# Fuel Saving Potential of Optimal Engine Cooling System

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The engine cooling system in trucks is one of the main sources of parasite load. Thus optimal control of the engine thermal management system with the objective of minimizing energy consumption can substantially improve fuel efficiency. Existing methods on the engine thermal control system concentrate mainly on regulating the engine coolant temperature within a safety range. This paper explicitly calculates the energy consumption of the cooling system using the optimal control methods to decide the trajectories of the control values of the cooling system. During the optimal operation, the engine cooling system serves as another energy buffer to balance the engine workload in conventional trucks. To expose the maximal fuel saving potential of the optimal engine thermal control system, we apply dynamic programming in the investigation and the results are compared with a simple state feedback controller.

Topics / Energy efficient vehicles, Other related topics in vehicle control

## 1. INTRODUCTION

Fuel consumption contributes to more than 30% of operational cost in trucks [1]. Many methods are proposed to improve fuel efficiency in trucks. Besides the ongoing research on engine optimization, different new technologies have been introduced in the last decade. For example, powertrain hybridization has proved promising for improving fuel efficiency; yet, the hybridization process is usually very costly, thus the price for hybrid vehicles compared to conventional vehicles are higher. Another possibility to improve fuel efficiency is the optimal control of auxiliaries such as braking, cooling, power steering, etc. to manage their energy consumption [1], [2]. The energy consumption in auxiliaries can also be reduced by electrification, e.g. the conventional mechanical water pump and radiator fan can be substituted with electrical ones [3], [4].

One of the main methods for improving fuel efficiency is to manage the energy flow in the vehicle. Although energy management is a hot research topic in hybrid vehicles [5], it is not yet studied in conventional vehicles extensively. A way to optimize the energy flow is to manage different energy buffers in the vehicle such as battery, vehicle kinetic energy, coolant and engine temperature, cabin temperature, etc. Fundamentally, the only energy source in the vehicle is fuel; however, some subsystems of vehicle act as energy buffers, i.e., they can store some amount of energy when the engine workload is low. For example, the vehicle kinetic energy can be considered as an energy buffer, which collects free energy during downhill. The coolant and the engine itself can also be considered as an energy buffer, since their temperature rises when the engine is working, and the high temperature means energy in the system. In thermal management system (TMS), the components such as engine can store heat energy in one instance (in the engine block), and release it in another instant. This is similar to the battery operation. The idea of treating different sub systems in the vehicle as energy buffers and then managing them in a central controller has not been discussed in detail, to the knowledge of authors. However, optimal control of TMS is investigated in different works using other strategies which mainly emphasize on benefits of electrification of TMS and cannot be generalized for other subsystems [6]. Holistic energy buffer control in trucks can be a way to increase the fuel efficiency to considerable extents.

In this paper, the concept of energy buffer control in the conventional trucks is presented. The engine cooling system is then isolated from the other energy buffers, and the potential fuel saving of optimal control of TMS is investigated using dynamic programming (DP) for global optimization. The rationale behind the fuel consumption reduction is explained using the concept of energy buffer control in the conventional trucks. The results are compared with a simple state feedback controller, which represents a very simple controller being used in practice. Although improvement in fuel efficiency is rather limited, it shows the potential of fuel efficiency improvement of optimal energy buffer control.

This paper is organized as follow. A description of energy buffer control concept in conventional trucks is presented in section 2. In section 3, modeling of the truck and cooling system is explained. The simple controller and dynamic programming algorithm are described in section 4. In section 5, the results of simulations are presented and discussed. The paper is concluded in section 6 by a discussion and conclusion.

## 2. ENERGY BUFFER CONTROL

The concept of energy buffer control is well known in HEVs and other types of hybrid vehicles while it has not been discussed extensively in the context of conventional vehicles. In general, all of the subsystems in vehicle that can store energy in a period of time and release energy when needed can be considered as energy buffers. The stored energy can be in any form, electrical, thermal, kinetic, etc. In conventional vehicles, thermal system, AC system, vehicle mass and the battery are considered as energy buffers. Benefits of each of these systems as an energy buffer are explained here. Most of the benefits from optimal control of energy buffers are from preventing engine to work in inefficient region. It is also important to note that prediction of upcoming situation is necessary when the energy buffer control is considered in the way discussed in this paper.

### 2.1 Holistic view of energy buffer system

It is important to have a holistic view on how the energy flow between different subsystems is. A schematic of energy flow between different energy buffers is given in Figure 1; however, only the engine cooling system is considered in the control design and simulation in Sections III, IV, and V. The battery is not yet included in the model and control design.

### 2.2 Battery

Electrical energy required for different electric auxiliaries and subsystems in the vehicle is mainly provided by engine through the alternator. If in some cases the alternator cannot provide electrical current that is necessary for auxiliary subsystems, battery will be discharged to compensate the lack of energy from the alternator. Conversely if the energy provided by the alternator is more than the energy required by auxiliaries, the battery will be charged. This is beneficial in situations where a charging situation is followed by a discharging one. A clear example is where the vehicle is driven uphill followed by a downhill. In this case, the battery will be discharged in the uphill, which results in less torque demand from alternator that consequently prevents engine from operating in less efficient regions. The battery will be charged again in the downhill. This process can contribute to fuel efficiency.

### 2.3 Thermal system

The engine can be considered as a big storage of thermal energy. When the engine works, heat is produced and its temperature increases. The heat is carried away by coolant flow using the water pump, and then dissipated in the radiator. In situations where an upcoming uphill is predicted, TMS can cool down the engine before starting the uphill. This results in less torque demand from the engine during the uphill, which consequently prevents the engine from working in less efficient region during the uphill. Another example is when a sudden acceleration is predicted.

### 2.4 vehicle kinetic energy

The whole vehicle can be considered as storage for energy. Vehicle can be accelerated in a period of time and store energy as kinetic energy, and then releases it when necessary. A familiar example is when speed of vehicle is increased when an upcoming uphill is predicted in the near future. In this case, engine can work in the more efficient region during the uphill as a consequence of less torque demand in the uphill.

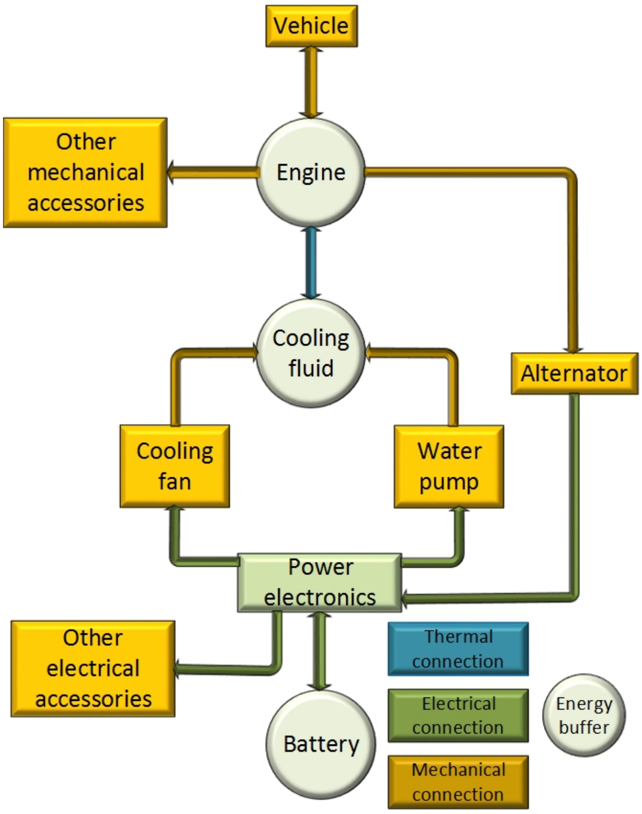


Figure 1. Schematic of energy flow between different energy buffers

## 3. MODELING

### 3.1 Vehicle specification

A conventional truck (4X2 tractor) FH 13 with electrified auxiliaries is used as the platform for analysis and simulations. Some of the vehicle specifications are presented in Table 1.

Table 1. Specification of truck

| Component | Description |
|---|---|
| Final drive ratio | 2.5 |
| Wheel radius | 520 mm |
| Frontal area: | 9.7 m^2 |
| Drag coefficient | 0.53 |
| Vehicle mass | 40000 kg |
| Diesel engine | 12800 CC |
| Maximum Power output | 460 hp, 1400-1900 rpm |
| Transmission | 12 speed Automatic gearbox |
| Maximum Torque output | 2300 Nm, 1400-1900 rpm |

### 3.2 Longitudinal dynamics

The model describing longitudinal dynamics of the vehicle is described here. The main outcome from the model is power requirement at each instant. It is assumed that the power required for all auxiliaries including water pump and radiator fan are directly supplied by engine. In the real truck, the auxiliaries are powered through alternator. It is also assumed that gear number is known and is considered as an input to the system.

$$P_{dem} = P_{rol} + P_{aer} + P_{slp} + P_{acc} + P_{aux} \tag{1}$$

where $P_{dem}$ is power demand from the engine, $P_{rol}$ is power required to overcome rolling resistance, $P_{aer}$ is power required to overcome aerodynamic force, $P_{slp}$ is power required for climbing slopes, $P_{acc}$ is power required for accelerating the vehicle and $P_{aux}$ is power required for auxiliaries. Each of demanded powers are calculated as,

$$P_{rol} = v \cdot f \cdot m \cdot g \cdot \cos\alpha \tag{2}$$

where $f$ is rolling resistant coefficient, $\alpha$ is road gradient angle and $v$ is vehicle speed.

$$P_{aer} = 0.5 \cdot \rho \cdot c_w \cdot A \cdot v \cdot (v + v_0)^2 \tag{3}$$

where $c_w$ is drag coefficient, $A$ is frontal area and $v_0$ is headwind speed.

$$P_{slp} = m \cdot g \cdot v \cdot \sin\alpha \tag{4}$$

$$P_{acc} = m \cdot v \cdot \frac{dv}{dt} \tag{5}$$

$$P_{aux} = P_p + P_f \tag{6}$$

$P_p$ and $P_f$ are demanded power for operating pump and fan respectively. Note that moment of inertia of engine, gearbox, shaft and wheels are neglected.

### 3.3 Engine cooling system modeling

The engine cooling system used in this study consists of an electric coolant pump, five identical electric fans and a 3-way valve. The valve is assumed to just have two states of open and close; however, during the simulations it is assumed that the thermostat is always open. The schematic of the engine cooling system is shown in Figure 2.

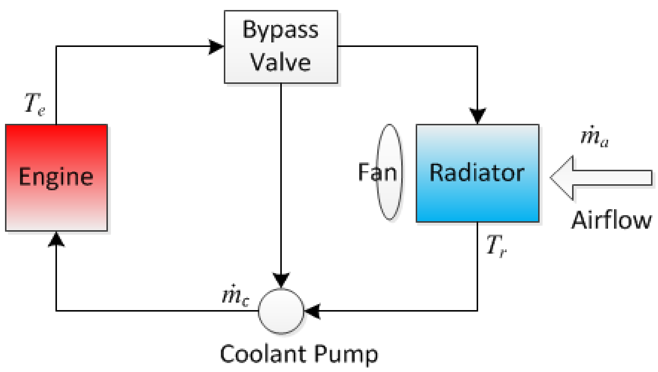


Figure 2. The Schematic of the Simplified Cooling System

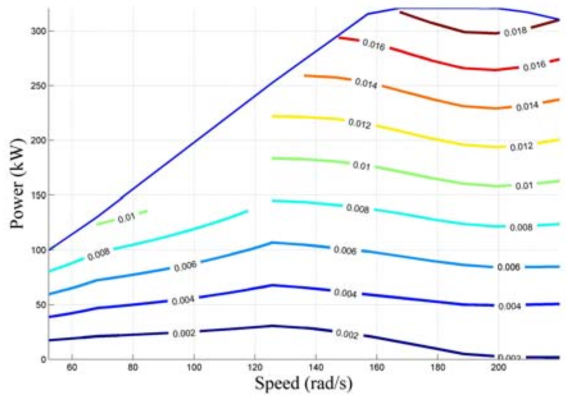


Figure 3. Engine map

The cooling system is modeled as a simple 2 states model. The state-space representation of the model is explained here. The control inputs are coolant mass flow rate through the pump and air mass flow rate through the radiator. The time delays in the system caused by, e.g., pipe length, are ignored. The heat transfer inside the engine is also ignored. It is assumed that the heat is only taken away from engine by coolant flow, so the effects of the air going around the engine, radiation and the heat taken away by exhaust are all ignored. The model has been used and verified in different papers [7], [3], [8]. Heat exchange equations are written using the second law of thermodynamics. The model can be described as:

$$\dot{T}_e = \frac{1}{C_e} \cdot Q_{in} - \frac{c_c}{C_e} \cdot \dot{m}_p \cdot (T_e - T_r) \tag{7}$$

$$\dot{T}_r = \frac{\dot{m}_p}{C_r} \cdot (T_e - T_r) - \frac{\varepsilon \cdot c_a}{C_r} \cdot \dot{m}_f \cdot (T_e - T_\infty) \tag{8}$$

$$x = \begin{bmatrix} T_e \\ T_r \end{bmatrix} \quad u = \begin{bmatrix} \dot{m}_p \\ \dot{m}_f \end{bmatrix} \quad v = [Q_{in}] \quad y = \begin{bmatrix} T_e \\ T_r \end{bmatrix}$$

where $T_e$ is engine temperature, $C_e$ is engine heat capacity, $c_c$ is coolant specific heat, $T_r$ is radiator temperature, $C_r$ is radiator heat capacity, $c_a$ is specific heat of air, $\varepsilon$ is constant, $\dot{m}_p$ is coolant flow in pump, $\dot{m}_f$ is air flow in radiator and $Q_{in}$ is generated heat in engine. $Q_{in}$ is calculated using the fuel rate map (Figure 3).

### 3.4 Electric water pump

The variable speed coolant pump is a centrifugal water pump driven by a DC motor. The equations describing the electric coolant pump are described here.

$$\frac{di_p}{dt} = \frac{1}{L_p}\left(V_p - R_p i_p - K_{EMF}\,\omega_p\right) \tag{9}$$

where $K_{EMF}$ is back EMF in pump (rad/s), $V_p$ is pump voltage, $i_p$ is pump current, $R_p$ is pump electrical resistance, $\omega_p$ is pump speed and

$$\frac{d\omega_p}{dt} = \frac{1}{J_p}\left(-\left(b_p + R_f V_0^2\right)\cdot \omega_p + K_{mp} i_p\right) \tag{10}$$

where $R_f(P,Q)$ is nonlinear fluid resistance and $V_0$ is fluid volume per radian. $b_p$ is pump system viscous damping, $K_{mp}$ is pump torque constant, $J_p$ is moment of inertia in pump. We also have

$$\dot{m}_p = \rho_c \cdot Q = \rho_c \cdot \left(2\pi r_1 b_1 V_r\right) \tag{11}$$

where $V_r$ is the inlet radial velocity component for the design point flow rate and is calculated as:

$$V_r = r_1 \omega_p \tan\beta \tag{12}$$

where $b_1$ is inlet impeller width and $\beta$ is inlet impeller angle. By neglecting the dynamics in the pump and working out the equation, we will have

$$P_p = \alpha \dot{m}_p \tag{13}$$

and

$$\alpha = \frac{\left(b_p + R_f V_0^2\right)}{\left(2\pi\rho_c b_1 r_1^2 \tan\beta_1\right) K_{mp}} \tag{14}$$

### 3.5 Electric radiator fan

Five identical electrical axial fans are used to cool the radiator. Each fan is driven by a DC motor with a PWM speed control. It is assumed that all of the fans are working with the same speed thus having similar power consumption and air flow rate. The fan air speed is a nonlinear function of the fan rotational. The formulation described here is based on [7].

Similar to the pump, equations for electrical fan are:

$$\frac{di_f}{dt} = \frac{1}{L_f}\left(V_f - R_f i_f - K_{EMF}\omega_f\right) \tag{15}$$

$$\frac{d\omega_f}{dt} = \frac{1}{J_f}\left(-B_r\omega_f + K_f i_f - \rho_a A_f R_f V_{af}^2\right) \tag{16}$$

$V_{af}$ is fan air speed. It can be calculated as:

$$V_{af} = \left(\left(\frac{K_f}{\eta_f \rho_f A_f}\right)\cdot i_f \cdot \omega_f\right)^{0.3} \tag{17}$$

$\eta_f$ is fan efficiency. The radiator air mass flow can also be calculated as:

$$\dot{m}_f = \beta_r \rho_a A_f V_{af} + \dot{m}_{ram} \tag{18}$$

$J_f$ and $B_r$ are equivalent moment of inertia and viscous damping of motor shaft and fan assembly. The power consumption for the fan can be calculated using the pressure difference in the two side of fan. The pressure increase can be calculated as

$$\Delta P_{fan} = P_{fan,out} - P_{fan,in} \tag{19}$$

And the power consumption by fan is

$$P_{fan,1} = \frac{\dot{m}_f \, \Delta P_{fan}}{\rho_a \cdot \eta_{fan} \cdot \left(\dot{m}_f \, \Delta P_{fan}\right)} \tag{20}$$

where $\eta_{fan}$ is the combined efficiency of fan and DC motor. the efficiency can be calculated using a look up table. The overall power consumption of fans will be

$$P_{fan} = N_f \cdot P_{fan,1} \tag{21}$$

where $N_f$ is number of fans.

## 4. CONTROLLER DESIGN

### 4.1 Dynamic programming

Dynamic programming (DP) is a well-known method for solving optimal control problems [9]. The solution found by DP is guaranteed to be globally optimal. DP is a numerical method which uses the decision making based on principle of optimality to sequence of decisions which together define an optimal policy and trajectory. The DPM tool [10] is used for simulation and the formulation of cost-to-go, etc. The state space model explained in (7)-(8) is discretized with time step of 1 second as

$$T_e(k+1) = \frac{1}{C_e}\cdot Q_{in}(k) - \frac{c_c}{C_e}\cdot \dot{m}_P(k)\cdot\left(T_e(k) - T_r(k)\right) + T_e(k) \tag{22}$$

$$T_r(k+1) = \frac{\dot{m}_P(k)}{C_r}\cdot\left(T_e(k) - T_r(k)\right) - \frac{\varepsilon \cdot c_a}{C_r}\cdot \dot{m}_f(k)\cdot\left(T_e(k) - T_\infty\right) + T_r(k) \tag{23}$$

The main objective of the controller is to regulate the engine temperature and in the meantime minimize the fuel consumption. Since the main intention of using DP is global optimization, computational load is not very important at this stage, so the grid for states and inputs is considered relatively small to guarantee optimality and decrease the error from interpolation.

### 4.2 State feedback controller

To evaluate the effect of Dynamic Programming, a simple state feedback controller is designed using pole-placement method. The controller tries to stabilize the engine and radiator temperatures to reference values. For designing the controller, the engine cooling system is linearized around the reference temperature, which is assumed constant in this case. The linearized system is

$$\begin{cases} \dot{x} = Ax + B_u u + B_v v + F \\ y = Cx + D_u u + D_v v + G \end{cases} \tag{24}$$

where

$$A = \left(\frac{\partial f}{\partial x}\right)_{(x_0,u_0,v_0)} \quad B_u = \left(\frac{\partial f}{\partial u}\right)_{(x_0,u_0,v_0)} \quad B_v = \left(\frac{\partial f}{\partial v}\right)_{(x_0,u_0,v_0)}$$

$$C = \left(\frac{\partial g}{\partial x}\right)_{(x_0,u_0,v_0)} \quad D_u = \left(\frac{\partial g}{\partial u}\right)_{(x_0,u_0,v_0)} \quad D_v = \left(\frac{\partial g}{\partial v}\right)_{(x_0,u_0,v_0)}$$

$$F = f(x_0, u_0, v_0) - Ax_0 - B_u u_0 - B_v v_0$$
$$G = g(x_0, u_0, v_0) - Cx_0 - D_u u_0 - D_v v_0$$

$$A = \frac{\partial f}{\partial x}\Big|_{(x_0,u_0,v_0)} = \begin{bmatrix} \frac{\partial f_1}{\partial x_1} & \frac{\partial f_1}{\partial x_2} \\ \frac{\partial f_2}{\partial x_1} & \frac{\partial f_2}{\partial x_2} \end{bmatrix}_{(x_0,u_0,v_0)} = \begin{bmatrix} -\frac{c_c}{C_e}\dot{m}_p & \frac{c_c}{C_e}\dot{m}_p \\ \frac{c_c}{C_r}\dot{m}_p - \frac{\varepsilon c_a}{C_r}\dot{m}_f & -\frac{c_c}{C_r}\dot{m}_p \end{bmatrix}_{(x_0,u_0,v_0)} \tag{25}$$

$$B_u = \frac{\partial f}{\partial u}\Big|_{(x_0,u_0,v_0)} = \begin{bmatrix} \frac{\partial f_1}{\partial u_1} & \frac{\partial f_1}{\partial u_2} \\ \frac{\partial f_2}{\partial u_1} & \frac{\partial f_2}{\partial u_2} \end{bmatrix}_{(x_0,u_0,v_0)} = \begin{bmatrix} \frac{1}{C_e}\cdot\frac{\partial F_2}{\partial \dot{m}_p} - \frac{c_c}{C_e}(T_e - T_r) & \frac{1}{C_e}\cdot\frac{\partial F_2}{\partial \dot{m}_f} \\ \frac{c_c}{C_r}(T_e - T_r) & -\frac{\varepsilon c_a}{C_r}(T_e - T_\infty) \end{bmatrix}_{(x_0,u_0,v_0)} \tag{26}$$

$$B_v = \frac{\partial f}{\partial v}\Big|_{(x_0,u_0,v_0)} = \begin{bmatrix} \frac{\partial f_1}{\partial v_1} & \frac{\partial f_1}{\partial v_2} & \frac{\partial f_1}{\partial v_3} \\ \frac{\partial f_2}{\partial v_1} & \frac{\partial f_2}{\partial v_2} & \frac{\partial f_2}{\partial v_3} \end{bmatrix}_{(x_0,u_0,v_0)} = \begin{bmatrix} \frac{\frac{\partial F_2}{\partial \omega_{eng}}}{C_e} & \frac{\frac{\partial F_2}{\partial T_{drv}}}{C_e} & \frac{\frac{\partial F_2}{\partial V_{veh}}}{C_e} \\ 0 & 0 & 0 \end{bmatrix}_{(x_0,u_0,v_0)} \tag{27}$$

$$C = \frac{\partial g}{\partial x}\Big|_{(x_0,u_0,v_0)} = \begin{bmatrix} \frac{\partial g_1}{\partial x_1} & \frac{\partial g_1}{\partial x_2} \\ \frac{\partial g_2}{\partial x_1} & \frac{\partial g_2}{\partial x_2} \\ \frac{\partial g_3}{\partial x_1} & \frac{\partial g_3}{\partial x_2} \end{bmatrix}_{(x_0,u_0,v_0)} = \begin{bmatrix} 1 & 0 \\ 0 & 1 \\ 0 & 0 \end{bmatrix}_{(x_0,u_0,v_0)} \tag{28}$$

$$D_u = \frac{\partial g}{\partial u}\Big|_{(x_0,u_0,v_0)} = \begin{bmatrix} \frac{\partial g_1}{\partial u_1} & \frac{\partial g_1}{\partial u_2} \\ \frac{\partial g_2}{\partial u_1} & \frac{\partial g_2}{\partial u_2} \\ \frac{\partial g_3}{\partial u_1} & \frac{\partial g_3}{\partial u_2} \end{bmatrix}_{(x_0,u_0,v_0)} = \begin{bmatrix} 0 & 0 \\ 0 & 0 \\ \frac{\frac{\partial F_1}{\partial \dot{m}_p}}{\dot{m}_p} & \frac{\frac{\partial F_1}{\partial \dot{m}_f}}{\dot{m}_f} \end{bmatrix}_{(x_0,u_0,v_0)} \tag{29}$$

Then the poles of the system are identified and the controller tries to put the poles in the required positions. The system has two poles and one of them is very close to the unstable region and should be moved in order to guarantee stability of the system. The proper value for the new location of the poles is decided based on trial an error and studying the behavior of system. The gains for the controller are calculated using Matlab control toolbox. This controller is called simple controller in the rest of the paper.

## 5. SIMULATION RESULTS

### 5.1 Driving cycles

In order to evaluate the results, proper driving cycles should be chosen. A simple driving cycle and a more complicated one are chosen in this work. The former is an imaginary cycle with up and downhill developed purely for this study which is called simple cycle in the rest of the report. The latter is based on a real cycle which is a distance traveled between two cities in Sweden. It is called real cycle in the rest of the report. Since the model for vehicle in this study is simple and does not include model for gear changing, etc., the truck has been modeled and simulated in Autonomie [11] using the two cycles, and different outputs from Autonomie have been used as the inputs for simulations in this study. Autonomie is a Matlab/Simulink based software which is used to analyze different powertrain systems primarily for fuel efficiency and emission comparisons.

### 5.2 Results

Results from the simulation using dynamic programming are presented here. Simulations are done for both simple and real driving cycles. The simple driving cycle is used to explain the way energy buffer control reduces fuel consumption.

The plots for driving cycle and engine temperature are presented in Figure 4 and coolant pump actuation is presented in Figure 5. As can be seen in the Figure 4, a drop in the temperature of the engine can be noticed before the two situations of high power demand from engine; at 200 seconds which is uphill and around 800 seconds which is high acceleration demand. This can be due to the fact that engine and coolant flow can behave like an energy buffer, which can store heat energy. This helps engine to avoid operating in low efficiency regions by pre-cooling the engine in the situations in which power demand from the engine is not high, and thus less power demand for pump and fan in the high power demand situations. This can contribute to some fuel saving in the vehicle. The actuation of coolant power is presented in Figure 5. Moreover, Optimal control of the thermal system decreases the overall usage of actuators, which also contribute to fuel consumption reduction. Another aspect of optimal TMS is that by using optimal control methods, it can be guaranteed that engine can work in higher temperature, without going above the limit. This can also contribute in some fuel saving, which is not considered in this work. During the simulations, an improvement in the fuel consumption can be noticed when comparing the simple controller and the global optimal case. Although the exact values of improvements cannot be determined due to lack of accurate data (accurate engine map, accurate parameters in the thermal system model, etc.), the potential of improvement can be confirmed. In the case of the simple cycle, an improvement of 1.6% is seen.

Similar simulations have been done on the real cycle.

In this cycle, fuel consumption could be improved by 1.2% compared to the simple controller. The temperature trajectory in the real cycle is presented in Figure 6.

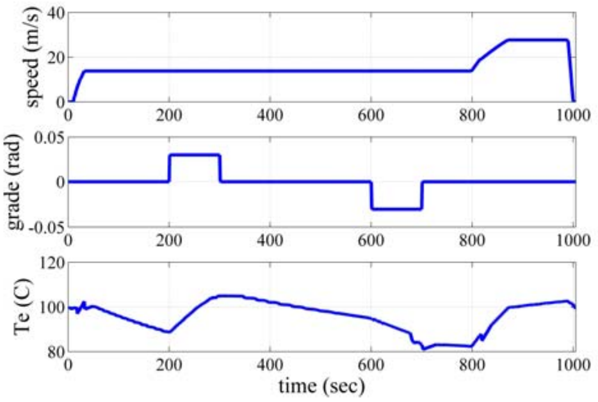


Figure 4. Engine temperature trajectory in the simple driving cycle

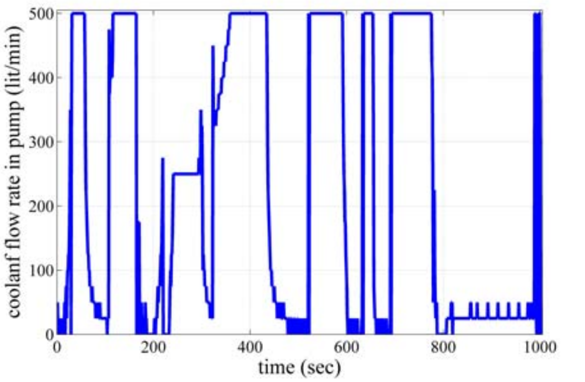


Figure 5. Coolant pump actuation in the simple driving cycle

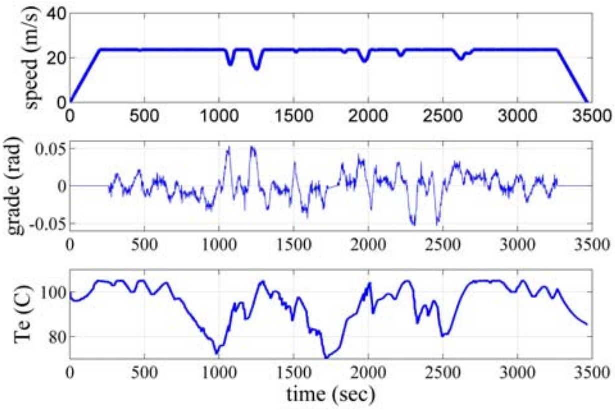


Figure 6. Engine temperature trajectory in the real driving cycle

## 6. CONCLUSION

Fuel saving potential of optimal management of TMS is investigated in this paper. The idea of energy buffer control in conventional vehicles is described to show the rationale behind this improvement.

Results have been compared with a simple state feedback controller to show the performance of optimal control of TMS. The results show improvement in fuel efficiency when a simple controller is compared with the global optimal case. This means that using more advanced controller for controlling engine TMS can improve fuel efficiency. Although the fuel efficiency improvement seems limited, it can be higher in real vehicle. In the application of the simple controller, several operational limits exist, which do not allow the temperature to rise as close to the maximum allowable temperature as it is assumed in this paper; in contrast, the more advance control methods such as model predictive control, stochastic dynamic programming, etc. can handle these limits thus resulting more fuel efficiency improvement.

In this paper, only the engine cooling system has been considered. Consideration of other energy buffers e.g. battery is the subject of ongoing project. Although the potential of fuel saving using optimal control is shown in this paper, no optimal real time controller is presented. Several optimal control strategy based on prediction are currently being investigated.

## 7. ACKNOWLEDGEMENT

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