Nonlinear-Control Strategy for Advanced Vehicle Thermal-Management Systems

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Abstract—Advanced thermal-management systems for internal combustion engines can improve coolant-temperature regulation and servomotor power consumption by better regulating the combustion process with multiple computer-controlled electromechanical components. The traditional thermostat valve, coolant pump, and clutch-driven radiator fan are upgraded with servomotor actuators. When the system components function harmoniously, desired thermal conditions can be accomplished in a power-efficient manner. In this paper, a comprehensive nonlinear-control architecture is proposed for transient-temperature tracking. An experimental system has been fabricated and assembled which features a variable-position smart valve, variable-speed electric water pump, variable-speed electric radiator fan, engine block, and various sensors. In the configured system, the steam-based heat exchanger emulates the heat generated by the engine’s combustion process. Representative numerical and experimental results are discussed to demonstrate the functionality of the thermal-management system in accurately tracking the prescribed temperature profiles and minimizing electrical power consumption.

Index Terms—Control systems, cooling, instrumentation, road vehicles, testing.

I. INTRODUCTION

INTERNAL-COMBUSTION-ENGINE active thermal-management systems offer enhanced coolant-temperature tracking during transient and steady-state operation. Although the conventional automotive cooling system has proven to be satisfactory for many decades, servomotor-controlled cooling components have the potential to reduce the fuel consumption, parasitic losses, and tailpipe emissions [1]. Advanced automotive cooling systems replace the conventional wax thermostat valve with a variable-position smart valve and replace the mechanical water pump and radiator fan with electric-and/or hydraulic-driven actuators [7]. This later action decouples the water pump and radiator fan from the engine crankshaft. Hence, the problem of having over/undercooling, due to the mechanical coupling, is solved, as well as parasitic losses reduced which arose from operating mechanical components at high rotational speeds [3].

An assessment of thermal-management strategies for large on-highway trucks and high-efficiency vehicles has been reported in [20]. In [5], the benefits of engine cooling with fuel economy and emissions over the federal test procedure drive cycle on a dual-voltage 42–12-V minivan have been studied. Cho et al. [6] investigated a controllable electric water pump in a class-three medium-duty diesel-engine truck. It was shown that the radiator size could be reduced by replacing the mechanical pump with an electrical one. Chalgren and Allen [2] and Chalgren and Traczyk [4] improved the temperature control, while decreasing parasitic losses, by replacing the conventional cooling system of a light-duty diesel truck with an electric cooling system.

To create an efficient automotive thermal-management system, the vehicle’s cooling-system behavior and transient response must be analyzed. Wagner et al. [17]–[19] pursued a lumped-parameter modeling approach and presented multinode thermal models which estimated internal engine temperature. Eberth et al. [9] created a mathematical model to analytically predict the dynamic behavior of a 4.6-L spark-ignition engine. To accompany the mathematical model, analytical/empirical descriptions were developed to describe the smart cooling system components. Henry et al. [10] presented a simulation model of powertrain cooling systems for ground vehicles. The model was validated against test results, which featured basic system components (e.g., radiator, water pump, surge (return) tank, hoses and pipes, and engine thermal load).

A multiple-node lumped-parameter-based thermal network with a suite of mathematical models, describing controllable electromechanical actuators, was introduced in [16] to support controller studies. The proposed simplified cooling system used electrical immersion heaters to emulate the engine’s combustion process and servomotor actuators, with nonlinear-control algorithms, to regulate the temperature. In their experiments, the water pump and radiator fan were set to run at constant speeds, while the smart thermostat valve was controlled to track coolant-temperature setpoints. In [8], three cooling-control schemes (e.g., closed loop, model based, and mixed) were tested and compared against a traditional “thermostat-based” controller. Page et al. [11] conducted experimental tests on a medium-sized tactical vehicle that was equipped with...
an intelligent thermal-management system. They investigated improvements in the engine’s peak fuel consumption and thermal-operating conditions. Finally, Redfield et al. [13] operated a class-eight tractor at highway speeds to study potential energy saving and demonstrated engine cooling to with \( \pm 3 \) °C of a set-point value.

In this paper, nonlinear-control strategies are presented to actively regulate the coolant temperature in internal combustion engines. An advanced thermal-management system has been implemented on a laboratory test bench that featured a smart thermostat valve, variable-speed electric water pump, and a steam-based heat exchanger to emulate the combustion-heating process. The proposed backstepping-robust-control strategy, selected due to the nonlinear mathematical formulation of the system and to accommodate disturbances and uncertainties, has been verified by simulation techniques and validated by experimental testing. In Section II, a set of mathematical models is presented to describe the automotive cooling components and thermal-system dynamics. A nonlinear-tracking-control strategy is introduced in Section III. Section IV presents the experimental test bench, while Section V introduces numerical and experimental results. The conclusion is contained in Section VI.

II. AUTOMOTIVE THERMAL-MANAGEMENT MODELS

A suite of mathematical models will be presented to describe the dynamic behavior of the advanced cooling system. The system components include a 6.0-L diesel engine with a steam-based heat exchanger to emulate the combustion heat, a three-way smart valve, a variable-speed electric water pump, and a radiator with a variable-speed electric fan.

A. Cooling-System Thermal Descriptions

A reduced-order two-node lumped-parameter thermal model (refer to Fig. 1) describes the cooling system’s transient response and minimizes the computational burden for in-vehicle implementation. The engine block and radiator behavior can be described by

\[
C_e \dot{T}_e = Q_{in} - C_{pc} \dot{m}_r (T_e - T_i) \tag{1}
\]

\[
C_e \dot{T}_r = -Q_o + C_{pc} \dot{m}_r (T_e - T_r) - \varepsilon C_{pa} \dot{m}_f (T_c - T_\infty). \tag{2}
\]

The variables \( Q_{in}(t) \) and \( Q_o(t) \) represent the input heat generated by the combustion process and the radiator-heat loss due to uncontrollable air flow, respectively. An adjustable double-pass steam-based heat exchanger delivers the emulated heat of combustion at a maximum of 55 kW in a controllable and repeatable manner. In an actual vehicle, the combustion process will generate this heat, which is transferred to the coolant through the block’s water jacket.

For a three-way servo-driven thermostat valve, the radiator coolant mass flow rate \( \dot{m}_r(t) \) is based on the pump flow rate and normalized valve position, as \( \dot{m}_r = H \dot{\bar{m}}_c \), where the variable \( H(t) \) satisfies the condition \( 0 \leq H \leq 1 \). Note that \( H = 1(0) \) corresponds to a fully closed (open) valve position and coolant flow through the radiator (bypass) loop. To facilitate the controller-design process, three assumptions are imposed.

A1) The signals \( Q_{in}(t) \) and \( Q_o(t) \) always remain positive in (1) and (2) (i.e., \( Q_{in}(t), Q_o(t) \geq 0 \)). Furthermore, the signals \( Q_{in}(t), Q_{in}(t), Q_{in}(t) \), and \( Q_o(t) \) remain bounded at all times, such that \( Q_{in}(t), \dot{Q}_{in}(t), \ddot{Q}_{in}(t), Q_o(t) \in L_\infty \).

A2) The surrounding ambient temperature \( T_\infty(t) \) is uniform and satisfies \( T_e(t) - T_\infty(t) \geq \varepsilon_1, \forall t \geq 0 \), where \( \varepsilon_1 \in \mathbb{R}^+ \) is a constant.

A3) The engine-block and radiator temperatures satisfy the condition \( T_e(t) - T_i(t) \geq \varepsilon_2, \forall t \geq 0 \), where \( \varepsilon_2 \in \mathbb{R}^+ \) is a constant. Furthermore, \( T_e(0) \geq T_i(0) \) to facilitate the boundedness of signal argument.

The final assumption allows the engine and radiator to initially be at the same temperature (e.g., cold start). The unlikely case of \( T_e(0) < T_i(0) \) is not considered.
B. Variable-Position Smart Valve

A dc servo-motor has been actuated in both directions to operate the multiposition smart thermostat valve. The compact motor, with integrated external potentiometer for position feedback, is attached to a worm-gear assembly that is connected to the valve’s piston. The governing equation for the motor’s armature current \( i_{av}(t) \) can be written as

\[
\frac{di_{av}}{dt} = \frac{1}{L_{av}} \left( V_p - R_{av}i_{av} - K_{bv}\frac{d\omega_v}{dt} \right). \tag{3}
\]

The motor’s angular velocity \( d\theta_v(t)/dt \) may be computed as

\[
\frac{d^2\theta_v}{dt^2} = \frac{1}{J_v} \left( -b_v \frac{d\omega_v}{dt} + K_{nv}i_{av} + 0.5 dN \right) \cdot \left( A_p\Delta P + c \text{ sgn} \left( \frac{dh}{dt} \right) \right). \tag{4}
\]

Note that the motor is operated by a high-gain proportional control to reduce the position error and to speed up the overall piston response.

C. Variable-Speed Water Pump

A computer-controlled electric motor operates the high-capacity centrifugal water pump. The motor’s armature current \( i_{ap}(t) \) can be described as

\[
\frac{di_{ap}}{dt} = \frac{1}{L_{ap}} \left( V_p - R_{ap}i_{ap} - K_{bp}\omega_p \right) \tag{5}
\]

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

\[
\frac{d\omega_p}{dt} = \frac{1}{J_p} \left( - (b_p + R_i V_o^2) \omega_p + K_{mp}i_{ap} \right). \tag{6}
\]

The coolant mass flow rate for a centrifugal water pump depends on the coolant density, shaft speed, system geometry, and pump configuration. The mass flow rate may be computed as \( \dot{m}_c = \rho_c(2\pi\nu L) \), where \( \nu = (r\omega_p) \tan \beta \). It is assumed that the inlet radiator velocity \( \nu(t) \) is equal to the inlet fluid velocity, and that the flow enters normal to the impeller.

D. Variable-Speed Radiator Fan

A cross-flow heat exchanger and a dc servo-motor-driven fan form the radiator assembly. The electric motor directly drives a multiblade fan that pulls the surrounding air through the radiator assembly. The air mass flow rate going through the radiator is affected directly by the fan’s rotational speed \( \omega_f(t) \) so that

\[
\frac{d\omega_f}{dt} = \frac{1}{J_f} \left( -b_f\omega_f + K_{mf}\omega_f^3 - \rho_a A_f R_i V_f^2 \right) \tag{7}
\]

where \( V_f = \left[ \left( K_{mf}\omega_f^3 - \rho_a A_f R_i V_f^2 \right) \right]^{0.3} \). The corresponding air mass flow rate is written as \( \dot{m}_f = \beta_\rho A_f V_f + \dot{m}_{ram} \). The last term denotes the ram-air mass-flow-rate effect due to vehicle speed or ambient-wind velocity. The fan motor’s armature current \( i_{af}(t) \) can be described as

\[
\frac{di_{af}}{dt} = \frac{1}{L_{af}} \left( V_f - R_{af}i_{af} - K_{bf}\omega_f \right). \tag{8}
\]

Note that a voltage-divider circuit has been inserted into the experimental system to measure the current drawn by the fan and estimate the power consumed.

III. THERMAL-SYSTEM-CONTROL DESIGN

A Lyapunov-based nonlinear-control algorithm will be presented to maintain a desired engine-block temperature \( T_{ed}(t) \). The controller’s main objective is to precisely track engine-temperature setpoints while compensating for system uncertainties [i.e., combustion-process input heat \( Q_{in}(t) \), radiator-heat loss, \( Q_r(t) \)] by harmoniously controlling the system actuators. Although other linear- and nonlinear-control algorithms may be formulated, this particular control strategy demonstrated outstanding disturbance-rejection qualities.

Referring to Fig. 1, the system servo-actuators are a three-way smart valve, a water pump, and a radiator fan. Another important objective is to reduce the electric power consumed by these actuators \( P_{st}(t) \). The main concern is pointed toward the fact that the radiator fan consumes the most power of all cooling system components followed by the pump. It is also important to point out that, in (1) and (2), the signals \( T_{ed}(t), T_r(t), \) and \( T_{\infty}(t) \) can be measured by either thermocouples or thermistors, and the system parameters \( C_{pr}, C_{pa}, C_e, C_r, \) and \( \varepsilon \) are assumed to be constant and fully known.

A. Backstepping-Robust-Control Objective

The control objective is to ensure that the actual temperatures of the engine \( T_e(t) \) and the radiator \( T_r(t) \) track the desired trajectories \( T_{ed}(t) \) and \( T_{vr}(t) \)

\[
|T_{ed}(t) - T_e(t)| \leq \varepsilon_e, \quad |T_r(t) - T_{vr}(t)| \leq \varepsilon_r \quad t \to \infty \tag{9}
\]

while compensating for the system-variable uncertainties \( Q_{in}(t) \) and \( Q_r(t) \), where \( \varepsilon_e \) and \( \varepsilon_r \) are positive constants.

A4 The engine-temperature profiles are always bounded and chosen, such that their first three time derivatives remain bounded at all times [i.e., \( T_{ed}(t), T_{ed}(t), T_{ed}(t) \), and \( T_{ed}(t) \) \( \in L_\infty \)]. Furthermore, \( T_{ed}(t) \geq T_{\infty}(t) \) at all times.

Remark 1: Although it is unlikely that the desired radiator-temperature setpoint \( T_{vr}(t) \) is required (or known) by the automotive engineer, it will be shown that the radiator setpoint can indirectly be designed based on the engine’s thermal conditions and commutation strategy (refer to Remark 2).

To facilitate the controller’s development and quantify the temperature-tracking-control objective, the tracking-error signals \( \varepsilon(t) \) and \( \eta(t) \) are defined as

\[
\varepsilon \triangleq T_e - T_{ed}, \quad \eta \triangleq T_r - T_{vr}. \tag{10}
\]
By adding and subtracting $MT_{vr}(t)$ to (1) and expanding the variables $M = C_{pc}m_o$ and $\dot{m}_r = m_o + \bar{m} = H_o \dot{m}_e + \dot{H} \dot{m}_e$, the engine and radiator dynamics can be rewritten as

$$C_e \ddot{T}_e = Q_{in} - M(T_e - T_{vr}) - C_{pc}\bar{m}(T_e - T_T) + M \eta \tag{11}$$
$$C_r \dot{T}_r = -Q_o + C_{pc}(m_o + \bar{m})(T_e - T_T) - \varepsilon C_{pa} \dot{m}_t(T_e - T_\infty) \tag{12}$$

where $\eta(t)$ was introduced in (10), and $m_o$ and $H_o$ are positive design constants.

### B. Closed-Loop Error-System Development and Controller Formulation

The open-loop error system can be analyzed by taking the first time derivative of both expressions in (10) and then multiplying both sides of the resulting equations by $C_e$ and $C_r$, respectively. Thus, the system dynamics described in (11) and (12) can be substituted and then reformatted to realize

$$C_e \dot{e} = C_e \dot{T}_{ed} - Q_{in} + M(T_e - T_{vro}) - u_e - M \eta \tag{13}$$
$$C_r \dot{\eta} = M(T_e - T_T) - Q_o + u_r - C_r \dot{T}_{vr} \tag{14}$$

In these expressions, (10) was utilized, as well as $T_{vr} \overset{\Delta}{=} T_{vro} + T_{vr}$, $u_e = M \dot{T}_{vr} - C_{pc} \bar{m}(T_e - T_T)$, and $u_r = C_{pc} \bar{m}(T_e - T_T) - \varepsilon C_{pa} \dot{m}_t(T_e - T_\infty)$. The parameter $T_{vro}$ is a positive design constant.

**Remark 2:** The control inputs $\dot{m}_t(t)$, $\dot{T}_{vr}(t)$, and $\dot{m}_e(t)$ are unipolar. Hence, commutation strategies are designed to implement the bipolar inputs $u_e(t)$ and $u_r(t)$ as

$$\dot{m}_t \overset{\Delta}{=} u_e \left[ \frac{\text{sgn}(u_e)}{2C_{pc}(T_e - T_T)} - 1 \right]$$
$$\dot{T}_{vr} \overset{\Delta}{=} \frac{u_r}{2M} \left[ 1 + \frac{\text{sgn}(u_e)}{2C_{pc}(T_e - T_T)} \right]$$
$$\dot{m}_e \overset{\Delta}{=} \frac{F \left[ 1 + \frac{\text{sgn}(F)}{2C_{pa}(T_e - T_\infty)} \right]}{2\varepsilon C_{pa}(T_e - T_\infty)} \tag{15}$$

where $F \overset{\Delta}{=} C_{pc} \bar{m}(T_e - T_T) - u_r$. The control input $\dot{m}_e(t)$ is obtained from (15) after $\dot{m}_t(t)$ is computed. From these definitions, it is clear that if $u_e(t)$, $u_r(t) \in L_\infty \forall t \geq 0$, then $\dot{m}_t(t)$, $\dot{T}_{vr}(t)$, and $\dot{m}_e(t) \in L_\infty \forall t \geq 0$.

To facilitate the subsequent analysis, the expressions in (13) and (14) are rewritten as

$$C_e \dot{e} = \bar{N}_e + N_{ed} - u_e - M \eta$$
$$C_r \dot{\eta} = \bar{N}_r + N_{rd} + u_r - C_r \ddot{T}_{vr} \tag{16}$$

where the auxiliary signals $\bar{N}_e(T_e, t)$ and $\bar{N}_r(T_e, T_{vr}, t)$ are defined as

$$\bar{N}_e \overset{\Delta}{=} N_e - N_{ed}, \quad \bar{N}_r \overset{\Delta}{=} N_r - N_{rd} \tag{17}$$

Furthermore, the signals $N_e(T_e, t)$ and $N_r(T_r, T_{vr}, t)$ are defined as

$$N_e \overset{\Delta}{=} C_e \dot{T}_{ed} - Q_{in} + M(T_e - T_{vro})$$
$$N_r \overset{\Delta}{=} M(T_e - T_T) - Q_o \tag{18}$$

with both $N_{ed}(t)$ and $N_{rd}(t)$ represented as

$$N_{ed} \overset{\Delta}{=} N_e|_{T_e = T_{ed}} = C_e \ddot{T}_{ed} - Q_{in} + M(T_{ed} - T_{vro})$$
$$N_{rd} \overset{\Delta}{=} N_r|_{T_r = T_{rd}, T_e = T_{vr}} = M(T_{ed} - T_{vr}) - Q_o. \tag{19}$$

Based on (17)–(19), the control laws $u_e(t)$ and $u_r(t)$ introduced in (16) are designed as

$$u_e = K_e e, \quad u_r = -K_r \eta + \bar{u}_e \tag{20}$$

where $\bar{u}_e(t)$ is selected as

$$\bar{u}_e = \begin{cases} 0 & \text{if } \forall u_e \in (-\infty, 0) \\ \left(2M - K_e \frac{C_{pc}^2}{C_{pc}^2 + C_{pa} K^2} e - C_{pc} K e \right) & \forall u_e \in [0, \infty) \end{cases} \tag{21}$$

Knowledge of $u_e(t)$ and $u_r(t)$, based on (20) and (21), allows the commutation relationships of (15) to be calculated, which provides $\dot{m}_e(t)$ and $\dot{m}_t(t)$. Finally, the voltage signals for the pump and fan are prescribed using $\dot{m}_e(t)$ and $\dot{m}_t(t)$ with an a priori empirical relationships.

### C. Stability Analysis

A Lyapunov-based stability analysis guarantees that the advanced thermal-management system will be stable when applying the control laws introduced in (20) and (21).

**Theorem 1:** The controller given in (20) and (21) ensures the following: 1) All closed-loop signals stay bounded all the time, and 2) tracking is uniformly ultimately bounded in the sense that $(|e(t)| \leq \varepsilon_e, |\eta(t)| \leq \varepsilon_\eta$ as $t \to \infty$).

**Proof:** See Salah [14] for the complete Lyapunov-based stability analysis.
D. Normal-Radiator-Operation Strategy

The electric radiator fan must be harmoniously controlled with the other thermal-management-system actuators to ensure proper power consumption. From the backstepping-robust-control strategy, a virtual reference for the radiator temperature $T_{vr}(t)$ is designed to facilitate the radiator-fan control law (refer to Remark 1). A tracking error signal $\eta(t)$ is introduced for the radiator temperature. Based on the radiator’s mathematical description in (2), the radiator may operate normally, as a heat exchanger, if the effort of the radiator fan $\varepsilon C_{pa} \dot{m}_f (T_e - T_\infty)$, denoted by $u_r(t)$ in (22), is set to equal the effort produced by the water pump $C_{pc} \dot{m}_r (T_e - T_r)$, denoted by $u_e(t)$ in (23). Therefore, the control input $u_e(t)$ provides the signals $\dot{m}_r(t)$ and $\dot{m}_f(t)$.

To derive the operating strategy, the system dynamics (1) and (2) can be written as

$$C_e \dot{T}_e = Q_{in} - u_e$$
$$C_t \dot{T}_t = -Q_o + u_e - u_f. \tag{22}$$

If $u_e(t)$ is selected so that it equals $u_e(t)$, then the radiator operates normally. The control input $u_e(t)$ can be designed, utilizing a Lyapunov-based analysis, to robustly regulate the temperature of the engine block as

$$u_e = -(K_e + \alpha_e) [e - e_o]$$
$$- \int_{t_o}^{t} [\alpha_e (K_e + \alpha_e) e(\tau) + \rho_s \text{sgn}(e(\tau))]d\tau \tag{24}$$

where the last term in (24) compensates for the variable unmeasurable input heat $Q_{in}(t)$ (refer to [16] for more details on this robust-control-design method).

**Remark 3:** The control input $\dot{m}_r(t)$ is unipolar. Again, a commutation strategy may be designed to implement the bipolar input $u_e(t)$ as

$$\dot{m}_r = \frac{u_e [1 + \text{sgn}(u_e)]}{2C_{pc}(T_e - T_r)}. \tag{25}$$

From this definition, if $u_e(t) \in L_\infty \ \forall t \geq 0$, then $\dot{m}_r(t) \in L_\infty \ \forall t \geq 0$. The choice of the valve position and water pump’s
Fig. 4. First experimental test for the backstepping robust controller with emulated vehicle speed of 20 km/h and $Q_{in} = 35$ kW. (a) Experimental engine and radiator temperatures with a desired engine temperature $T_{ed} = 363$ K. (b) Experimental engine-temperature tracking error. (c) Experimental coolant mass flow rate through the pump. (d) Experimental air mass flow rate through the radiator fan.

speed to produce the required control input $\dot{m}_f(t)$, defined in (25), can be determined based on energy-optimization issues. Furthermore, this allows $\dot{m}_f(t)$ to approach zero without stagnation of the coolant, since $\dot{m}_c = H\dot{m}_c$, and $0 \leq H(t) \leq 1$. Another commutation strategy is needed to compute the unipolar control input $\dot{m}_f(t)$ so that

$$\dot{m}_f = \frac{u_r [1 + sgn(u_r)]}{2\pi C_{pa}(T_e - T_\infty)} \tag{26}$$

where $u_r(t) = u_{re}(t)$. From this definition, if $u_r(t) \in L_\infty \forall t \geq 0$, then $\dot{m}_f(t) \in L_\infty \forall t \geq 0$.

IV. THERMAL TEST BENCH

An experimental test bench (refer to Fig. 2) has been fabricated to demonstrate the proposed advanced thermal-management-system controller design. The assembled test bench offers a flexible, rapid, repeatable, and safe testing environment. Clemson University facility-generated steam is utilized to rapidly heat the coolant circulating within the cooling system via a two-pass shell-and-tube heat exchanger. The heated coolant is then routed through a 6.0-L diesel engine block to emulate the combustion-process heat. From the engine block, the coolant flows to a three-way smart valve and, then, either through the bypass or radiator to the water pump to close the loop. The thermal response of the engine block to the adjustable externally applied heat source emulates the heat-transfer process between the combustion gases, cylinder wall, and water jacket in an actual operating engine. As shown in Fig. 1, the system sensors include three type-J thermocouples (e.g., $T_1 = $ engine temperature, $T_2 = $ radiator temperature, and $T_3 = $ ambient temperature), two mass flow meters (e.g., $M_1 = $ coolant mass flow meter and $M_2 = $ air mass flow meter), and electric voltage and current measurements (e.g., $P_1 = $ valve power consumed, $P_2 = $ pump power consumed, and $P_3 = $ fan power consumed).

The steam bench can provide up to 55 kW of energy. High-pressure saturated steam (412 kPa) is routed from the campus facilities plant to the steam test bench, where a pressure regulator reduces the steam pressure to 172 kPa before it enters the low-pressure filter. The low-pressure saturated steam is
Fig. 5. Second experimental test scenario for the backstepping robust controller, where the input heat and ram-air disturbance vary with time. (a) Experimental engine and radiator temperatures with a desired engine temperature $T_{ed} = 363$ K. (b) Experimental engine-temperature tracking error. (c) Experimental coolant mass flow rate through the pump. (d) Experimental air mass flow rate through the radiator fan.

then routed to the double-pass steam heat exchanger to heat the system’s coolant. The amount of energy transferred to the system is controlled by the main valve mounted on the heat exchanger. The mass flow rate of condensate is proportional to the energy transfer to the circulating coolant. Condensed steam may be collected and measured to calculate the rate of energy transfer. From steam tables, the enthalpy of condensation can be acquired. To facilitate the analysis, pure saturated steam and condensate at approximately $T = 100$ °C determines the enthalpy of condensation. Baseline testing was performed to determine the average energy transferred to the coolant at various steam-control-valve positions. The coolant temperatures were initialized at $T_e = 67$ °C before measuring the condensate. Each test was executed for different time periods.

V. NUMERICAL AND EXPERIMENTAL RESULTS

In this section, the numerical and experimental results are presented to verify and validate the mathematical models and control design. First, a set of MATLAB/Simulink simulations have been created and executed to evaluate the backstepping-robust-control design and the normal-radiator-operation strategy. The proposed thermal-model parameters used in the simulations are $C_e = 17.14$ kJ/K, $C_r = 8.36$ kJ/K, $C_{pc} = 4.18$ kJ/kg·K, $C_{pa} = 1$ kJ/kg·K, $\varepsilon = 0.6$, and $T_{\infty}(t) = 293$ K. Second, a set of experimental tests has been conducted on the steam-based thermal test bench to investigate the control-design and operation strategies.

A. Backstepping Robust Control

A numerical simulation of the backstepping-robust-control strategy, introduced in Section III, has been performed on the system dynamics (1) and (2) to demonstrate the performance of the proposed controller in (20) and (21). For added reality, band-limited white noise was added to the plant using a MATLAB block (noise power = 0.1). To simplify the subsequent analysis, a fixed smart-valve position of $H = 1$ (e.g., fully closed for 100% radiator flow) has been applied to investigate the water pump’s ability to regulate the engine.
temperature. An external ram-air disturbance was introduced to emulate a vehicle traveling at 20 km/h with varying input heat of $Q_{in} = [50, 40, 20, 35 \text{ kW}]$, as shown in Fig. 3. The initial simulation conditions were $T_e(0) = 350 \text{ K}$ and $T_r(0) = 340 \text{ K}$. The control-design constants are $T_{vro} = 356 \text{ K}$ and $m_o = 0.4$. Similarly, the controller gains were selected as $K_e = 40$ and $K_r = 0.005$. The desired engine temperature varied as $T_{ed} = 363 + \sin(0.05 t) \text{ K}$. This time-varying setpoint allows the controller’s tracking performance to be studied.

In Fig. 3(a), the backstepping robust controller readily handles the heat fluctuations in the system at $t = [200, 500, 800 \text{ s}]$. For instance, when $Q_{in} = 50 \text{ kW}$ (heavy thermal load) is applied from $0 \leq t \leq 200 \text{ s}$, as well as when $Q_{in} = 20 \text{ kW}$ (light thermal load) is applied at $500 \leq t \leq 800 \text{ s}$, the controller is able to maintain a maximum absolute-value tracking error of 1.5 K. Under the presented operating condition, the error in Fig. 3(b) fluctuates between $-0.4 \text{ K}$ and $-1.5 \text{ K}$. In Fig. 3(c) and (d), the coolant pump (maximum flow limit of 2.6 kg/s) works harder than the radiator fan, which is ideal for power minimization.

**Remark 4:** The error fluctuation in Fig. 3(b) is quite good when compared to the overall amount of heat handled by the cooling system components.

Two scenarios have been implemented to investigate the controller’s performance on the experimental test bench. The first case applies a fixed input heat of $Q_{in} = 35 \text{ kW}$ and a ram-air disturbance which emulates a vehicle traveling at 20 km/h, as shown in Fig. 4. From Fig. 4(b), the controller can achieve a steady-state absolute-value temperature-tracking error of 0.7 K. In Fig. 4(c) and (d), the water pump works harder than the radiator fan, which, again, is ideal for power minimization. Note that the water pump reaches its maximum mass flow rate of 2.6 kg/s and that the fan runs at 73% of its maximum speed (e.g., maximum air mass flow rate is 1.16 kg/s). The fluctuation in the coolant and air mass flow rates during $0 \leq t \leq 400 \text{ s}$ [refer to Fig. 4(c) and (d)] is due to the fluctuation in the actual radiator temperature about the radiator-temperature virtual reference $T_{vro} = 356 \text{ K}$, as shown in Fig. 4(a).

The second scenario varies both the input heat and disturbance. Specifically, $Q_{in}(t)$ changes from 50 to 35 kW at
Fig. 7. First experimental test results for the normal-radiator-operation controller with emulated vehicle speed of 20 km/h and $Q_{\text{in}} = 35$ kW. (a) Experimental engine and radiator temperatures with a desired engine temperature $T_{\text{ed}} = 363$ K. (b) Experimental engine-temperature tracking error. (c) Experimental coolant mass flow rate through the pump. (d) Experimental air mass flow rate through the radiator fan.

t = 200 s, while $Q_{\text{in}}(t)$ varies from 20 to 40 to 20 km/h at $t = 400$ and 700 s (refer to Fig. 5). From Fig. 5(b), it is clear that the proposed control strategy handles the input heat and ram-air variations nicely. During the ram-air variation between 550 and 750 s, the temperature error fluctuates within 1 K due to the oscillations in the water-pump and radiator-fan flow rates as per Fig. 5(c) and (d). This behavior may be attributed to the supplied ram air that causes the actual radiator temperature $T_r(t)$ to fluctuate about the radiator-temperature virtual reference $T_{\text{vrc}} = 356$ K in Fig. 5(a).

B. Normal-Radiator-Operation Strategy

The normal-radiator-operation strategy, introduced in Section III, has numerically been simulated using system dynamics (1) and (2) to investigate the robust-tracking-controller performance given in (24). The simulated thermal system’s parameters, initial simulation conditions, and desired engine temperature were equivalent to Section V-A. Again, a band-limited white noise was added to the plant using a MATLAB block with noise power $= 0.1$. A fixed 100% radiator-flow smart-valve position allows the water pump’s ability to regulate the engine temperature to be studied. The external ram air emulated a vehicle traveling at 20 km/h; the input heat was varied, as shown in Fig. 6 (e.g., $Q_{\text{in}} = [50, 40, 20, 35]$ kW). The control gains were set as $K_e = 10$, $\alpha_e = 0.005$, and $\rho_e = 0.1$. Although the normal radiator operation accommodated the heat variations in Fig. 6(a), its performance was inferior to the backstepping robust control. However, the normal radiator operation achieved less tracking error under the same operating condition when Figs. 3(b) and 6(b) are compared. In this case, the maximum-temperature-tracking error fluctuation was 1 K. In Fig. 6(c) and (d), the pump works harder than the fan, which is preferred for power minimization. Note that the power consumption is larger than that achieved by the backstepping robust controller [refer to Figs. 3(c) and (d) and 6(c) and (d)].

The same two experimental scenarios presented for the backstepping robust controller are now implemented for the normal-radiator-operation strategy on the thermal test bench. In the first scenario, a fixed input heat and ram-air disturbance ($Q_{\text{in}} = 35$ kW and 20 km/h vehicle speed) were...
Fig. 8. Second experimental test scenario for the normal-radiator-operation controller, where the input heat and ram-air disturbance vary with time. (a) Experimental engine and radiator temperatures with a desired engine temperature $T_{ed} = 363$ K. (b) Experimental engine-temperature tracking error. (c) Experimental coolant mass flow rate through the pump. (d) Experimental air mass flow rate through the radiator fan.

Table I

<table>
<thead>
<tr>
<th>Description</th>
<th>Error $e_{td}$ [°C]</th>
<th>Power $P_{d}$ [W]</th>
</tr>
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<tbody>
<tr>
<td>Backstepping robust control</td>
<td>Simulation 0.616</td>
<td>Experiment 0.695</td>
</tr>
<tr>
<td></td>
<td>Simulation 31.625</td>
<td>Experiment 33.231</td>
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<tr>
<td>Normal radiator operation strategy</td>
<td>0.105</td>
<td>0.175</td>
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<tr>
<td></td>
<td>38.052</td>
<td>38.699</td>
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<tr>
<td>Adaptive control</td>
<td>1.003</td>
<td>1.075</td>
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<td></td>
<td>37.497</td>
<td>37.968</td>
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<tr>
<td>Robust control</td>
<td>0.905</td>
<td>0.935</td>
</tr>
<tr>
<td></td>
<td>34.346</td>
<td>35.786</td>
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</table>

applied. In Fig. 7(a), the normal-radiator-operation overshoot and settling time are larger than the backstepping robust control [refer to Fig. 4(a)]. As shown in Fig. 7(b), an improved engine-temperature-tracking error was demonstrated but with greater power consumption in comparison to the backstepping robust control [refer to Fig. 4(b)]. Finally, the water pump operated continuously at its maximum as per Fig. 7(c).

For the second test scenario, the input heat and disturbance are both varied as previously described for the backstepping robust control. The normal radiator operation maintained the established control gains. In Fig. 8(b), the temperature error remains within a $\pm 0.4$-K neighborhood of zero despite variations in the input heat and ram air. Although the temperature-tracking error is quite good, this strategy does not minimize power consumption in comparison to the backstepping-robust-control strategy.

The simulation and experimental results are summarized in Table I to compare the controller strategies. To ensure uniform operating conditions, all reported data corresponds to the first-scenario thermal conditions. Furthermore, the controller gains, initial conditions, and temperature setpoints were maintained for both the simulation and experimental tests. Note that adaptive and robust controllers were also designed and
implemented [14] for comparison purposes. However, the
designs are not reported in this paper. For these two controllers,
the radiator-temperature setpoint was required, which may be
considered a weakness.

Overall, the normal-radiator-operation strategy was better
than the adaptive- and robust-control strategies. However,
it is not as good as the backstepping control when com-
pared in terms of power consumption, despite achieving less
temperature-tracking error. Therefore, the backstepping-robust-
control strategy is considered to be the best among all con-
trollers and operation strategies. The power measure is the
minimum, the heat-change handling is more satisfactory,
and a setpoint for the radiator temperature is not required. From
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VI. CONCLUSION

Advanced automotive thermal-management system can have
a positive impact on gasoline- and diesel-engine cooling sys-
tems. In this paper, a suit of servo-motor-based cooling sys-
tem components have been assembled and controlled using
a Lyapunov-based nonlinear-control technique. The control
algorithm has been investigated using both simulation and
experimental tests. Two detailed and two supplemental con-
trollers were applied to regulate the engine temperature. In
each instance, the controllers successfully maintained the engine
block to setpoint temperatures with small error percentages. It
has also been shown that the power consumed by the system ac-
tuators can be reduced. Overall, the findings demonstrated that
setpoint temperatures could be maintained satisfactory while
minimizing power consumption, which ultimately impacts fuel

REFERENCES

[1] C. Brace, H. Burnham-Slipper, R. Wijetunge, N. Vaughan, K. Wright, and
Heavy Duty 42/54 V Electric Powertrain Cooling System, SAE Tech.
Cooling Thermal Management System on a Dual Voltage 42 V-14 V
able electric coolant pump for fuel economy and cooling performance
improvement,” in Proc. ASME EMCE, Adv. Energy Syst. Division,
for an Optimized Engine Cooling Thermal Management*, SAE Tech. Paper
based approach;” in Proc. ASME Internal Combustion Engine Division,
of Automotive “Smart” Thermal Management System Architecture*, SAE
Validation, of Power Train Cooling System for a Truck Application*, SAE
21st Century—Improved Thermal Control & Fuel Economy in an Army
 bust control tracking for thermal management systems.” Clemson Univ.,
Clemson, SC, CRB Tech. Rep. CUCR/B/10/06/1#. [Online]. Available:
http://www.ces.clemson.edu/cece/crb/public/tr.htm
thermal management system: Nonlinear control and test,” *IEEE/ASME
ponents for Temperature Prediction and Fluid Flow Regulation*, SAE
[18] J. Wagner, M. Ghone, D. Dawson, and E. Marotta, *Coolant Flow Con-
 trol Strategies for Automotive Thermal Management Systems*, SAE Tech.
Pump Control for Engine Thermal Management Systems*, SAE Tech.

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