work. A complete Nomenclature List and derivation of the desired cooling air temperature at the radiator outlet are offered in the Appendix.

2. LIBRARY OF MATHEMATICAL MODELS

An engine thermal management system, featuring electric driven actuators, will be mathematically modeled to establish a basis for controller designs. The M-ATV proposed cooling system configuration consists of an internal combustion engine (ICE), electric coolant pump, smart valve, variable speed fan, and radiator (refer to Fig. 3). A lumped parameter approach will be pursued to develop the differential and algebraic equations.

2.1. Thermal Dynamics of the Engine and Heat Exchanger

The coolant is circulated by the electric pump through the cooling system as shown in Fig. 3. The smart valve allows a temperature dependent portion of the total coolant flow through the radiator and the remainders goes back to engine directly. The coolant discharges the combustion heat in the radiator via forced convective heat transfer. The governing thermal equation for the coolant temperature change at the engine outlet, T_h , may be expressed as

$$M_{w,e} c_{p,w} \frac{dT_h}{dt} = Q_e + \dot{m}_w c_{p,w} (T_m - T_h) \quad (1)$$

where $M_{w,e}$ is the coolant mass inside the engine. The parameter $c_{p,w}$ is the specific heat of water. The term Q_e represents the engine heat of combustion. The second term on the right side of Eq. (1) is the heat removal attributed to the coolant mass flow rate, $\dot{m}_w$.

The coolant at the pump's outlet, or engine inlet, is a mixture of the cold and hot coolant streams. The smart valve regulates the coolant mass flow through the radiator and bypass. A portion of the coolant will flow into the radiator and discharge the waste heat to the ambient surroundings. The bypass circuit directs the remaining fluid flow right to pump. In this study, the valve is electrically driven and linearly controlled such that $0 \leq K_v \leq 100\%$. When $K_v = 100\%$, the valve is fully open and all the coolant will be directed into the radiator. In contrast, $K_v = 0\%$ means that the valve is closed and all the coolant is routed through the bypass to the pump.

The the coolant temperature at the pump's outlet, T_m , is a linear combination of the hot, T_h , and cold, T_c , coolant temperatures defined as

$$T_m = (1 - K_v)T_h + K_v T_c \quad (2)$$

The differential equation for cold coolant temperature at the radiator outlet, T_c , may be written as

$$M_{w,r} c_{p,w} \frac{dT_c}{dt} = K_v \dot{m}_w c_{p,w} (T_h - T_c) - Q_r \quad (3)$$

where $M_{w,r}$ is the coolant masses inside the radiator, the variable Q_r on the right side is the heat transfer from the coolant to the cooling air in the radiator. The term $K_v \dot{m}_w c_{p,w} (T_h - T_c)$ denotes the heat transferred by the coolant flow itself due to temperature differences. Finally, the dynamic change in the cooling air temperature at the radiator outlet, $T_{a,o}$, is modeled as

$$M_{a,r} c_p \frac{dT_{a,o}}{dt} = Q_r + \dot{m}_a c_p (T_{amb} - T_{a,o}) \quad (4)$$

In this expression, the parameter $M_{a,r}$ is the air mass inside the radiator control volume, and c_p denotes the specific heat of air. The term T_{amb} represents the temperature of the ambient air, which will be delivered through the radiator by the variable speed fan at mass flow rate, $\dot{m}_a$.

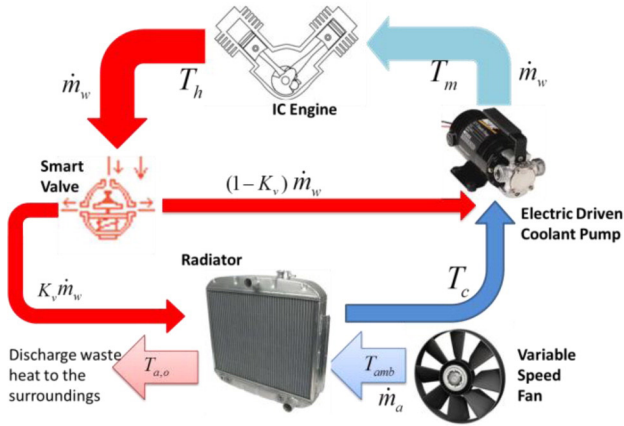


Figure 3. Engine Cooling System Configuration

2.2. Cooling Actuators - Radiator Fan and Pump

The total power consumption of the cooling system should be reduced to help satisfy the legislated fuel and emissions requirements. Only the power consumed by the radiator fan and the coolant pump will be analyzed, given that the power used by the smart valve is negligible. The radiator heat transfer is directly affected by the cooling air mass flow rate provided by the fan which may be written as

$$\dot{m}_a = D_{fan}^3 C_{flow} \rho_a \left(\frac{N_f}{60} \right)^2 \quad (5)$$

Where N_f is the fan speed, and C_{flow} is the flow rate coefficient. The parameters D_{fan} and ρ_a denote the equivalent fan diameter and the cooling air density.

The total air pressure increase due to the fan, dP_{fan} may be calculated as

$$dP_{fan} = D_{fan}^2 C_{pres} \rho_a \left(\frac{N_f}{60} \right)^2 \quad (6)$$

where the parameter C_{pres} is defined as the pressure drop coefficient across the radiator. The fan power consumption rate, E_{fan} , is given as

$$E_{fan} = D_{fan}^3 C_{flow} dP_{fan} \left(\frac{N_f}{60} \right) \quad (7)$$

For a given cooling system, the values of C_{flow} , C_{pres} , D_{fan} , and ρ_a are constant. Thus, the estimation of the radiator fan power consumption can be characterized as a function of the air mass flow rate so that

$$E_{fan} = k_{fan} \dot{m}_a^3 \quad (8)$$

with $k_{fan} = D_{fan}^{-4} C_{pres} \rho_a^{-1} C_{flow}^{-2}$. The coolant mass flow rate, $\dot{m}_w$, driven by the electric pump, is dependent on the pump rotation speed, N_p , such that

$$\dot{m}_w = D_{is} \rho_w \left(\frac{N_p}{60} \right) \quad (9)$$

where the term D_{is} is the maximum pump displacement. The parameter ρ_w denotes the water density while the variable N_p is the pump speed.

The total pressure drop across the radiator and the pipe in the coolant flow is calculate as

$$dP_w = \rho_w C_{p,rad} \left(\frac{N_p D_{is}}{60 A_{flow}} \right)^2 + \dot{m}_w^2 C_{p,pipe} \quad (10)$$

where $C_{p,rad}$ and $C_{p,pipe}$ denote the pressure rise coefficients in the radiator and pipe, respectively. The parameter A_{flow} is the coolant flow cross section area in the radiator.

The pump power consumption rate, E_{pump} , is defined as

$$E_{pump} = D_{is} dP_w \left(\frac{N_p}{60} \right) \quad (11)$$

The pump's power consumption can be summarized as a function of the coolant mass flow rate given the parameters A_{flow} , $C_{p,rad}$, $C_{p,pipe}$, D_{is} and ρ_w for a given cooling system as

$$E_{pump} = k_{pump} \dot{m}_w^3 \quad (12)$$

where k_{pump} is defined as $\rho_w^{-1} C_{p,pipe} + \frac{A_{flow}^2}{60^2} C_{p,rad}^2$.

Note that the power consumption of the water pump and radiator fan are both proportional to the cube of the given fluid flow rate.

3. CONTROLLER DESIGNS

An optimal nonlinear controller, based on the thermal model, will be designed to minimize the total power consumption of the prescribed ICE cooling system. For comparison purposes, two other control strategies will also be introduced including a state flow controller, and a classical PI controller.

3.1. Optimal Nonlinear Control

An optimal nonlinear controller will simultaneously regulate the operation of the coolant pump, radiator fan, and smart valve to track the desired engine temperature while using the theoretically minimum total power. The primary control objective is to stabilize the coolant temperature, T_h , to the prescribed target value, $T_{h,d}$.

3.1.1. Pump Speed Control

The coolant pump electric motor shaft speed will be controlled by an adaptive nonlinear controller. Define the coolant temperature tracking error as

$$e_h = T_h - T_{h,d} \quad (13)$$

The T_m term in Eq. (1) may be replaced by Eq. (2), and divide both side of the equation by $M_{w,e}c_{p,w}$, the coolant temperature change at the engine outlet can be simplified as

$$\frac{dT_h}{dt} = \frac{1}{M_{w,e}} \left[\frac{Q_e}{c_{p,w}} - K_v \dot{m}_w (T_h - T_c) \right] \quad (14)$$

Considering that the waste heat of combustion, Q_e , is not measurable, an estimation of engine heat generation, $\hat{Q}_e$, is applied in the controller design. So, define the heat generation estimation error as

$$\tilde{Q}_e = \hat{Q}_e - Q_e \quad (15)$$

The desired target coolant temperature, $T_{h,d}$, is constant. The tracking error dynamic change may be evaluated by computing the time derivative of Eq. (13) so that

$$\dot{e}_h = \dot{T}_h - \dot{T}_{h,d} = \frac{dT_h}{dt} \quad (16)$$

To stabilize the coolant temperature, the control objective can be stated as

$$e_h \leq \varepsilon \text{ as } t \rightarrow \infty \quad (17)$$

To realize this control purpose, a Lyapunov based backstepping adaptive controller may be developed to regulate the pump speed by determining the coolant mass flow rate, $\dot{m}_w$. First, define the second term in Eq. (3) as

$$U_1 = \frac{K_v \dot{m}_w}{M_{w,e}} (T_h - T_c) \quad (18)$$

A Lyapunov function, V , can be selected as

$$V = \frac{1}{2} (e_h^2 + \tilde{Q}_e^2) \quad (19)$$

The Lyapunov function must be positive definite. If its time derivative is negative definite, the coolant temperature tracking error and the heat generation estimation error converges to zero eventually [15]. Assuming that the change of Q_e is slow, then the Lyapunov function time derivative can be written as

$$\dot{V} = e_h \dot{e}_h + \tilde{Q}_e \dot{\tilde{Q}}_e \quad (20)$$

A control law, U_1 , may be proposed with the form

$$U_1 = k_e e_h + \frac{\hat{Q}_e}{M_{w,e} c_{p,w}} \quad (21)$$

Replace the second term in Eq. (14) by Eq. (21), so that the dynamic change of the coolant temperature tracking error becomes

$$\dot{e}_h = -k_e e_h - \frac{\tilde{Q}_e}{M_{w,e} c_{p,w}} \quad (22)$$

Now design the heat generation estimate as

$$\dot{\hat{Q}}_e = \frac{e_h}{M_{w,e} c_{p,w}} \quad (23)$$

In the next step, substitute Eqs. (22) and (23) into Eq. (20), the derivative of the Lyapunov function can be written as

$$\dot{V} = -k_e e_h^2 + \tilde{Q}_e \left(\dot{\hat{Q}}_e - \frac{e_h}{M_{w,e} c_{p,w}} \right) = -k_e e_h^2 \quad (24)$$

The Lyapunov function converges to zero with a negative definite time derivative. This indicates that the coolant tracking error, e_h , can be stabilized by applying the proposed input U_1 in Eq. (21). The desired coolant mass flow rate in the engine can be solved with Eqs. (19) and (22) as