Development and Implementation of a Controllable Thermostat for an Engine Cooling System

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Abstract

Reducing fuel consumption is one of the main aims when developing new trucks at Scania. For instance, there are a lot of new EU regulations on what emission levels are allowed. Also, having a low fuel consumption gives Scania an edge over their competitors. This master thesis will investigate one possible way of reducing the fuel consumption in a truck. The aim is to allow a higher engine temperature which gives a higher oil temperature causing less friction in the engine. Thus leading to a more efficient engine and reduced fuel consumption. This needs to be done without risking that the engine overheats. The method used to achieve this is a controllable thermostat which regulates the engine temperature with respect to different driving scenarios. In this work a control strategy for a controllable thermostat is produced with help of a Simulik environment where it’s tested. A controllable thermostat consisting of a three way valve and an actuator is then installed in a truck and tested on roads. The tests in the truck shows that the controllable thermostat can efficiently regulate the engine temperature to different reference temperatures and results from simulations showed that a fuel reduction is possible.
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<table>
<thead>
<tr>
<th>Variable</th>
<th>Unit</th>
<th>Description</th>
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<tbody>
<tr>
<td>$\dot{m}_{coolant}$</td>
<td>[kg/s]</td>
<td>Mass flow coolant</td>
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<tr>
<td>$\dot{m}_{c,coolant}$</td>
<td>[kg/s]</td>
<td>Mass flow coolant through radiator</td>
</tr>
<tr>
<td>$\dot{m}_{air}$</td>
<td>[kg/s]</td>
<td>Mass flow air through the radiator</td>
</tr>
<tr>
<td>$m_{truck}$</td>
<td>[kg]</td>
<td>Mass of the truck</td>
</tr>
<tr>
<td>$c_{p,c}$</td>
<td>[J/kgK]</td>
<td>Coolant specific heat capacity</td>
</tr>
<tr>
<td>$c_{p,air}$</td>
<td>[J/kgK]</td>
<td>Air specific heat capacity</td>
</tr>
<tr>
<td>$T_{eng}$</td>
<td>[°C]</td>
<td>Temperature of the coolant after the engine</td>
</tr>
<tr>
<td>$T_{ref}$</td>
<td>[°C]</td>
<td>Reference temperature for the coolant after the engine</td>
</tr>
<tr>
<td>$Q_{Rad}$</td>
<td>[J/s]</td>
<td>Heat flow from zone 2 to the ambient air via the radiator</td>
</tr>
<tr>
<td>$(UA)_{Rad}$</td>
<td>[J/sK]</td>
<td>Total heat transfer coefficient radiator to ambient air</td>
</tr>
<tr>
<td>$n_{eng}$</td>
<td>[rpm]</td>
<td>Engine speed</td>
</tr>
<tr>
<td>$C_N$</td>
<td>[-]</td>
<td>Calibration parameter, $\dot{m}_{coolant}$</td>
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<tr>
<td>$T_{frac}$</td>
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<td>Thermostat opening</td>
</tr>
<tr>
<td>$u_{PI}$</td>
<td>[-]</td>
<td>Control signal from the PI-controller</td>
</tr>
<tr>
<td>$u_{PWM}$</td>
<td>[V]</td>
<td>PWM voltage from the engine control system</td>
</tr>
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<td>$u_{act}$</td>
<td>[V]</td>
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Chapter 1

Introduction

In the vehicle industry there is an ongoing effort to reduce the fuel consumption in the vehicles. First of all fuel is the greatest cost for an operating vehicle. For Scania which makes premium trucks, good fuel efficiency is the number one selling argument for the customer. There are also requirements from EU on the emission levels that are allowed. A combustion engine consists of different subsystems that work together to achieve efficient combustion. Improving the efficiency or reducing the losses in one of these subsystems can reduce the fuel consumption. One such subsystem is the cooling system.

The aim of the cooling system is to keep the engine at a good operating temperature without overheating. This is regulated with a coolant which is sent through the engine. If the engine needs cooling, the coolant is then sent through the radiator, otherwise it’s sent through a bypass. The thermostat determines the flow ratio between the radiator and the bypass. The current thermostat is a mechanical solution with wax pellets that melt and expand when the temperature reaches a certain level. The expanding pellets pushes up a valve which leads the coolant to the radiator.

The thermostat is dimensioned for a worst case scenario where the cooling demand is at its greatest. This isn’t optimal in regular operation. If the thermostat instead is regulated using logic that takes the current operating conditions into account a more optimal engine temperature can be achieved that may give a better combustion. If the engine temperature is allowed to increase, the oil temperature would also increase which leads to less friction in the engine and thus a better combustion. However the temperature can’t be allowed to increase if it involves a risk that the engine will overheat. This master thesis will investigate the benefits of using a controllable thermostat in the cooling system for a combustion engine in a truck.

This report contains some trade secrets which have to be censored in the published report like this: ****.

1.1 Problem description

The goal of this thesis was to investigate what fuel savings can be achieved if a controllable thermostat was implemented. The thermostat would allow for regulation of the engine temperature with respect to different driving scenarios. In this work there are two factors that determines the driving scenario, ambient temperature and vehicle mass. These two factors don’t usually change much during operation and were therefore used to determine what temperature to regulate against. The new thermostat should be able to regulate against any given temperature. This master thesis focus on the design of the controller and the implementation of the new
thermostat in the truck.

The master thesis was conducted in close collaboration with another master’s thesis. The two thesis had the same end goal and a shared background. The other theses was conducted by Carl Wahlström, student at Uppsala universitet. The focus of the other thesis was to evaluate what temperature to regulate against.

The resulting fuel reduction shall be calculated from data gathered during test runs on various routes with different topography. This is done in order to cover normal operating conditions for a haulage truck.

1.2 Previous work

This wasn’t the first time a controllable thermostat for a truck has been investigated. For instance, Scania had a master thesis back in 1992 that looked at the benefits from a controllable thermostat [1]. This thesis had a different focus, for instance one of their aims was to reduce the pressure losses in the cooling system. Other studies have shown that a controllable pump together with a controllable thermostat can give a more efficient temperature control [2], this was however not investigated in this thesis. This master’s thesis differs from other studies because the focus was to regulate the temperature with respect to different driving scenarios.

1.3 Common goals

The common end goal for the theses are a reduced fuel consumption through optimal temperature regulation. The common goals were:

- Develop a model of the system with the new controllable thermostat implemented.
- Implement the new controllable thermostat in a real truck.
- Quantify the fuel reduction as a function of higher engine temperature.
- Quantify the fuel reduction through simulation on different routes.
- Quantify fuel reduction in a real truck.

1.4 Limitation

The following limitations have been made when designing the control.

- No attempts have been made to minimize pressure losses in the cooling system.
- No controllable coolant pump is used.

1.5 Temperature reference

The other collaborating master’s thesis will investigate what temperature to regulate against. By looking at different parameters the optimal temperature to regulate against without risking that the engine overheats could be determined. The main parameters were outside temperature and vehicle mass. But other parameters were also considered to further increase the performance. For instance cruise control with active prediction could be used to identify future cooling needs or opportunities for more efficient cooling.
1.6 Control design

The controller should be able to efficiently follow the temperature reference. The controller must have satisfactory performance in terms of speed and oscillations. The main thing to control is the valve through an actuator which replaces the old thermostat. The following shall be conducted in this thesis:

- Model the flow system with the new controllable thermostat.
- Design a control strategy.
- Install the controllable thermostat in a truck.
- Control the thermostat using the truck’s engine control system.

- Verify the models in the test rig and truck.
- Verify that controller follows the requested temperature.
- Give the measure of control performance; speed, oscillations, overshoot.
- Investigate how fast the actuator needs to be in order to handle transients.

- The thermostat shall be fully open when the fan is used.
- Use the controllable thermostat so that the need for fan use is minimized.

- The control shall be faster than the old thermostat.
- The amplitude of the oscillations shall be less than the old thermostat.
Chapter 2

System Description

2.1 Cooling system

The cooling system with its different parts can be seen in Figure 2.1. The engine, the retarder and the EGR-cooler are the most significant heat sources in the system. The retarder is an hydraulic break which is used in addition to the regular break system. The retarder generates a lot of heat when used. The retarder is placed after the engine. The EGR is a system where the exhaust is fed back in to the combustion in order to reduce nitrogen oxide emissions. The exhaust needs too be cooled before it is fed back and this is done in the EGR-cooler. The EGR-cooler is placed after the engine and only a portion of the coolant is lead through the EGR-cooler. The portion of the coolant that goes through the EGR-cooler is connected to the coolant after the radiator.

Figure 2.1: Cooling system
2.1.1 Pump

The pump pumps the coolant through the cooling system. The pump is driven by the engine so the mass flow coolant is proportional to the engine speed, according to (2.1). The constant $C_N$ is known for the cooling system with the old thermostat. How the new thermostat affects the mass flow needed to be investigated. This was done in a flux rig at Scania with the new thermostat installed. In the flux rig coolant was pumped through the cooling system with a sensor attached to it that measured flux and pressure. The pump in the rig was calibrated to match the pump in the truck so one could simulate different engine speeds and get the same performance. The pressure was measured to make sure that the new thermostat didn’t give rise to any harmful pressures. Other than that, no attempts was made to optimise the pressure in the system. The flux sensor was mounted such that the total flux and the flux in the radiator could be measured. With these sensors the total flux could be measured as a function of the engine speed, see Figure 2.2. From the figure we get that $C_N = \frac{\dot{m}_{\text{coolant}}}{n_{\text{eng}}}$ with the new thermostat.

$$\dot{m}_{\text{coolant}} = C_N n_{\text{eng}}. \quad (2.1)$$

![Figure 2.2: Mass flow of coolant as a function of the engine speed](image)
2.1.2 Thermostat

The thermostat leads the coolant either through the radiator, where the coolant is cooled, or the bypass. The old thermostat is a mechanical solution with wax pellets that melts at a given temperature and presses up two discs that redirects the coolant to the radiator when cooling is needed, see Figure 2.3. The new thermostat implemented in this project was a three way valve that was controlled via an actuator. An actuator is a device that takes in control signal and gives a corresponding output, in this case an control voltage was feed in and a motor gave the valve a corresponding angle. The valve used as a prototype during this project was a shunt valve designed for household heating systems with provided actuator [5]. The actuator was controlled with control voltage, $u_{act}$, in the range 2 - 6 V which corresponds to 0 - 100 % open thermostat. It takes 15 s for the actuator to shift from fully closed to fully open and vice versa. This can be compared with the old thermostat that can take about one minute to do the same, depending on the temperature step. The degree of opening of the thermostat, $T_{frac}$, together with the total mass flow of the coolant gave the mass flow coolant to the radiator as shown in (2.2).

![Figure 2.3: The (old) mechanical thermostat, [6].](image)

\[
\dot{m}_{c,\text{coolant}} = \dot{m}_{\text{coolant}} T_{frac}, \quad (2.2)
\]

The dynamics of the new thermostat was also investigated in the flux rig. Different control voltages was sent to the actuator and then it was recorded what percentage of the total mass flow coolant that was going through the radiator. The result can be seen in Figure 2.4. It can be noted that the control voltage where in the range 2.2 - 6 V which differs from the data sheet. This could be due to a manufacture error or damage caused to the actuator. But the actuator still showed linear properties in that range so this problem was easily avoided. The following linear model was used when simulating the new thermostat (2.3). The new controllable thermostat demands that the temperature was measured and could be used as a control signal. A sensor is placed after the engine, which is standard for all Scania trucks. The temperature of the coolant after the engine is a good indicator of the temperature inside the engine and is therefore denoted engine temperature, $T_{\text{eng}}$.

\[
\dot{m}_{c,\text{coolant}} = C_C (u_{act} - 2.2) \dot{m}_{\text{coolant}}, \quad C_C = \frac{1}{3.8} \quad (2.3)
\]
2.1.3 Radiator

The radiator is the main heat sink in the cooling system and it has some features that needs to be understood. These features are used when designing the controller for the thermostat. The cooling medium for the radiator is the ambient air. The air is taken in at the front of the truck and then led through the radiator, cooling the coolant inside the radiator in process. The coolant is led through the radiator by a series of pipes. A cross-section of the radiator can be seen in Figure 2.5. In the figure, it can be noticed that the part of the radiator where air is led through is much larger than the coolant part, roughly four times larger. This is due to the fact that the heat capacity for the coolant is approximately four times bigger than the heat capacity of the air.

Figure 2.4: Percentage of the mass flow through the radiator as a function of control voltage
Figure 2.5: Cross section of the radiator. The air flows through grills seen to the left. To the right five narrow pipes for the coolant can be seen, [6].

The heat flow can be described by the thermodynamic formula (2.4), [4, p. 201]. Since the cooling medium for the radiator is the ambient air, the ideal case would be an equal temperature of the coolant after the radiator as to the ambient air temperature. Then the heat flow would be maximized for a given mass flow of coolant to the radiator.

\[ \dot{Q}_{\text{Rad}} = \dot{m}_{\text{coolant}} c_{p,\text{coolant}} (T_{\text{eng}}^{(\text{before})} - T_{\text{eng}}^{(\text{after})}) \]  

(2.4)

But the heat flow from the radiator is also limited by the cooling medium. Since the air is used to transfer the heat from the radiator, the cooling capacity is given by

\[ \dot{Q}_{\text{rad,cap}} = \dot{m}_{\text{air}} c_{p,\text{air}} (T_{\text{air}}^{(\text{before})} - T_{\text{air}}^{(\text{after})}). \]  

(2.5)

The ideal case here would be for \( T_{\text{air}}^{(\text{after})} \) to be equal to \( T_{\text{eng}}^{(\text{before})} \). In the ideal case the radiator could be described by the formula

\[ \dot{Q}_{\text{rad}} = (UA)_{\text{rad,ideal}} (T_{\text{eng}} - T_{\text{air}}). \]  

(2.6)

Where \((UA)_{\text{rad,ideal}}\) is the ideal total heat transfer constant. \((UA)_{\text{rad,ideal}}\) is a function of \( \dot{m}_{\text{coolant}} \) times the coolant’s specific heat coefficient \( c_{p,\text{coolant}} \) and it’s limited by \( \dot{m}_{\text{air}} \) times \( c_{p,\text{air}} \). The specific heat coefficients are given in Table 2.1. Figure 2.6 shows \((UA)_{\text{rad,ideal}}\) for different mass flows of coolant and air.
The real radiator doesn’t possess these ideal properties. Especially at large mass flows the radiator doesn’t have time to affect the temperature of the coolant and air enough. But it’s still possible to find a relation between the mass flows and the total heat transfer function. Figure 2.7 shows the real \((UA)_{rad}\) relation, which is produced using data from the manufacturer. As can be seen, the ideal properties are somewhat true when the mass flow coolant is small and the mass flow air is large and vice versa. The relations between \(\dot{m}_{air}\) and \(\dot{m}_{c,coolant}\) of the real \((UA)_{rad}\) are used when designing the controller.
2.1.4 Fan

The mass flow air limits how much heat can be transferred from the engine. Neglecting wind conditions gives that the vehicle speed determines the mass flow air during ordinary conditions. However if the heat transfer needs to increase in order to maintain acceptable temperatures in the engine, the fan is used. The fan sets in when the temperature approaches critical temperatures. Since the goal was to increase the temperature in certain driving scenarios a new fan control was implemented that was dependent on the driving scenario. For more information on how the new fan control was designed, please see Carl Wahlström’s report [3]. In general the fan was started at $10^\circ$C over the reference temperature. The fan wasn’t considered when designing the controller for the thermostat other than to make certain the thermostat was fully open when the fan was in use.
Chapter 3

Control Design

When the cooling system was understood the control could be designed and then implemented. The cooling system with the controller in place can be seen in Figure 3.1.

![Figure 3.1: Cooling system with controller](image)

The investigation of the reference temperature and the design of the controller was first done in a simulations environment and then implemented in a truck. In Figure 3.2 box diagram of how the control system was implemented is shown with the input and output signals. The different blocks will be discussed further below.
3.1 Driving scenarios

The key feature investigated with the controllable thermostat was the ability to control the thermostat with respect to different driving scenarios. The controller needed to be able to track different reference temperatures and reject disturbances. But then it needed to be investigated what temperature it’s wise to regulate against at different driving scenarios. This was done in the other master’s thesis in close collaboration with this one. The goal is to keep as high temperature as possible, but with a sufficient thermal buffer so that the engine doesn’t risk running dangerously hot when driving in ambient conditions like steep inclines which wasn’t included in the driving scenario. In order to investigate what reference temperature was suitable at different driving scenarios, simulations where the truck was run on a section of road containing several steep inclines. Conditions were set on what top temperature and what temperature average was acceptable. Then a substantial number of simulations where run on the selected road with sufficiently many driving scenarios and reference temperatures. After the simulations the data were studied in order to see what reference temperature met the requirements at the different driving scenarios. For more information how the reference temperature was set, please see Carl Wahlström’s report [3].

3.2 Controller

3.2.1 Designing the controller

When the truck was started, the mass of the truck and the ambient temperature was immediately measured and used to calculate a reference temperature for the controller. This reference value would then stay pretty much constant during the whole run. It might need to be recalculated a few times due to the temperature shifts over the course of the day, but for the majority of the run it will stay constant. This made it so that from a controller perspective it’s basically a servo problem with reference tracking and disturbance attenuation.

The main thing considered when designing the controller was the radiator. As described in Chapter 2, the cooling capacity is limited by the air mass flow, $\dot{m}_{\text{air}}$. If the wind conditions are neglected, $\dot{m}_{\text{air}}$ is a function of the vehicle velocity and the fan speed. The first demand put on the controller was that the thermostat should be fully open when the fan is used. The fan takes
a lot of energy when in use so the fan use should be minimized. So when the fan is needed the
cooling effect should be maximized during use. This gives that $\dot{m}_{\text{air}}$ flow can be seen as only a
function of the vehicle velocity in the control area. A truck isn’t allowed to drive faster than 90
$km/h$, which gives the maximum $\dot{m}_{\text{air}}$ around.
The coolant mass flow to the radiator, $\dot{m}_{\text{coolant}}$, is dependent on the total mass flow coolant,
$\dot{m}_{\text{coolant}}$, and the degree of opening of the thermostat. The degree of opening of the thermostat
is determined by the controller and the total mass flow coolant is as mentioned dependent on the
engine speed. In normal driving conditions the engine speed is in a interval of 800 - 1800 $rpm$
and the majority of the time the automatic gearbox keeps a gear such that the engine speed is
around 1200 $rpm$. This gives a total mass flow coolant of around.

Taking these two facts into consideration one can realize that the cooling capacity for the radi-
ator, in normal driving conditions without fan, is easily saturated. For example, if the truck is
driven at 80 $km/h$ with an engine speed of 1200 $rpm$, $(U\alpha)_{\text{rad}}$ is $J/sK$ with the thermostat
30 % and at 100 % open $(U\alpha)_{\text{rad}}$ increases with only 7.7 %. This motivates that the controller
should work with small opening angles as much as possible. Large opening angles will only give
a small increase in the cooling capacity and it will take longer time for the thermostat to close,
should the temperature drop.

The control implemented was a PI-controller (3.1). The error signal feed into the controller was
temperature difference $e(t) = T_{\text{ref}}(t) - T_{\text{eng}}(t)$. The output was a control signal, $u_{\text{PI}}$, in the
range 0 - 1 which corresponds to 0 - 100 % open thermostat. The integral part was used to
eliminate static errors but since the cooling capacity of the radiator gets saturated very fast, the
integrated part is very limited. The integrated error isn’t allowed to grow over 30. The P part
was also small for this reason.

$$U_{\text{pi}} = P e(t) + I \int_0^t e(\tau)d\tau$$

(3.1)

The performance of the controller was evaluated in the Simulink model. When trying out
PI-parameters a normal driving condition was simulated; $v_{\text{veh}} = 80 \text{ km/h}$ and $n_{\text{eng}} = 1200 \text{ rpm}$.
A step in the reference temperature was evaluated with different PI-parameters at different
vehicle loads and ambient temperatures. The parameters that showed the best performance
with respect to speed and overshoot was $P = 0.09$ and $I = 0.0033$. These gains together with
the caped integrated error and the fact that the fan starts at $10^\circ$C over the reference temperature
makes it so that the thermostat is fully open when the fan sets in. A plot of the resulting step
response can be seen in Figure 3.3. The degree of opening of the new thermostat looks quantized,
cause the thermostat is modelled with a fixed opening speed and a quite large sample time of 0.1
$s$. The real controllable thermostat doesn’t have these properties. For more information about
the Simulink model and how it’s used see Section 5.1 in the Appendix.
To compare the new thermostats performance with the old one a new simulation was done. The same driving condition was simulated. But instead of changing the reference temperature, since the old thermostat doesn’t have a reference temperature, the retarder was connected as an arbitrary heat source at $t = 100$ s and disconnected at $t = 300$ s. The result can be seen in Figure 3.4. The old thermostat can’t get the temperature down to it’s opening temperature as the new one manages and the new thermostat keeps a lower temperature during the step. The I-part gives the new thermostat a dip when the retarder wasn’t used any more, but it stabilises much faster. The Simulink model of the old thermostat didn’t model the delay of the thermostat which gave the old thermostat a much smoother result than expected.
3.2.2 Gain scheduling

In order for the controller to deal with some known issues, gain scheduling was implemented on the P-part. One of the problems faced with the old thermostat was time delay on the feedback at extreme temperatures. Since low engine speeds give low mass flow coolant it takes longer for the coolant to pass through the whole system. If the temperature of the ambient air is very high then the cooling capacity is poor. If this is combined with slow feedback the result can be that the temperature in the engine gets dangerously high before the controller can react properly. These problems was dealt with using gain scheduling where different gains where added to the P-part dependent on the driving scenario.

3.3 Engine control system and filter circuit

The logic and the controller were programmed into the trucks engine control system. A PWM-signal, $u_{PWM}$, was used as output from the control unit. This meant that the control signal from the PI-controller, $u_{PI}$ needed to be converted into a duty cycle for the PWM-signal that gives the corresponding control voltage, $u_{act}$. In order to get a more analogue voltage level for the actuator, a low pass filter circuit was designed which the PWM-signal was sent through.
The filter circuit can be seen in Figure 3.5. The filter circuit consists of first a voltage divider which reduced the voltage down from 28 to 6 V. Then an OP-amp was connected as a voltage follower followed by an RC-filter. The voltage after the filter was sent to the actuator. After the RC-filter there was another voltage divider and then this voltage was sent back to the control unit. The voltage was also sent back to the control unit through a voltage divider. This voltage was compared to a modelled voltage, $V_{Mod}$, in order to control that the voltage sent to the actuator corresponds to the control signal, $U_{pi}$. If they didn’t corresponds due to some error in the filter circuit or engine control unit, the difference in voltage was converted to a error added to the control signal. The purpose second voltage divider was to make sure that the voltage sent back didn’t exceed 5 V, which was demanded by the engine control unit.

![Figure 3.5: The filter circuit](image-url)
Chapter 4

Results

The result section consist of two parts. Simulated results from Simulink and results from test runs in the real truck. In the truck the logic and controller was tested to see if they worked as expected.

4.1 Quantify fuel reduction

When quantifying the fuel reduction in Simulink two runs was made with the same parameters. One simulation with the old thermostat and one with the new. The two Simulations where then compared and a estimate of the difference in fuel consumption was calculated. When calculating the fuel reduction two variables were considered, engine temperature and fan use, all other variables were equal for both simulations. Higher engine temperature leads to higher oil temperature which gives less friction in the engine. On the other hand, if the temperature rose so high that the fan needed to be used the energy loss increased a lot during fan use. In the simulations the two runs were compared side by side and more fan use where recalculated to higher energy losses and higher engine temperature was recalculated to less energy losses in the system. So ideally the new logic and controller should keep a higher engine temperature without increasing the fan use. Simulations showed that a power reduction of up too kW was possible with the new thermostat. This means a fuel reduction of over , which was a good result. For more information of how the fuel reduction was quantified and how well it worked with different driving scenarios, please see Carl Wahlström’s report [3].

4.2 Truck results

The tests was conducted on Scania’s test track with a load frame on the truck which gave the truck a total mass of about 14 ton. The temperature was about 10 °C, so the scenario was tested in easy conditions. The test track is quite short, contains many slopes and sharp turns that require breaking. The result from the test track was therefore more oscillating then results from road testing. The result from two similar test runs with the old and the new thermostat can be seen in Figure 4.1. As can be seen the resulting temperature oscillate quite a bit as expected. But the
controller managed to keep the temperature in a span of $\pm 2 \, ^\circ C$ for the new thermostat compared to the old thermostat which oscillates at about $\pm 4 \, ^\circ C$. The controller did this with quite small control outputs and the thermostat never opens beyond 30 \%.

To illustrate that the cooling capacity doesn’t get greater beyond a certain degree of opening of the thermostat another test run was made. In this test the retarder was used through out the run which resulted in a excessive heat gain. During a sequence from this test run the cooling system wasn’t able to regulate the temperature down to the requested level, seen in Figure 4.2. In the sequence the thermostat goes from having an opening percentage of about 15 \% up to almost 50 \% shown in the first plot. In the second plot the total heat capacity, $(UA)_{rad}$, is shown. Note that it’s almost constant during the whole sequence, even though the thermostat opens quite a lot in the end. The last plot shows the two mass flows to the radiator, $\dot{m}_{c,coolant}$ and $\dot{m}_{air}$. Note that $(UA)_{rad}$ follows $\dot{m}_{air}$ more than $\dot{m}_{c,coolant}$ which indicates that we are in the saturated area of Figure 2.7.

Figure 4.1: Result comparison from truck, new thermostat and old thermostat
Figure 4.2: Total heat capacity from test run
Chapter 5

Conclusions

The objective of this master’s thesis was to investigate what fuel savings could be made if the engine temperature was regulated according to different driving scenarios. The goal was to allow a higher temperature in the engine if the outside conditions allowed it, thus getting a higher oil temperature which leads to less friction. The method for achieving this was installing a controllable thermostat with logic that gave a reference temperature for the current driving scenario and a regulator that followed this reference temperature. Simulations of the logic and controller showed that a fuel reduction of more than 10% was possible and the controllable thermostat was successfully installed in a truck.

No quantification was made in the real truck. This was due to time limitations and the fact that no similar truck for performing reference runs was available. To fully evaluate the fuel reduction and the control performance two similar runs needed to be made during similar driving conditions. The controller did however show good result as it gave less oscillations than the old thermostat and was able to follow the reference temperature that was set.

5.1 Future Work

The controller needs to be tested further, preferably along side a reference truck with the old thermostat. And more driving scenarios need to be evaluated, at least the extreme once when it’s very cold or very hot. The test runs was performed during two weeks in the middle of June and the temperature didn’t change significantly during that time. Some interesting ideas for further work that has been discussed during the project but hasn’t been investigated are:

- One possibility of improvement would be to install more temperature sensors in the system for more efficient control. For instance the temperature of the coolant in to the engine could be measured in order to get a faster feedback.

- More parameters could be followed in the controller for more efficient control. For instance when the retarder is in use it produces a lot of heat, so the control could be made predictive by looking at retarder usage and act faster once it’s used.
Bibliography


## Appendix

### List of Symbols

<table>
<thead>
<tr>
<th>Variable</th>
<th>Unit</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$m_1$</td>
<td>[kg]</td>
<td>Section mass of the engine block zone one</td>
</tr>
<tr>
<td>$m_2$</td>
<td>[kg]</td>
<td>Mass of the coolant</td>
</tr>
<tr>
<td>$m_3$</td>
<td>[kg]</td>
<td>Section mass of the engine block zone three</td>
</tr>
<tr>
<td>$\dot{m}_{avg}$</td>
<td>[kg/s]</td>
<td>Mass flow exhaust gas</td>
</tr>
<tr>
<td>$\dot{m}_{coolant}$</td>
<td>[kg/s]</td>
<td>Mass flow coolant</td>
</tr>
<tr>
<td>$\dot{m}_{c,coolant}$</td>
<td>[kg/s]</td>
<td>Mass flow coolant through radiator</td>
</tr>
<tr>
<td>$\dot{m}_{air}$</td>
<td>[kg/s]</td>
<td>Mass flow air through the radiator</td>
</tr>
<tr>
<td>$\dot{m}_{EGR}$</td>
<td>[kg/s]</td>
<td>Mass flow EGR-gas</td>
</tr>
<tr>
<td>$c_{p,eng}$</td>
<td>[J/kgK]</td>
<td>Engine block specific heat capacity</td>
</tr>
<tr>
<td>$c_{p,c}$</td>
<td>[J/kgK]</td>
<td>Coolant specific heat capacity</td>
</tr>
<tr>
<td>$c_{p,gas}$</td>
<td>[J/kgK]</td>
<td>Exhaust gas specific heat capacity</td>
</tr>
<tr>
<td>$T_1$</td>
<td>[°C]</td>
<td>Temperature engine block zone one</td>
</tr>
<tr>
<td>$T_{eng}$</td>
<td>[°C]</td>
<td>Temperature of the coolant after the engine</td>
</tr>
<tr>
<td>$T_3$</td>
<td>[°C]</td>
<td>Temperature engine block zone three</td>
</tr>
<tr>
<td>$T_{Air}$</td>
<td>[°C]</td>
<td>Ambient temperature</td>
</tr>
<tr>
<td>$T_{avg}$</td>
<td>[°C]</td>
<td>Exhaust gas temperature</td>
</tr>
<tr>
<td>$T_{abs}$</td>
<td>[°C]</td>
<td>Absolute temperature, 0 °C</td>
</tr>
<tr>
<td>$\dot{Q}_{in}$</td>
<td>[J/s]</td>
<td>Heat flow from combustion to zone 1</td>
</tr>
<tr>
<td>$\dot{Q}_{1-2}$</td>
<td>[J/s]</td>
<td>Heat flow from zone 1 to zone 2</td>
</tr>
<tr>
<td>Variable</td>
<td>Unit</td>
<td>Description</td>
</tr>
<tr>
<td>------------</td>
<td>------------</td>
<td>-----------------------------------------------------------------------------</td>
</tr>
<tr>
<td>$\dot{Q}_{2-3}$</td>
<td>[J/s]</td>
<td>Heat flow from zone 2 to zone 3</td>
</tr>
<tr>
<td>$\dot{Q}_{EGR}$</td>
<td>[J/s]</td>
<td>Heat flow from the EGR-gas via the EGR-cooler to zone 2</td>
</tr>
<tr>
<td>$\dot{Q}_{Rad}$</td>
<td>[J/s]</td>
<td>Heat flow from zone 2 to the ambient air via the radiator</td>
</tr>
<tr>
<td>$\dot{Q}_{Air}$</td>
<td>[J/s]</td>
<td>Heat flow from zone 3 to the ambient air</td>
</tr>
<tr>
<td>$(UA)_{1-2}$</td>
<td>[J/sK]</td>
<td>Total heat transfer coefficient zone 1 to zone 2</td>
</tr>
<tr>
<td>$(UA)_{2-3}$</td>
<td>[J/sK]</td>
<td>Total heat transfer coefficient zone 2 to zone 3</td>
</tr>
<tr>
<td>$(UA)_{Rad}$</td>
<td>[J/sK]</td>
<td>Total heat transfer coefficient radiator to ambient air</td>
</tr>
<tr>
<td>$(UA)_{Air}$</td>
<td>[J/sK]</td>
<td>Total heat transfer coefficient zone 3 to ambient air</td>
</tr>
<tr>
<td>$n_{eng}$</td>
<td>[rpm]</td>
<td>Engine speed</td>
</tr>
<tr>
<td>$n_{fan}$</td>
<td>[rpm]</td>
<td>Fan speed</td>
</tr>
<tr>
<td>$C_1$</td>
<td>[-]</td>
<td>Calibration parameter $(UA)_{2-3}$</td>
</tr>
<tr>
<td>$C_2$</td>
<td>[-]</td>
<td>Calibration parameter $(UA)_{Air}$</td>
</tr>
<tr>
<td>$C_N$</td>
<td>[-]</td>
<td>Calibration parameter $\dot{m}_{coolant}$</td>
</tr>
<tr>
<td>$\dot{V}_{air}$</td>
<td>[m$^3$/s]</td>
<td>Volume flow air</td>
</tr>
<tr>
<td>$\rho_{air}$</td>
<td>[kg/m$^2$]</td>
<td>Density air</td>
</tr>
<tr>
<td>$v_{veh}$</td>
<td>[m/s]</td>
<td>Vehicle velocity</td>
</tr>
<tr>
<td>$p_{amb}$</td>
<td>[Pa]</td>
<td>Ambient pressure</td>
</tr>
<tr>
<td>$R$</td>
<td>[J/kgK]</td>
<td>Gas constant</td>
</tr>
<tr>
<td>$T_{frac}$</td>
<td>[-]</td>
<td>Thermostat opening</td>
</tr>
<tr>
<td>$A_{rad}$</td>
<td>[m$^2$]</td>
<td>Area radiator</td>
</tr>
</tbody>
</table>
Simulation environment

The simulation environment was build around an existing model. The given model simulates the coolant temperature by describing the heat flow in the motor which is divided into different zones. In the center is the combustion chamber which is enveloped by a engine block denoted hot engine block (zone 1). The coolant (zone 2) is directed through the engine past the hot engine block but also the cooler parts of the engine denoted cooler engine block (zone 3). The coolant is also connected to the EGR-cooler and courses radiator. The different zones and how they are connected through heat transfer can be seen in figure 5.1.

![Figure 5.1: Engine model heat zones](image)

The model for the coolant temperature is built on the energy balance for the three described zones. The equation for the energy balance is:

\[
m_1c_{p,eng}\dot{T}_1 = \dot{Q}_{In} - \dot{Q}_{1-2} \tag{5.1}
\]

\[
m_2c_{p,c}\dot{T}_{eng} = \dot{Q}_{1-2} + \dot{Q}_{EGR} - \dot{Q}_{Rad} - \dot{Q}_{2-3} \tag{5.2}
\]

\[
m_3c_{p,eng}\dot{T}_3 = \dot{Q}_{2-3} - \dot{Q}_{Air} \tag{5.3}
\]

The main goal is to control \( T_{eng} \) which is the temperature of the coolant after the engine, often denoted engine heat. From 5.2 we get the expression for \( T_{eng} \).

\[
T_{eng} = \frac{\int \dot{Q}_{1-2} + \dot{Q}_{EGR} - \dot{Q}_{Rad} - \dot{Q}_{2-3} dt}{m_2c_{p,c}} + T_{eng,0}. \tag{5.4}
\]

Where \( T_{eng,0} \) is the initial engine heat. The heat flow is given by a series of equations;

\[
\dot{Q}_{In} = \dot{m}_{avg}c_{p,gas}(T_{avg} - T_{Air}), \tag{5.5}
\]
\[
\dot{Q}_{1-2} = (UA)_{1-2}(T_1 - T_{eng}), \tag{5.6}
\]
\[
\dot{Q}_{2-3} = (UA)_{2-3}(T_{eng} - T_3), \tag{5.7}
\]
\[
\dot{Q}_{Rad} = (UA)_{Rad}(T_{eng} - T_{Air}), \tag{5.8}
\]
\[
\dot{Q}_{EGR} = \dot{m}_{EGR}c_{p,gas}(T_{avg} - T_{eng}). \tag{5.9}
\]

The \((UA)_x\) terms are the total heat transfer coefficient and is the product of the overall heat transfer coefficient, \(U_x\), and the area of the surface \(A_x\). These terms can be constant, which is the case with \((UA)_{1-2}\), but most of them depend on other parameters as seen below.

\[
(UA)_{2-3} = C_1\dot{m}_{coolant}^{0.8} \tag{5.10}
\]

\[
(UA)_{Rad} = f(\dot{m}_{air}, \dot{m}_{c,coolant}) \tag{5.11}
\]

\[
(UA)_{Air} = C_2 \frac{\dot{m}_{EGR}p_{gas}}{\dot{V}_{air}A_{Rad}^{0.67}} \tag{5.12}
\]

As can be seen the heat flow depends on quite a few parameters. Some are constants, but most are affected by control signals and noise. The most important signals are listed in Table 5.1. \(C_1\) in (5.10) is a calibration variable which comes from the fact that the engine heats up faster when it’s cold, \(C_1\) is thus a function of \(T_{eng}\). With a fixed pump the mass flow coolant is proportional to the engine speed as seen below,

\[
\dot{m}_{coolant} = C_N n_{eng}. \tag{5.13}
\]

The engine speed can be measured so this can be used as input when controlling the system. From (5.11) some more usable inputs can be derived to the controller. First mass flow air through the cooler, \(\dot{m}_{air}\), is calculated like:

\[
\dot{m}_{air} = \dot{V}_{air} \rho_{air}, \tag{5.14}
\]

\[
\dot{V}_{air} = f(v_{veh}, n_{fan}), \tag{5.15}
\]

\[
\dot{m}_{air} = \frac{p_{amb}}{R(T_{amb} + T_{abs})}. \tag{5.16}
\]

All variables which is either measurable or constant. Useful inputs to the controller is the vehicle velocity, \(v_{veh}\), and fan speed \(n_{fan}\) which is the main variables that determine the cooling
capacity. The mass flow through the radiator, $\dot{m}_{c,\text{coolant}}$, is the main variable to be controlled. This is done by controlling the degree of opening of the thermostat, $T_{frac}$. In the mechanical thermostat the degree of opening is proportional to the engine temperature as shown here,

$$\dot{m}_{c,\text{coolant}} = \dot{m}_{\text{coolant}} T_{frac}, \quad (5.17)$$

$$T_{frac} = f(T_{\text{eng}}). \quad (5.18)$$

The last variable that needs to be described in the model is the mass flow gas to the EGR, $\dot{m}_{EGR}$, which affect the system according to (5.9). The logic that controls $\dot{m}_{EGR}$ isn’t described in the model, but it’s measurable during operation of the truck. When the temperature in the engine is changed, the mass flow gas to the EGR is also changed. This affect is however neglected when treating log data in the project. It should be noted that $\dot{m}_{EGR}$ can be equal to zero.