Institutionen för systemteknik
Department of Electrical Engineering

Topography based fan control for heavy trucks

Master's thesis performed in Vehicular Systems
by

Niclas Lerede

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Titel
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Topografibaserad kylfläktsstyrning för tunga fordon
Topography based fan control for heavy trucks

Författare
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Niclas Lerede

Sammanfattning
Abstract

This thesis is a study of how cooling fan control can be improved by using road topography information. Two such controllers are presented, one that uses information available in vehicles produced today, and one that combines GPS-information with digital topographic maps to use information about the road ahead of the vehicle.

Simulations show that significant energy savings can be obtained, especially during warm conditions and hilly roads. Compared to conventional fan controllers, energy consumption can be cut by up to three quarters. Moreover, this is possible without any hardware redesign.

Nyckelord
Keywords

cooling system, fan control, auxiliary control, look ahead, look down

Abstract

This thesis is a study of how cooling fan control can be improved by using road topography information. Two such controllers are presented, one that uses information available in vehicles produced today, and one that combines GPS-information with digital topographic maps to use information about the road ahead of the vehicle.

Simulations show that significant energy savings can be obtained, especially during warm conditions and hilly roads. Compared to conventional fan controllers, energy consumption can be cut by up to three quarters. Moreover, this is possible without any hardware redesign.

Sammanfattning

Rapporten innehåller en analys av hur man med enkla medel kan utnyttja topografisk information om vägen framför fordonet för att styra kylfläkten på ett mer effektivt sätt. Två olika metoder presenteras, en som använder sig av information om vägens kommande topografi och en som endast använder information om nuvarande lutning och förändring av den. Båda strategierna innebär en avseevärd minskning av de energiförluster som orsakas av kylfläkten.

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The sincere will to help and devote time I encountered at Scania CV AB came somewhat as a surprise. Even though I was not yet a regular employee I never felt down prioritized. Before I acknowledge a couple of especially contributing persons I would therefore like to send a general thank to the staff at Scania Technical Centre, Södertälje

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A last thank is devoted to my friends and colleagues who made my time as master thesis worker at Scania CV AB, Södertälje among the best!

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Chapter 1

Introduction

When a driver is about to buy a new heavy truck, fuel consumption is of great interest. It is thus of great interest for truck manufacturers to produce vehicles with a low ditto. Besides economic aspects, which probably are most important to the vehicle owner, there are environmental gains to be obtained as there is a direct connection between consumed fuel and released emissions. This affects truck manufacturers as legislations restrict the allowed amounts of emissions and these restrictions get more stringent every year.

A truck engine uses a number of auxiliary units to perform different tasks. These units consume a rather substantial part of the produced engine power. The control of them is often based on an immediate need, and takes no regard to however it is efficient or not to engage the unit at the given time. The cooling fan for example, engages when the coolant fluid temperature is raised over a threshold, regardless of ambient conditions such as road slope.

If it is possible to use information about the topography ahead of the truck for control of auxiliary units, a decrease of fuel consumption may be achieved, without need of any hardware reconfiguration.

Since computers were introduced in vehicle control systems, a wide range of electronic controllers have replaced their mechanic dittos. One of the latest additions, and an area of bulky research, is the use of "Look Ahead" information, i.e. information about the road ahead of the truck, in cruise controllers. Using a GPS receiver and digitally saved topography maps, a driving strategy that is fuel optimized can be calculated. An even newer area of interest is the use of Look Ahead information to plan the engagement and drive of auxiliary units.

1.1 Look Ahead

An experienced driver adapts his driving behaviour to ambient conditions. It is then possible to reduce both travel time and fuel consumption, by preventively compensate for closing uphill slopes or downhill slopes. This has up until today been a major difference to cruise controllers, which detect uphill slopes first when the truck begins to decelerate due to an increased driving resistance. A driver can on the other hand increase the speed before the slope begins and thereby prevent an expected velocity decrease.

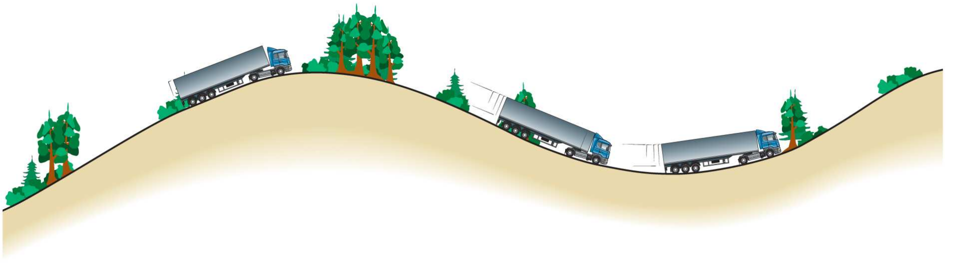

Figure 1.1. Example of a road where Look Ahead information could reduce energy-, and thereby fuel consumption if applied on cruise- or fan control e.g.

One way to make cruise controllers able to mimic the behaviour of a driver is to somehow provide the same information about the road ahead that the driver gets by looking through the windshield. An optimized strategy can then be calculated and run to minimize either fuel consumption or travel time. This is the concept of Look Ahead cruise controllers.

A digital map is stored onboard the truck. It works together with a GPS (global positioning system) receiver. A horizon with information about road slopes is then generated and can be utilized by various applications, e.g. cruise controllers, or in this case auxiliary control.

1.2 Objective, outline and constraints

There are several auxiliary units in a truck, and some of them consume a relatively large amount of the power produced by the engine. The objective of this thesis is partly to estimate the loss of energy with current cooling fan control. The main focus is however to develop strategies to control the cooling fan more efficiently, provided that the topography of the road is known or can be estimated.

This thesis project is focused on control of the cooling fan since it is the largest power consumer that easily can be controlled, among the auxiliaries. It is often engaged in uphill slopes when engine power is needed to propel the vehicle, and heat transfer from engine to coolant is considerable.

The outline of the thesis project is as follows:

- A study on related research is conducted. The purpose is to understand how cooling systems and conventional fan controllers work.
- A cooling system model is implemented, used for control strategy development and evaluation. It is then validated against an other, already existing model.
- The energy consumption caused by the fan with a conventional controller is investigated
- Two new fan control strategies are developed and evaluated against each other and against the conventional controller
- A study on optimal fan control on a predefined route, using dynamic programming, is conducted. The results are used to further evaluate the new control strategies.

A number of constraints are necessary to define the thesis project. It is assumed that no hardware should be modified in order to achieve an improved control. It is also assumed that topography information, drive resistance, vehicle- and engine speed and engine- and retarder torque are provided by already existing models.

1.3 Related research

An area that has been thoroughly researched is cooling systems and control of them. One work is presented in [7] where optimal control is used to control the water pump and cooling fan. Other auxiliaries are also analyzed and simulations show that the total auxiliary energy consumption is between 4.5 % and 8 % of the total consumption. Auxiliaries are often mechanically driven in heavy vehicles today. An electrically driven auxiliary can at a given time be controlled to match the current need, regardless of engine speed, and it is shown that significant energy savings can be obtained.

Paper [6] presents a similar study where the possible gains with a variable speed cooling fan is investigated. A simple model of the cooling system is used to simulate the system. Several control strategies for cooling systems are also presented and evaluated. The study is done for Valeo Engine Cooling. Another paper from the same company is [8]. It explains how different components in the engine compartment affect the cooling system, and a simple simulation model is presented. Differences between tests in laboratories and on road are also discussed.

Two works concerning estimation of engine coolant temperature to be able to detect and isolate failures, i.e. diagnosis applications, are presented in [5] and [4].

Paper [5] is a study made in cooperation with Scania CV AB and is focused on the cooling system in a heavy truck engine. A simulation model has been implemented in Simulink and validated against measured data. The authors however conclude that the model have its shortcomings and is best suited for desktop simulations. There are no models for the charge air cooler, the retarder or the fan control among others, thus creating a need of collected data, used to simulate the cooling model. Paper [4] contains a model over a regular internal combustion engine, thus lacking heavy truck auxiliaries such as a retarder brake. Both models have fairly good accordance with real systems, shown in validations.

Look Ahead is a rather novel area of research. There is however work presented mainly investigating usage in cruise controllers. Paper [11] presents three methods which reduce fuel consumption up to 3.4 %. The first method uses dynamic programming, the second piecewise linearization and linear optimization and the third is a more intuitive rule based method, controlling the vehicle according to certain strategies between specified positions in a climb or a descent.

Another work concerning Look Ahead is presented in [3]. Two aspects of implementation of Look Ahead cruise controllers are analyzed: road slope information and control sensitivity. A filter is derived that estimates road slope using a GPS receiver and standard heavy truck sensors. A sensitivity analysis of two controllers using different control strategies is also done. It concludes that errors in estimated vehicle mass or road slope affect the control performance. Control errors are a direct result of chosen control strategy approach.

Paper [2] is another study of a Look Ahead cruise controller. The algorithm uses dynamic programming and conducted experiments show that significant fuel consumption reductions can be achieved, with no or negligible time losses.

A cooling fan model is presented in [10]. It explains the behaviour of both mechanically and electrically driven fans. The intended use is simulations of engine compartment airflows.

1.4 Expected results

According to [7], the energy consumed by the auxiliaries is in the range of 4.5 % to 8 % of the total produced energy. That is of course the theoretical maximum of how much that can be saved with enhanced control. It is however in practical means not possible to have an auxiliary energy consumption of 0 %. First, some of the auxiliaries are directly connected to the driveline, making it impossible to affect their energy consumption without physically redesigning them and that is not part of this study. Further, some auxiliaries can for safety reasons not be active in descents or during braking only, e.g. the power steering pump.

It is however possible to control e.g. the cooling system so that fan drive during inefficient conditions is minimised. During a long ascent with 2 % inclination and an ambient temperature of 20° C the cooling system consumes barely 1 % of the total produced energy. When the fan is not engaged, the cooling system consumes approximately 0.05 % of the produced energy during a drive in the same ascent. The potential energy saving is thus 0.95 %, which is not negligible.

An energy reduction is expected, in what extent is however not stated at this point. It is also expected to get a more logic behaviour from the driver's point of view. It doesn't seem logic if the fan is engaged just before a crest e.g.

1.5 Report outline

The thesis presents a study over power consumption caused by the cooling fan, and two new control strategies that use information about road topography.

In Chapter 2, theory and simulation models are presented. The different components that contribute to the thermal dynamic in the cooling system are described.

Chapter 3 presents the controllers, starting with a conventional system used in vehicles today and followed by the controllers that have been developed during the proceeding of this thesis project. They are individually evaluated and compared to each other.

Summary, conclusions and suggestions for future work are presented in Chapter 4.

Chapter 2

Theory and Models

To be able to evaluate and compare the different control strategies, a simulation environment is needed. It consists of a set of physically derived mathematical models and measured maps. These models, and the theory that describe them are presented in this chapter.

2.1 Vehicle model

Figure 2.1. A Scania tractor with a semitrailer on a bridge in Austria, similar to the vehicle setup used in simulations

Component	**Characteristic**	**Type**
Vehicle	Tractor-semitrailer 40 tons G.W.	Scania R440
Engine	13 litre, 6 cylinder 440 hp, Euro 5	DC1310
Gear box	Scania Opticruise, 12 speeds with retarder	GRS895R

Table 2.1. Specification of the vehicle used in simulations

The chosen vehicle setup consists of a tractor semi-trailer combination, powered by a 13 litre 440 hp Euro 5 engine equipped with EGR (exhaust gas recirculation). The gross weight is 40 tons. The specifications are summarized in Table 2.1. No powertrain is modelled but the vehicle responds to the equation of motion, $\sum F = ma$. The forces acting on the vehicle are:

$$\begin{aligned}
\text{Engine: } & F_{eng} \\
\text{Brakes (retarder \& pedal): } & F_{brake} \\
\text{Fan: } & F_{fan} \\
\text{Air resistance: } & F_{air} = \frac{\rho_{air} A c_D v_{veh}^2}{2} \\
\text{Rolling resistance: } & F_{roll} = m(c_{r1} + c_{r2} v_{veh}) \\
\text{Gravitational force: } & F_{grav} = mg\sin\theta
\end{aligned} \tag{2.1}$$

where ρ_{air} is air density, A vehicle frontal area, c_D air resistance coefficient, v_{veh} vehicle speed, m vehicle mass, c_{r1}, c_{r2} rolling resistance coefficients, g gravitational acceleration and θ road angle. Further:

$$\begin{aligned}
v_{veh} &= \int \frac{\sum F}{m} \\
\omega_{eng} &= \frac{v_{veh}}{r_{wheel} i_{gear}}
\end{aligned}$$

where r_{wheel} is wheel radius and i_{gear} is gear ratio (transmission and final gear).

2.2 Drive cycles

Two different drive cycles are used to analyze and compare the controllers. They are both representative for long haulage traffic in Europe and consist of highway driving only. It is assumed that the vehicle holds a set speed of 80 km/h when it is possible. The speed varies however due to road inclination. When driving in steep uphill slopes, the engine can not deliver enough power to maintain the set speed. When driving in downhill slopes, the retarder is engaged when the vehicle speed reaches 89 km/h. This is assumed to be the highest allowed vehicle speed.

The two road sections are between the German cities of Koblenz and Trier, and the Swedish cities of Södertälje and Norrköping, respectively. They are both often used for reference driving within Scania development. The first section contains some rather steep uphill slopes and downhill slopes combined with more flat sections. The second route contains a more even distribution of road inclinations. The topography profile of both cycles are presented in Figure 2.2.

As a result of the different characteristics of the routes, fan usage differs rather much. The fan is for example never engaged during a drive between Södertälje and Norrköping at an ambient temperature of 20 °C, while it is so on several occasions on the other route.

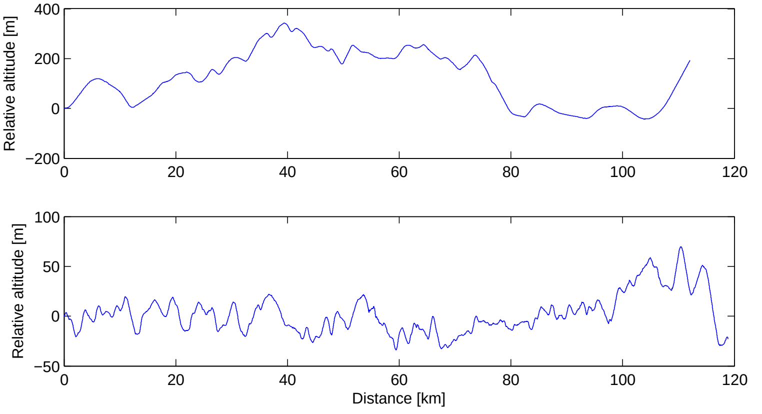

Figure 2.2. Road profile of Koblenz-Trier (upper) and Södertälje-Norrköping (lower) routes

2.3 Cooling system

The cooling system consists of several components. They can essentially be divided into heat sources and heat sinks. A heat source generates heat that is accumulated by the system and a heat sink releases heat from the system. To be able to understand how a cooling system works, relevant thermal theory is presented in Section 2.3.1. The following section presents the model and how the different components affect the thermal dynamic. A validation is presented in Section 2.3.3, while Section 2.3.4 describes a one state cooling system model. Finally, a short introduction to dynamic programming is presented in Section 2.4.

2.3.1 Theory

This section presents the basic theory that is needed to explain the behaviour of a cooling system. The first law of thermodynamics, different ways to transfer heat and how a heat exchanger works are described.

Thermal dynamics To explain the thermal behaviour of a system, control volumes are used (Figure 2.3). They are isolated against ambient conditions and only affected by energy flows in and out through the control surface. Thermal changes in a control volume are described by the first law of thermodynamics, Equation (2.2).

$$\dot{Q} - \dot{W} = \dot{m}_2 h_2 - \dot{m}_1 h_1 + \frac{dE}{dt} \qquad (2.2)$$

where $\dot{Q}$ and $\dot{W}$ are transferred heat and performed work respectively, $\dot{m}_1$ and $\dot{m}_2$ are mass flows into and out from the volume respectively, and h_1 and h_2 the corresponding enthalpys. It can be assumed that the two mass flows are of same

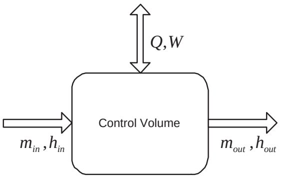

Figure 2.3. Sketch of a control volume

size, i.e. $\dot{m} = \dot{m}_1 = \dot{m}_2$. It can also be assumed that there is no change of potential or kinetic energy within the control volume. Finally, if the relation $\Delta h = c_p \Delta T$ is applied, Equation (2.2) can be rewritten as:

$$\dot{Q} - \dot{W} = \dot{m} c_p (T_2 - T_1) \tag{2.3}$$

where c_p is the specific heat coefficient at constant pressure. Equation (2.3) is applied in several submodels of the cooling system model.

Heat transfer There are three ways to transfer heat between and within bodies. They all depend on a temperature difference:

- *Conduction*
  Every collection of elements, whether it is a solid body or a gas, strives to get an energy balance. If there is a temperature difference within such a collection, energy will be transferred until there is a homogeneous temperature. This is called conduction.

- *Convection*
  Two elements that are physically in contact with each other and have different temperatures have an energy flow that strives to get the energy balanced. Convection is divided into natural and forced dittos. Forced convection implies that there is a relative velocity between the two elements which originates from an outer influence.

- *Radiation*
  Every body that has a temperature above the absolute zero ($-273.15\,°\mathrm{C}$), emits heat through radiation, i.e. electromagnetic waves. No medium is required to transfer the heat.

It is assumed that a solid body holds a homogeneous temperature, and that radiation can be neglected at the temperature boundaries that the cooling system works within.

$$\begin{aligned} & \dot{Q}_{ref} = \varepsilon \sigma A T^4 \text{ where} \\ & \qquad \varepsilon \epsilon [0,1] \text{ and } \sigma \approx 5.7 \cdot 10^{-8} \left[\frac{W}{m^2 K^4} \right] \\ \Rightarrow \quad & \dot{Q}_{rad} \ll \dot{Q}_{conv}, \text{ when } T \epsilon [80, 120]\,°\mathrm{C} \end{aligned}$$

Then, the heat flow through a control volume can be expressed as Equation (2.4), i.e. convection is the only way heat is transferred:

$$\dot{Q} = \lambda A(T_{wall} - T_{fluid}) \tag{2.4}$$

where λ is the heat transfer coefficient, A is the area through which the energy is transferred, T_{wall} is the wall temperature and T_{fluid} is the fluid temperature.

Heat exchanger The following relationship applies to heat exchangers, according to [9],

$$\dot{m}_{warm}(h_{i,warm} - h_{o,warm}) = \dot{m}_{cool}(h_{o,cool} - h_{i,cool})$$

where $\dot{m}$ are massflows and h enthalpys. The subscripts *warm* and *cold* corresponds to the cooled and cooling flows respectively. i and o correspond to flows into and out from the control volume. By replacing the enthalpys with the products of specific heat and temperature, Equation 2.5 can be derived.

$$\dot{m}_{warm}c_p(T_{i,warm} - T_{o,warm}) = \dot{m}_{cool}c_p(T_{o,cool} - T_{i,cool}) \tag{2.5}$$

Equation 2.5 is in all essentials the same as $\dot{Q}_{in} = \dot{Q}_{out}$.

2.3.2 Components and Models

The mathematic formulas used in the implementation of the cooling system model are taken from an existing cooling system model, implemented in an other simulation language, used at Scania today.

The cooling system model is implemented in SIMULINK and built by blocks representing the different physical components. Figure 2.5 at Page 15 shows the top level. The blocks are connected in series, just as the components in an actual vehicle. One should be aware of the several simplifications that are necessary to get a model that is possible to use. However, Section 2.3.3 contains a validation of the model which clearly shows that the accuracy is good, despite the mentioned simplifications. Figure 2.4 shows the principle layout of a cooling system.

Engine and Retarder The diesel engines used in Scania vehicles today are four stroke engines, where each stroke corresponds to a piston translation. The process is divided into four phaces:

1. **Intake**
   Air is injected into the cylinders while the piston moves down

2. **Compression**
   The air is compressed as the piston moves upwards.

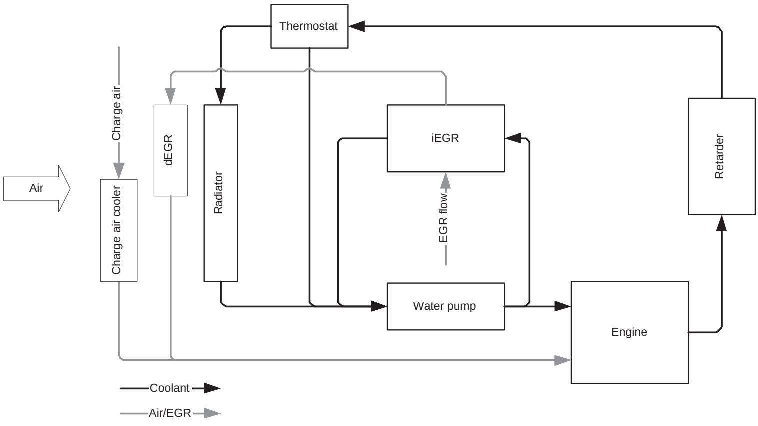

Figure 2.4. Principle layout of the cooling system. Each block represents a hardware component, and each line connecting two blocks represents a flow of either coolant (black) or gas (grey).

3. **Expansion**
   Fuel is injected into the cylinders, the combustion process begins and the gas expansion pushes the pistons downwards

4. **Exhaust**
   The exhaust gases are evacuated as the pistons move upwards and the process starts over again

Less than half of the energy that is extracted from the diesel through the combustion process is converted to kinetic energy, while the rest is converted to heat. The heat dissipates to the ambient air through exhaust gases, to the engine block and to the engine oil.

The retarder is a hydraulic brake connected to the power train at the end of the gearbox. When it is engaged, oil is injected to a chamber and forced to circulate through vanes. This causes a resistance that is converted to braking force. As the oil circulates it is heated by viscous friction and the heat is emitted to the cooling system through an *oil to coolant* heat exchanger.

Both the engine and retarder are consequently heat sources to the cooling system and are modelled according to the same principle. A mass is heated by energy losses and transfers heat to the cooling fluid. The temperature dynamic of the engine block and retarder oil are modelled as (compare with Equations (2.3) & (2.4)):

$$m_{block}c_{p,block}\dot{T}_{block} = P_{eng} + \lambda_{block}(T_{engcool} - T_{block}) \tag{2.6}$$

$$m_{retoil}c_{p,retoil}\dot{T}_{retoil} = P_{ret} + \lambda_{retoil}(T_{retcool} - T_{retoil}) \tag{2.7}$$

where m_{block} and m_{retoil} are the engine block mass and retarder oil mass, re-

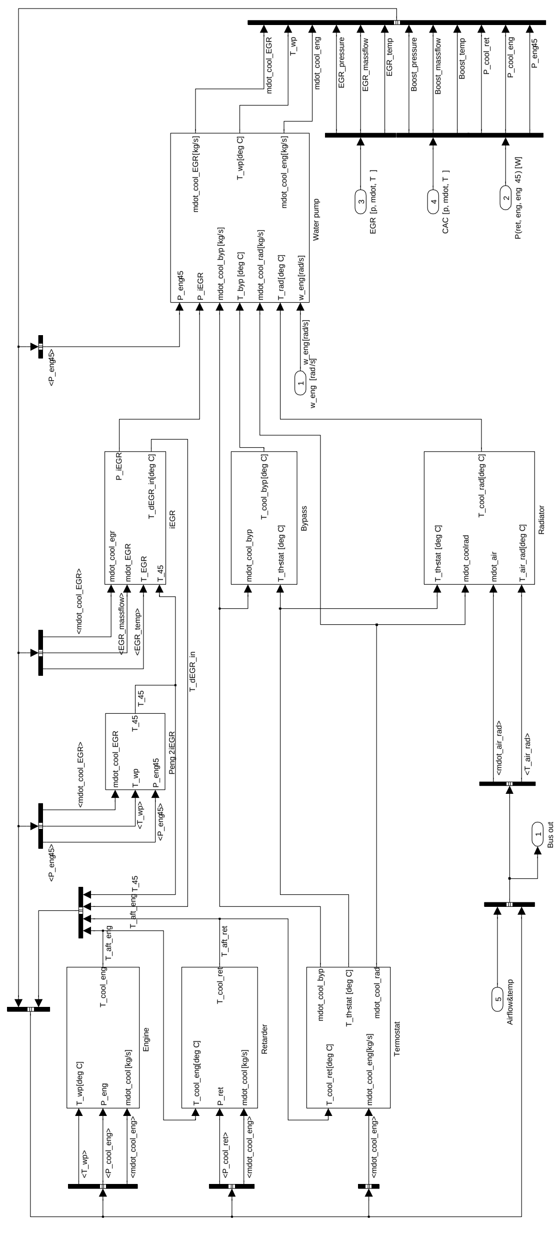

Figure 2.5. The cooling system model implemented in SIMULINK

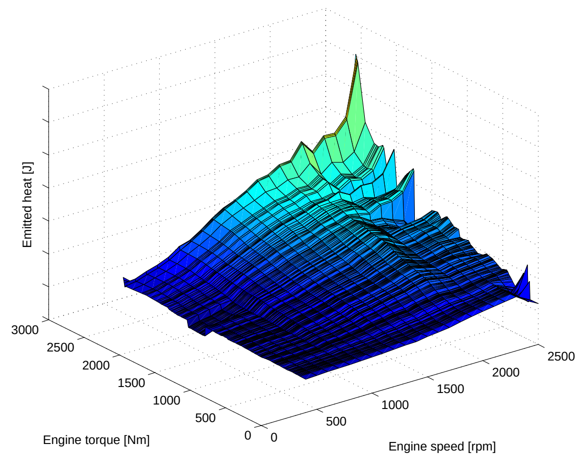

Figure 2.6. Heat power transfer map from engine to coolant

spectively, P_{eng} and P_{ret} are heat losses from the combustion process and from the retarder brake respectively, and are represented by predefined maps. Figure 2.6 shows the *engine to coolant* power transfer at different working points. $\lambda_{block}(T_{engcool} - T_{block})$ and $\lambda_{retoil}(T_{retcool} - T_{retoil})$ represent the heat transfer to the coolant. The corresponding equations for temperature dynamic of the coolant in the engine and retarder are:

$$\begin{aligned} m_{eng,cool}c_{p,cool}\dot{T}_{eng,cool} &= \dot{m}_{engcool}c_{p,cool}(T_{wp} - T_{eng,cool}) \\ &\quad + \lambda_{eng,block}(T_{eng,block} - T_{eng,cool}) \qquad (2.8) \\ m_{ret,cool}c_{p,cool}\dot{T}_{ret,cool} &= \dot{m}_{ret,cool}c_{p,cool}(T_{eng,cool} - T_{ret,cool}) \\ &\quad + \lambda_{ret,oil}(T_{ret,oil} - T_{ret,cool}) \qquad (2.9) \end{aligned}$$

where $m_{eng,cool}$ and $m_{ret,cool}$ are the mass of coolant in the engine and retarder respectively.

Thermostat The thermostat controls the coolant flow. When it is fully open, all coolant passes through the radiator, and when it is fully closed the coolant passes through the bypass pipe. The thermostat is a passive component: a wax body is expanded or compressed when heated or cooled by the coolant, thus affecting the coolant path. Equations (2.10) and (2.11) are used to calculate the wax and coolant temperature dynamics respectively:

$$m_{wax}c_{p,wax}\dot{T}_{wax} = \lambda_{wax}(T_{tstat,cool} - T_{wax}) \qquad (2.10)$$

$$\begin{aligned} m_{tstat,cool}c_{p,cool}\dot{T}_{tstat,cool} &= \dot{m}_{tstat,cool}c_{p,cool}(T_{ret,cool} - T_{tstat,cool}) \\ &\quad - \lambda_{wax}(T_{tstat,cool} - T_{wax}) \qquad (2.11) \end{aligned}$$

where m_{wax} and $m_{tstat,cool}$ is the mass of the wax body and coolant in the thermostat respectively, $c_{p,wax}$ is the specific heat of wax and λ_{wax} is the wax heat transfer coefficient.

The thermostat open degree is determined as a function of T_{wax} and illustrated in Figure 2.7. The coolant flow through the thermostat is the same as through the engine and retarder, i.e. $\dot{m}_{tstat,cool} = \dot{m}_{eng,cool}$. As a result, the two coolant flows exiting the thermostat are derived as:

$$\begin{aligned} \dot{m}_{byp} &= f(T_{wax})\dot{m}_{tstat,cool} \\ \dot{m}_{rad} &= (1 - f(T_{wax}))\dot{m}_{tstat,cool} \end{aligned} \tag{2.12}$$

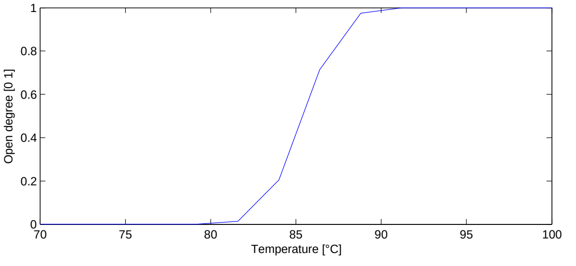

Figure 2.7. Part of coolant that is guided through the radiator as function of coolant temperature

Radiator, iEGR and Bypass There are two *fluid-to-gas* heat exchangers: the radiator and iEGR. For both, the conductive heat transfer is modelled as:

$$P_{cond} = (T_{inner} - T_{outer})P_{spec}(\dot{m}_{outer}, \dot{m}_{inner}) \tag{2.13}$$

where $P_{spec}(\dot{m}_{outer}, \dot{m}_{inner})$ is a mapped function.

The radiator is the most important heat sink in the cooling system. Its only purpose is to lower the coolant temperature by transferring heat to air that passes through. The heat exchange is a result of the temperature difference between the coolant in the radiator and the surrounding air. Note that even if it is the air that acts as coolant in the radiator, it is still the cooling *fluid* that is referred to as *coolant* (*cool* in equations). The temperature of the coolant is modelled as:

$$\begin{aligned} m_{rad,cool}c_{p,cool}\dot{T}_{rad,cool} &= \dot{m}_{rad,cool}c_{p,cool}(T_{tstat,cool} - T_{rad,cool}) - P_{rad,cool} \\ P_{rad,cool} &= (T_{tstat,cool} - T_{rad,air})P_{rad,spec}(\dot{m}_{rad,cool}, \dot{m}_{rad,air}) \end{aligned} \tag{2.14}$$

where $T_{tstat,cool}$ and $T_{rad,cool}$ are temperatures of the coolant entering and exiting the radiator, respectively. $\dot{m}_{rad,cool}$ and $\dot{m}_{rad,air}$ are the coolant flow and air flow through the radiator respectively. Finally, $m_{rad,cool}$ is the mass of coolant that continuously stays in the radiator.

EGR, or *exhaust gas recirculation*, is a concept of recirculating exhaust gases to the intake manifold. This reduces the oxygen level of the combustion gases thus reducing NO_x emissions. The amount of exhaust gases that is recirculated is controlled by a valve. When the exhaust gases exits the combustion chamber, they can be as warm as 700 °C and must consequently be cooled before mixed with the intake air. This takes place in the iEGR heat exchanger where heat is transferred from the exhaust gases to the coolant, thus acting as a heat source to the cooling system. The equation below is directly derived from Equation (2.13).

$$\begin{aligned} P_{iEGR} &= (T_{EGR} - T_{45})P_{iEGR,spec}(\dot{m}_{iEGR,cool}, \dot{m}_{EGR}) \\ T_{45} &= T_{wp} + \frac{P_{eng,45}}{\dot{m}_{iEGR,cool}, c_{p,cool}} \end{aligned} \tag{2.15}$$

where $P_{eng,45}$ is heat transfer between engine and iEGR coolant flow. It is assumed that it corresponds to a fix fraction of P_{eng}, i.e. $P_{eng,45} = kP_{eng}$.

The part of coolant that does not flow through the radiator flows through the bypass pipe. No significant heat transfer is conducted and the temperature equation is as follows.

$$m_{byp,cool}c_{p,cool}\dot{T}_{byp,cool} = \dot{m}_{byp,cool}c_{p,cool}(T_{tstat,cool} - T_{byp,cool}) \tag{2.16}$$

Water pump The water pump circulates the coolant through the cooling system. It is connected to the engine crank shaft, thus creating a direct connection between engine speed and coolant flow. The flow is modelled as two separate flows, i.e. $\dot{m}_{eng,cool}$ and $\dot{m}_{iEGR,cool}$.

$$\begin{aligned} \dot{m}_{eng,cool} &= k_e\omega_{eng} \\ \dot{m}_{iEGR,cool} &= k_E\omega_{eng} \end{aligned}$$

where $\dot{m}_{eng,cool}$ and $\dot{m}_{iEGR,cool}$ are coolant flows through the engine and iEGR respectively and k_e and k_E are constant parameters fitted to match the true coolant flow.

The water pump is modelled as a mass less unit, thus containing no temperature dynamic. The temperature is instead calculated as a sum of incoming heat powers, shown below.

$$\begin{aligned} \dot{m}_{eng,cool}c_{p,cool}T_{wp} = {} & \dot{m}_{byp}c_{p,cool}T_{byp,cool} \\ & + \dot{m}_{rad}c_{p,cool}T_{rad} \\ & + P_{iEGR} + P_{eng,45} \end{aligned} \tag{2.17}$$

CAC and dEGR There are two *gas-to-gas* heat exchangers: the CAC (*Charge Air Cooler*) and dEGR. For both, the conductive heat transfer is modelled as:

$$P_{cond} = (T_{inner} - T_{outer})P_{spec}(\dot{m}_{outer}, \dot{m}_{inner}) \tag{2.18}$$

where $P_{spec}(\dot{m}_{outer}, \dot{m}_{inner})$ is a mapped function. Note that it is the outer air that is referred to as coolant (*cool* in equations) for these components. The CAC and dEGR are both placed in front of the radiator. It is thus necessary to model the cooling air temperature rise through them to be able to calculate the heat transfer in the radiator correctly.

When the truck is driven at low velocities, a phenomenon called recirculation appears. There is not sufficient ram air to ventilate the engine compartment, causing a temperature rise of the cooling air that flows through the different heat exchangers. To take this in account the front temperature is calculated as:

$$T_{front} = T_{amb} + T_{recirc}(v_{veh}, \omega_{eng}) \tag{2.19}$$

where $T_{recirc}(v_{veh}, \omega_{eng})$ is a function of vehicle- and engine speed.

Before the intake air is injected into the cylinders, it is compressed. This results in a higher pressure but also a higher temperature. Density is reversely proportional to temperature ($\rho = \frac{m}{V} = \frac{p}{RT}$). To further increase the density, the air is cooled in the CAC. This results in a heat transfer:

$$\begin{aligned} P_{cac} &= (T_{inlet} - T_{front})P_{cac,spec}(\dot{m}_{cac}, \dot{m}_{cac,air}) \\ T_{inlet} &= \frac{T_{cac}\dot{m}_{cac}c_{p,cac} - P_{cac}}{\dot{m}_{cac}c_{p,cac}} \end{aligned} \tag{2.20}$$

where P_{cac} is the heat transferred to the cooling air, $\dot{m}_{cac}$ is the intake air mass flow through the intercooler (CAC) and $c_{p,cac}$ is the intake air specific heat constant. The model is derived from Equation (2.18).

Before the EGR-gas is redistributed to the combustion process it has to be cooled. This is done in two consecutive heat exchangers: iEGR and dEGR. The iEGR transfers heat to the cooling circuit and the dEGR transfers heat to the front air flow. The equation below is directly derived from Equation (2.18)

$$\begin{aligned} P_{dEGR} &= (T_{dEGR} - T_{front})P_{dEGR,spec}(\dot{m}_{EGR}, \dot{m}_{dEGR,air}) \\ T_{dEGR} &= \frac{T_{EGR}\dot{m}_{EGR}c_{p,EGR} - P_{iEGR}}{\dot{m}_{EGR}c_{p,EGR}} \end{aligned} \tag{2.21}$$

where P_{dEGR} is the heat transferred to the cooling air, $\dot{m}_{EGR}$ is the EGR mass flow and $c_{p,EGR}$ is the EGR specific heat constant.

Air flow and temperature The air flow through the different heat exchangers is a product of ram air and fan speed. Ram air originates from a pressure rise that appears in front of the vehicle when it is driven forward. If the ram air is not sufficient enough to cool the coolant, the fan is engaged, resulting in an increased air mass flow through the radiator. This is represented by a map of measured data, having vehicle speed and fan speed as input, and air volume flow as output.

$$\dot{m}_{air} = f(v_{veh}, \omega_{fan})\rho_{air}$$

ρ_{air} is air density, which is temperature and pressure dependent. The air flow through the three different heat exchangers (CAC, dEGR and Radiator) are separetely calculated. The fan is connected to the engine crank shaft via a hydraulic clutch and a gear. It has its own controller that regulates the oil pressure in the clutch to maintain the demanded fan speed. The fan can not be driven faster than the engine (multiplied with the gear ratio and minus a minimum clutch slip). It can neither be totally still, but is by friction force always driven with a minimum speed.

Since there is a controller regulating the fan speed to meet a demanded speed, fan speed is modelled as a time delay to demanded speed. There are two sources of power losses coupled to fan drive: air resistance and frictional losses in the clutch. They are modelled as below, and integrated to calculate the energy loss.

$$\begin{aligned}
\omega_{fan} &= \frac{1}{Ts+1}\omega_{fan,ref} \\
P_{fan} &= k_{pf}\omega_{fan}^3 \\
P_{clutch} &= k_{pf}\omega_{fan}^2(i_{fan}\omega_{eng} - \omega_{fan}) \\
E_{loss} &= \int (P_{fan} + P_{clutch})dt
\end{aligned} \tag{2.22}$$

where T is a time constant used to simulate the fan inertia, k_{pf} is a constant fitted to the chosen fan and i_{fan} is the fan gear ratio. Equation (2.22) is used to calculate the energy consumed by the fan.

The air entering the radiator is heated when passing through the dEGR and CAC coolers, and at low velocities by engine compartment recirculation. Equation (2.23) is used to calculate the temperature.

$$T_{air} = T_{front} + \frac{Q_{dEGR} + Q_{CAC}}{c_{p,air}\dot{m}_{air}} \tag{2.23}$$

where $c_{p,air}$ is specific heat of air and T_{front} is calculated as previously shown in equation (2.19).

2.3.3 Model validation

There is nothing like a perfect model, they are always more or less complex imitations of a real system. More complex models tend to be more accurate with worse calculation performance as natural drawback, i.e. slower simulations. A trade-off between accuracy and calculation performance is hence necessary. This section presents a validation of the used cooling system model against the model used at Scania for other purposes, which in turn is validated against measured vehicle data.

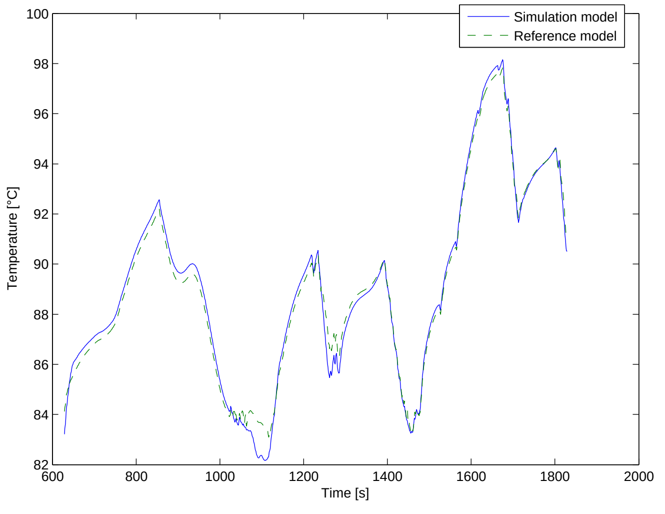

Figure 2.8. Cooling system model validation. Data are recorded from simulations with the cooling system model used at Scania today, and compared to the result of a simulation with the cooling system model presented in this paper. The figure presents a comparison of coolant temperatures and the accuracy is as seen quite good.

As seen in Figure 2.8, the simulation model follows the reference ditto well. Data were recorded during simulations at the Koblenz Trier route with both models respectively. There are anomalies, especially after rapid temperature descents. This is a result of a different way to calculate the air mass flow, compared to the reference model.

2.3.4 One state cooling system model

In order to find an optimal control of the cooling fan, a cooling system model is necessary. The one presented above describes the system fairly exact. It is however too complex for use in control design since it would generate an unreasonably complex optimization problem. This section presents an one state cooling model, taken from [7] and slightly modified. Please refer to [7] for a more thorough derivation of the equations.

The temperature dynamic is based on energy balance. Heat generated by the different heat sources minus heat released in the radiator.

$$\dot{T} = c_1\dot{Q}_{eng,ret} - c_2\Delta\tau\phi_s \tag{2.24}$$

where T is the coolant temperature minus the ambient temperature, c_1 and c_2 are scaling factors, $\Delta\tau$ is temperature difference between cooling air and coolant. ϕ_s is specific heat in the radiator and $\dot{Q}_{eng,ret}$ is heat generated by engine, retarder and iEGR and transferred to the coolant. Equation (2.24) can be broken down to the following two equations:

$$\begin{aligned}
\Delta\tau &= T - \frac{c_3\dot{Q}_{CAC,dEGR}}{\dot{m}_{air}} && (2.25)\\
\phi_s &= \frac{\dot{m}_{cool}\dot{m}_{air}}{c_4\dot{m}_{cool} + \dot{m}_{cool}\dot{m}_{air} + c_5\dot{m}_{air}} && (2.26)\\
\dot{m}_{air} &= a_1\omega_{fan}^2 + a_2\omega_{fan} + a_3v_{veh} + a_4 && (2.27)\\
\dot{m}_{cool} &= a_5\omega_{eng}
\end{aligned}$$

where $c_3 - c_5$ and $a_1 - a_5$ are constant parameters fitted to match the true system, $\dot{Q}_{CAC,dEGR}$ is heat transferred from CAC and dEGR to the cooling air, $\dot{m}_{cool}$ is the coolant flow, $\dot{m}_{air}$ is the cooling air mass flow, ω_{fan} is fan speed, ω_{eng} is engine speed and v_{veh} is vehicle speed. This results in Equation (2.28).

$$\dot{T} = c_1\dot{Q}_{eng,ret} - c_2(T - \frac{c_3\dot{Q}_{CAC,dEGR}}{\dot{m}_{air}})\frac{\dot{m}_{cool}\dot{m}_{air}}{c_4\dot{m}_{cool} + \dot{m}_{cool}\dot{m}_{air} + c_5\dot{m}_{air}} \tag{2.28}$$

It is obvious that a model like the one just presented must have its shortcomings. It captures the main dynamic of the cooling system, but the accuracy is not very good. There is also an important simplification that has to be done to be able to use the one state model. Since the coolant temperature in the thermostat is unknown, it is not possible to correctly estimate the open degree. The model is consequently modified so that all of the coolant flow always is run through the radiator. This results in lower coolant temperatures when engine and retarder heat emission is low, and is hence a significant anomaly from the true system.

As stated above, the main dynamic is captured by the one state model. There are however anomalies, especially when the retarder is used and emits heat, which for example is the case between 400 s and 600 s in Figure 2.9.

2.4 Dynamic programming

Dynamic programming[1] is a method of solving optimization problems. A complex problem is broken down to a sequence of less complex dittos. In this application, every sequence correspond to a time step, and has a number of states attached

[1] *programming* has in these applications no direct connection to computer programming, but comes from *mathematical programming*, i.e. optimization

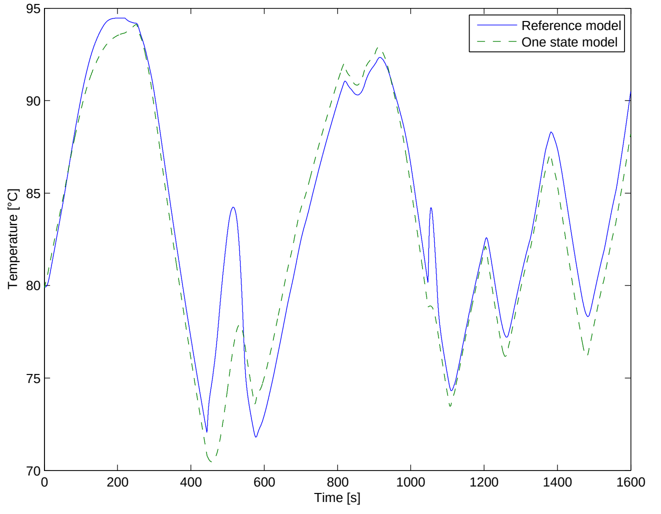

Figure 2.9. Validation of one state cooling model. As seen, the main dynamic is captured but the accuracy is not very good, which could be expected when using such a simplified model.

to it. The choice of states and their constraints will affect the result and calculation performance, since there is an exponential relation between the number of states, and the calculation time required to solve the problem. A span between a maximum- and a minumum allowed temperature is divided into a number of intervals, where each value represent a state.

To be able to solve the optimization problem, Equation (2.24) has to be discretisized. This is done with Euler's method $\frac{d}{dt}(T) = \frac{T_{k+1}-T_k}{h_k}$, and a step size ($h_k$) of 2 s is used. All data are then sampled at 2 s when recorded. This results in Equation (2.29):

$$T_{k+1} = T_k + h_k(c_1\dot{Q}_{k,eng,ret} - c_2\Delta\tau_k\phi_{k,s}) \tag{2.29}$$

For each time step, the air mass flow needed to change the coolant temperature by a given value is calculated through Equation (2.29), combined with Equation (2.26). A corresponding fan speed needed to generate the air mass flow is then calculated with Equation (2.27) and used in the cost function (compare to Equations (2.22)):

$$J = \begin{cases} 0 & \text{in downhill slopes with} \\ & \text{retarder usage} \\ k_{pf}(\omega_{fan}^3 + \omega_{fan}^2(i_{fan}\omega_{eng} - \omega_{fan})) & \text{otherwise} \end{cases}$$

The cost function is later used to find the *cheapest* (most efficient) path between the first- and final time steps. Every change of states has a *cost* attached to it, calculated as below:

$$ctg(k, j) = \min_{\forall j}(J(k, j) + ctg(k + 1, j)) \tag{2.30}$$

where $J(k, j)$ corresponds to the cost for the actual step, and $ctg(k + 1, j)$ (*cost to go*) is the accumulated cost to optimally reach the final time step, from step $k + 1$. The later is already calculated, since the algorithm solves the problem *back to front*. The results are stored in a matrix, depicted in Figure 2.10.

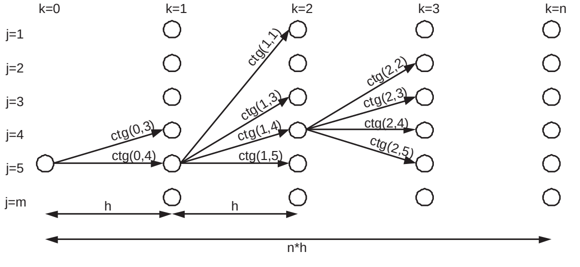

Figure 2.10. Sketch showing the principle layout of the optimization problem. $ctg(k, j)$ represent the *cost* that corresponds to a change from current state to state j during time step k to $k + 1$. It is calculated as the cost for the step itself plus the *cheapest* path from $k + 1$, see Equation (2.30).

At last, the solution is unravelled and a certain fan speed is associated to every time step. The problem is very similar to the *shortest path problem*, i.e. the problem of finding the path between two nodes where the sum of costs of its constituent edges is minimized.

The formal optimization problem is presented below:

$$\begin{aligned} & \min \sum_{k=1}^{n} J(k, j) \\ & \text{subject to} \\ & \qquad T_{min} < T_{coolant} < T_{max} \\ & \qquad T_1 = T_{start} \\ & \qquad T_n \leq T_{final} \\ & \qquad \omega_{fan,min} < \omega_{fan} < \omega_{fan,max} \end{aligned}$$

Chapter 3

Controllers

The basic idea of the alternative control strategies is that it is more efficient to use the fan in descents than ascents. No engine power has to be used to propel the fan. Ideally, the fan is only driven in descents where the retarder is engaged due to too high vehicle speeds. Then no kinetic energy is lost, since there is a surplus that has to be removed anyway.

This chapter describes a conventional control strategy and two alternative strategies. They are then evaluated and compared.

3.1 Conventional fan control

There are five different components that can demand fan in a conventional system: engine, retarder, boost air system, climate control system[1] and external devices[2]. The control is mainly based on current coolant temperature and an immediate need. A fan speed reference is calculated through static temperature- and engine speed maps. The system is in this way very robust to failures. Since these kinds of systems are designed with worst case scenarios in mind, there is a rather large margin that causes unnecessary energy losses during normal conditions.

An adjusted conventional fan controller is presented in Section 3.4.5. The temperature maps in a regular controller has been modified so that the fan is engaged at a higher temperature. This reduces energy consumption but causes higher temperature tops, thus increasing material wear. It is compared to a regular conventional fan controller and the Look Ahead fan controller, presented below.

[1]The climate control system can not demand fan in the cooling system model, presented in Chapter 2. It is complex to estimate the cabin temperature since it depends on a vast number of conditions and is hence somewhat stochastic.

[2]No external devices are considered in the cooling system model, presented in Chapter 2

3.2 Look Down fan control

The Look Down fan controller is the first step towards a more intelligent system. In addition to temperature, it uses information about current conditions such as road slope, and changes thereof. Different strategies can then be chosen depending on expected temperature development.

The road slope can be estimated by calculating the drive resistance from the law of motion. This signal is then differentiated to detect changes of the slope and hence to be able to detect crests. Figure 3.1 represents an implementation of the probability generator top level. A more detailed description of the implementation can be found in Appendix A on Page 45.

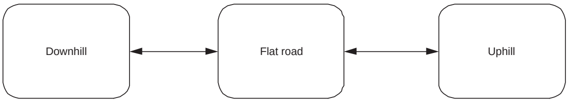

Figure 3.1. Top level of the stateflow implementation. The three states represent *downhill slopes*, *flat roads* and *uphill slopes*. Only one state is active at a given time, and the arrows show in what order a change of state is possible.

This information, together with vehicle speed, engine- and retarder torque and coolant temperature derivative is used to determine a probability to engage the fan. There are three major probability levels for fan engagement: *NoFan*, *LowFan* and *HighFan*. A *fan engagement temperature*, T_{engage}, is also determined. Below this temperature, the probability to engage the fan is set to zero. The span between the engagement temperature and a critical coolant temperature is divided into segments with different probabilities for fan engagement, and together with the major probability level a temperature map for fan engagement is generated.

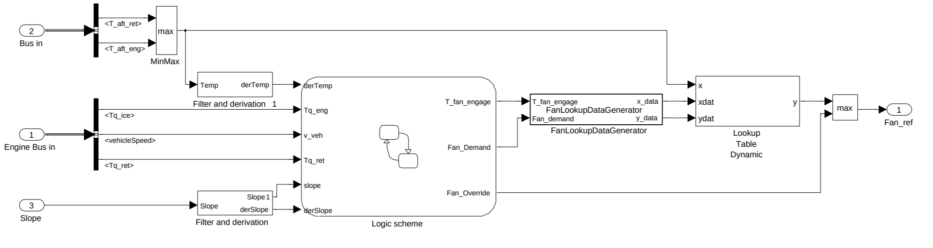

Figure 3.2. Top level of the Look Down controller implementation

Figure 3.2 shows the top level of the Look Down fan controller. It generates the reference fan speed by generating the mentioned probability map and compares it to the coolant temperature.

The basic idea is to lower the probability for fan engagement during drive in uphill slopes. It is assumed that energy losses are significantly smaller if the fan is used during drive in downhill slopes instead. Then kinetic energy can be used to drive the fan. There is however an important saving clause. The only situation where it is *costless* to run the fan, i.e. no energy is lost, is when the downhill is so steep and long that the vehicle maximum velocity is reached and it must be slowed down. Then there is a surplus of kinetic energy that can be used to power the fan. In these situations, the fan can act as a small retarder, even though the effect thereof is quite insignificant compared to the hydraulic retarder brake torque.

Opposed to uphill slopes, the probability to engage the fan is increased when driving downhill. This is a result of the reasoning above; it is *cheaper* to run the fan in a descent. The probability for fan engagement is further increased when the speed rises above a set speed and continues to increase. It is then likely that the retarder will be engaged to keep the speed below the maximum allowed velocity.

Since the retarder generates a lot of heat when it is used, an increased cooling need is expected and the fan is therefore preventively engaged. This is done using an override function, provided that the coolant temperature is above a threshold. If the temperature is below this threshold, the thermostat will not direct any of the coolant through the radiator and it would be unnecessary to run the fan. The reason to why the override function is implemented is to get a faster response when the temperature increases.

The fan speed override function is however also vehicle speed dependent. As long as the coolant temperature is above its threshold, the fan will be engaged with a medium speed when the vehicle speed rises over its threshold. Íf the vehicle speed continues to increase and the retarder is engaged, a higher override fan speed will be demanded. As the coolant temperature rises as a result of the retarder usage, fan speed may, if necessary, be further increased through the fan speed maps.

3.3 Look Ahead fan control

To further decrease fan usage during inefficient conditions and get a more robust controller, information about the road ahead can be used. There is always a risk to wrongly guess a coming crest when only looking at current conditions and changes thereof. If on the other hand information about the road ahead is provided through a map, combined with a GPS receiver, the controller will not make those mistakes.

The Look Ahead fan controller consists of three major subsystems: a Hill state generator, a fan engagement probability generator and a fan speed map generator. These subsystems work together to create a fan engagement probability and a fan engagement temperature which is finally compared to engine- and retarder coolant temperature to generate a fan speed demand.

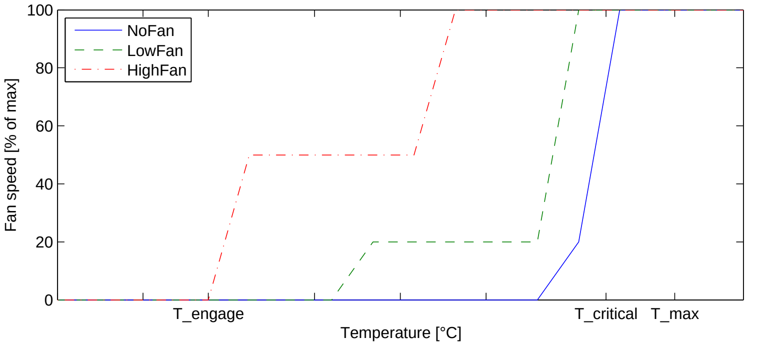

Figure 3.3. Fan speed map for the Look Down and Look Ahead fan controllers. The three curves correspond to the three fan engagement probability levels, and the values are given as percentage of maximum fan speed. The horisontal axle is dynamic and the values depend on T_{engage}. $T_{critical}$ is however constant.

The HILL state generator analyses the topographic information and classifies the road segments as one of four different states: *flat road*, *uphill*, *downhill* or *crest*. It receives data from a *horizon provider* containing information about road slopes 2000 m ahead of the vehicle, divided into segments of a previously determined length. The segments are first classified as *flat road*, *uphill* or *downhill*. The segment is an *uphill* or a *downhill* if the inclination is above or below certain thresholds respectively, otherwise a *flat road*. If an *uphill* is, within a certain distance, followed by a *downhill*, or vice versa, the segments in between are classified as *crest*. This is done continiously and the data is stored in a vector, in which every element represent a road segment. Output from the module is road classification for the current position, next classification (that differs from current classification) and the distance to the later. Figure 3.4 presents an example of how a short road is classified.

The fan engagement probability generator takes information from the HILL state generator and current engine and vehicle information, and generates a probability for fan engagement. As for the Look Down fan controller, probability for fan engagement is lower during drive in ascents, and minimized when closing a crest or a downhill. It is on the other hand increased when the vehicle is in a downhill slope, especially if the vehicle speed is high. The difference compared to the Look Down controller is that a crest or downhill can be detected much earlier and without risk of *guessing* wrong.

There is also an override function, which can engage the fan even if the coolant temperature is below the fan engagement temperature. This is done to preventively start the fan when the retarder is expected to be engaged. The function has two override fan speeds with corresponding vehicle speeds and a coolant temper-

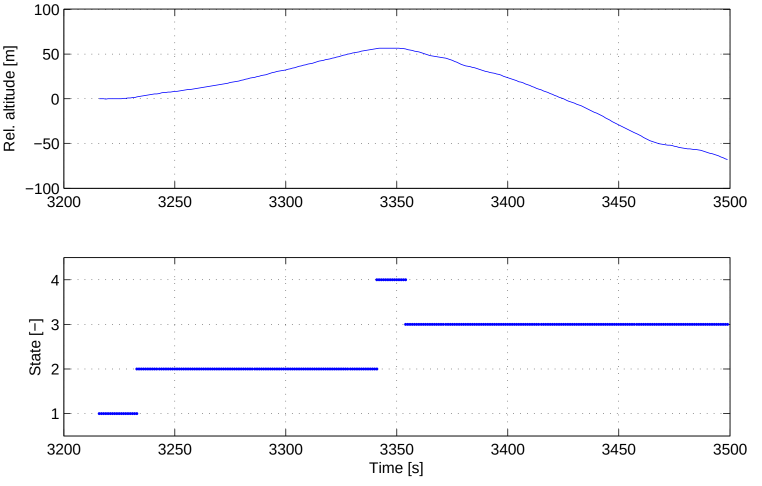

Figure 3.4. Example of how road segments are divided into one of four different states: 1 = *flat road*, 2 = *uphill*, 3 = *downhill* and 4 = *crest*

ature threshold. If the coolant temperature is above its threshold and the vehicle speed rises above its lower threshold, the fan is engaged and run at a medium speed. If the vehicle speed rises above its second threshold, the retarder will probably soon be engaged and the cooling need will be increased. It will also be *costless* to run the fan, hence is the fan speed set to a higher override speed. When the retarder finally is engaged and the coolant temperature rises, the controller will further increase the fan speed through its fan speed maps if necessary.

The fan engagement probability generator is implemented as a Stateflow diagram in SIMULINK. The top level states are represented by the four different road classifications and can be seen in Figure 3.5. A more detailed description can be found in Appendix A on Page 46.

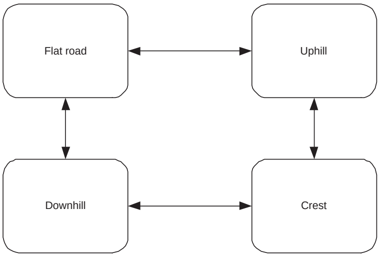

Figure 3.5. Top level of the stateflow implementation. The four states represent *downhill slopes*, *flat roads*, *uphill slopes* and *crests*. Only one state is active at a given time, and the arrows show in what direction a change of states is possible.

The fan speed map generator takes information about fan engagement probability and fan engagement temperature, and creates a fan speed map. The map takes coolant temperature as input and has requested fan speed as output. These maps can easily be adjusted through a calibration file. The output is compared to the override speed, and the largest value is forwarded to the fan speed controller.

Figure 3.6 shows the top level of the Look Ahead fan controller, consisting of the mentioned subsystems. As will be discussed in the following section, simulations show that the fan runs significantly less and more seldom when the Look Ahead fan controller is used, compared to a system that takes no regard to ambient conditions.

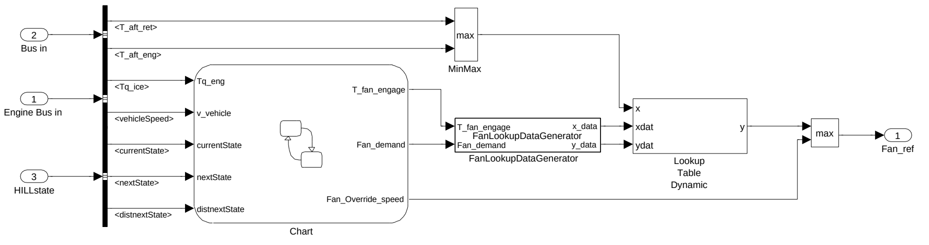

Figure 3.6. Top level of the Look Ahead fan controller implementation

3.4 Evaluation and comparison

3.4.1 Energy losses due to fan drive

Simulations show that significant energy savings can be obtained by the suggested improved fan controllers. The largest step is taken when using information about current conditions, as the Look Down fan controller does. The energy consumption can then be cut by approximately two thirds. Slope information is however insecure, since it is calculated from vehicle resistance and changes thereof. When utilizing digital map topography information, energy consumption can be cut by three quarters compared to a conventional controller. This system is more robust than a Look Down controller and can thus take advantage of having smaller margins.

There are two things that has to be said about Table 3.1. First, the fan is not engaged during any uphill slopes by any controller on the Södertälje-Norrköping route. The reason there although exist an energy consumption is the fan idle speed, caused by friction in the hydraulic clutch.

Second, it is assumed that energy is only consumed from the system when engine power is used to propel the vehicle. This is however not an absolute truth, since

Route		Conventional	Look Down	Look Ahead
Södertälje-Jönköping	(20 °C):	0.41	0.41	0.41
Södertälje-Jönköping	(30 °C):	0.41	0.41	0.41
Koblenz-Trier	(20 °C):	0.84	0.42	0.39
Koblenz-Trier	(30 °C):	1.56	0.59	0.39

Table 3.1. The table shows how much energy [MJ] the fan consumes using different fan controllers. Data is collected during drives at different locations and different conditions, but always during 1 hour.

kinetic energy is used to power the fan in downhill slopes. That is however compensated for through the law of motion, see Equations (2.1). The fan works thus as a retarder when driven in downhill slopes. This could mean that the vehicle would travel a shorter distance on the corresponding time, compared to a conventional controller. If so, that would have to be weighted against the saved amount of fuel. This is however not the case, since conventional fan controllers use engine power to drive the fan in uphill slopes, thus *stealing* engine power before it is converted to kinetic energy, and thus reducing vehicle speed. Simulations show that the vehicle actually travels a shorter distance with a conventional controller, compared to the controllers presented in this study. That effect is however negligible.

3.4.2 Logic behaviour and Comfort

There are situations where a conventional fan controller engages the fan before an imminent crest. It may not seem logic to a driver that the fan is driven when the heat generation is soon significantly reduced and a natural temperature decrease is expected. The Look Down controller handles this by estimating changes of the slope. A problem with the Look Down fan controller is however the risk to falsely detect a crest or a downhill. Fan engagement probability is then reduced during a short while, and this can cause an unnecessary rise of the coolant temperature. It can also cause illogical fan behaviour if the fan is momentarily slowed down or stopped while the temperature is still rising and the uphill slope has not ended.

This is on the other hand not a problem when using the Look Ahead fan controller. The driver will not be disturbed by fan engagement prior to crests or downhill slopes, nor will he experience the risk of a somewhat random behaviour of the Look Down fan controller during uphill slopes with varying inclination.

Both the Look Ahead and Look Down fan controllers take advantage of the fact that it in a sense is more efficient to run the fan during drive in downhill slopes. This causes an increased usage of the fan when the driver may think that there is no cooling need. This is especially true when the retarder is not used.

In downhill slopes where the retarder is actually used though, the early engagement of the fan leads to a better retarder efficiency. The retarder brake force is namely reduced when the coolant temperature gets too high. If the fan is engaged

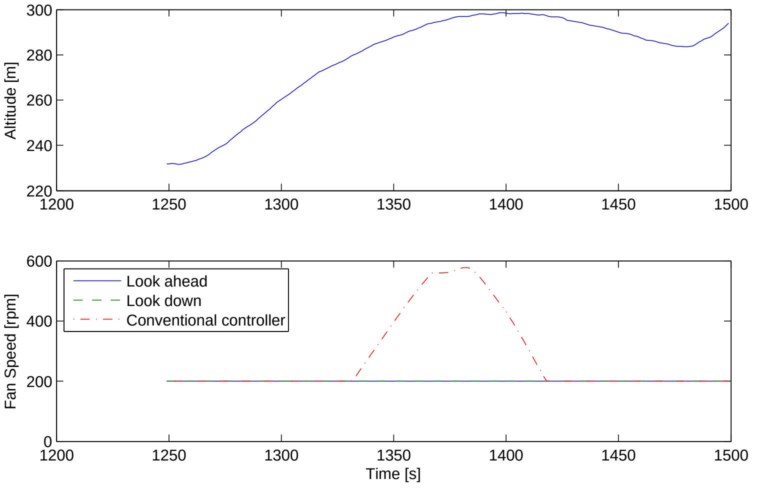

Figure 3.7. The figure shows an exampe of how a conventional fan controller engages the fan before a crest. Both the improved controllers have detected the crest and prohibit fan engagement. Note that it is only the conventional controller that engages the fan.

early, this can be prevented and the retarder can be used a longer distance.

There is also an aspect of comfort. When the fan is run above a certain speed, it generates a disturbing sound. This causes discomfort to the driver and should be avoided. Conventional controllers do not take this in regard, but it is easy to manipulate the fan speed maps in the Look Down and Look Ahead controllers to prevent this effect.

3.4.3 Temperature variations

The natural drawback when the fan usage is decreased is of course higher coolant temperatures. The mean temperature during the Koblenz-Trier route is rised by approximately 3 °C at an ambient temperature of 30 °C when using the Look Ahead controller compared to the conventional controller. What is more important is that the amplitudes of the temperature cycles are increased. This can lead to a decreased pressure[3] in the cooling system and if the vehicle is run at high altitudes with lower ambient pressure than normal, the pressure and consequently the boiling point of the coolant is lowered. This is however only a problem when the vehicle is driven during extreme conditions, such as very high ambient temperatures, high loads and high altitudes during a long time.

Fast temperature changes are also especially important since an inhomogeneous warm-up causes tensions in the materials. This is mainly a problem in the radi-

[3] The pressure is rised when the temperature is increased. If it gets too high, a dump valve will release pressure to protect the system. This leads to a generally lower coolant pressure.

ator where the coolant temperature difference is great. An increased number of temperature cycles is, for the same reason, also bad.

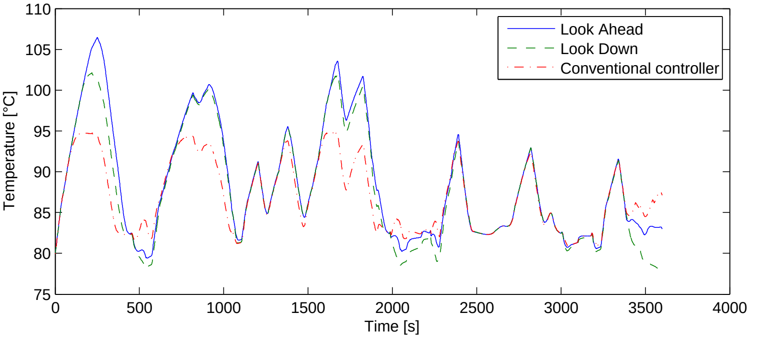

Figure 3.8. Engine coolant temperatures during a simulated drive at the Koblenz-Trier route, with an ambient temperature of 30 °C

As seen in Figure 3.8 the expected higher temperature tops are a fact when using the new controllers. The gradients of the temperature changes are also larger, especially at temperature reductions. There is however not an increased number of temperature cycles.

As the fan also is driven to prevent expected temperature rises caused by retarder usage in downhill slopes, the temperature of the coolant entering the engine can be lower than normal during these conditions. This has effects on the lubricating effect of the engine oil, which is deteriorated at lower temperatures. The coolant flow through the radiator is however decreased by the thermostat if the temperature is low, reducing this effect.

3.4.4 Key values

A rather straightforward way to compare the controllers is to use a number of key values representing interesting aspects of the control performance. They are presented in Table 3.2.

Key value		Conventional	Look Down	Look Ahead
Mean temperature	[°C]:	87.8	88.1	88.4
Min/max temperature	[°C/°C]:	80.0/102.1	76.4/101.4	77.6/103.9
Temperature variance	[-]:	23.5	33.9	36.3
Fan engagements (in uphill slopes)	[-/s]:	6/670	2/125	0/0
High speed fan run (above comfort level)	[s]:	10	75	55
Energy consumption	[MJ]:	1.14	0.48	0.39

Table 3.2. The table presents a number of key values, used to compare the controllers. The values are collected during a drive at the Koblenz-Trier route with an ambient temperature of 25 °C. The variance is a measure of statistical dispersion.

The mean temperature is slightly rised when using the improved controllers. There is namely a non-linear relation between mean coolant temperature and energy consumption caused by the fan. This effect gets stronger at higher ambient temperatures. One has to be careful when comparing the mean coolant temperature between different controllers though, since they can behave very different. It can be seen in Table 3.3 that even if the mean temperature is rised using a conventional controller, the energy consumption is significantly higher compared to the Look Ahead controller. Compared to the original conventional controller, the energy consumption is reduced though.

Key value		Original	Adjusted	Look Ahead
Mean temperature	[°C]:	87.8	88.4	88.4
Min/max temperature	[°C/°C]:	80.0/102.1	80.0/102.0	77.6/103.9
Energy consumption	[MJ]:	1.14	0.81	0.39

Table 3.3. The table presents a comparison between two conventional (original and adjusted) controllers with different temperature maps, and the Look Ahead controller. The values are collected during a drive at the Koblenz-Trier route with an ambient temperature of 25 °C.

As seen in Figure 3.9, a conventional fan controller has a less varying coolant temperature. This can be seen at the temperature variance which is clearly higher for the new controllers. The minimum and maximum coolant temperatures does not differ very much though, especially not the maximum ditto. This is however a result of retarder usage, seen at $T \approx 3500\ s$ in Figure 3.9. The improved controllers are better on suppressing these temperature rises since they engage the fan quite early when retarder usage is expected.

The number of fan engagements and fan run time in uphill slopes reflects how often and how long the fan is running during inefficient conditions. It is quite obvious that the improved controllers, especially the Look Ahead controller, does this much better than a conventional fan controller.

As discussed in Section 3.4.2, when the fan is driven above a certain speed (called *comfort level*), it generates a disturbing sound. This causes discomfort to the driver and should be avoided if possible. As seen in Table 3.2, the fan is run above this comfort level more often with the improved controllers. This is however a result of the fast fan speed response in downhill slopes where the retarder is engaged. Note that it is not the override function that causes these high fan speeds, but it is a result of increased coolant temperature. The disturbing sound must be weighted against the extended time the retarder can be used before its effect is restricted due to high temperatures.

The reason the Look Ahead controller causes any energy consumption at all (note the third paragraph of Section 3.4.1) is the fan idle speed. If it would be possible to fully disengage the fan from the engine, the Look Ahead controller would not generate any energy losses at all during this cycle.

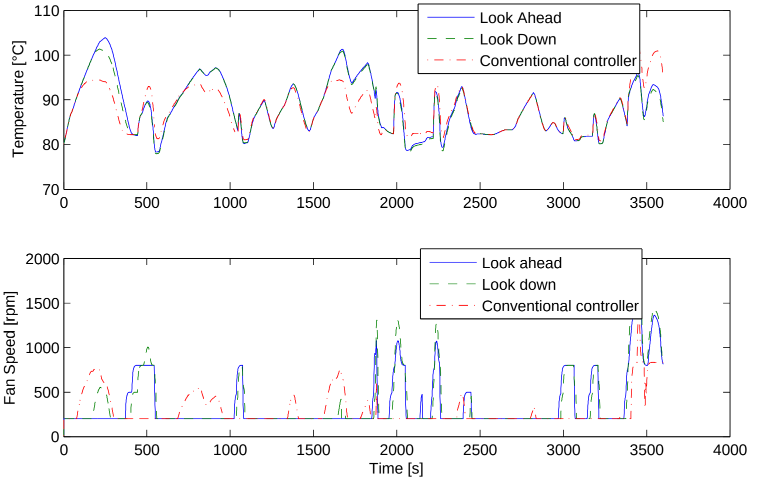

Figure 3.9. Coolant temperature after the retarder, and fan usage for both the conventional controller and the improved controllers. Data is collected from a simulation at the Koblenz-Trier route. See Table 3.2 for the key values associated to this simulation.

3.4.5 Adjusted conventional controller

One way to reduce energy consumption caused by fan usage is simply to reconfigure an existing controller. If for example the temperature maps are changed so the equivalent to the *fan engagement temperature* is rised, the fan will more seldomly be used. The natural consequence is of course a higher mean temperature, compared to an original conventional controller. Table 3.4 shows a comparison between two conventional controllers with different temperature maps and the Look Ahead controller.

Key value		Original	Adjusted	Look Ahead
Mean temperature	[°C]:	87.8	89.1	88.4
Min/max temperature	[°C/°C]:	80.0/102.1	80.0/103.4	77.6/103.9
Temperature variance	[-]:	23.5	36.6	36.3
Fan engagements (in uphill slopes)	[-/s]:	6/670	3/159	0/0
High speed fan run (above comfort level)	[s]:	10	10	55
Energy consumption	[MJ]:	1.14	0.48	0.39

Table 3.4. The table presents a number of key values, used to compare the controllers. The values are collected during a drive at the Koblenz-Trier route with an ambient temperature of 25 °C. The engagement temperature of the adjusted fan speed map is rised approximately ten degrees compared to the original controller.

Both the mean temperature and the maximum temperature are rised by approximately one degree each for the adjusted controller compared to the original. The minimum temperature is unchanged but it should be noted that the temperature variance is even higher than for the Look Ahead controller. This implies an increased material wear, as discussed in Section 3.4.3. The gain, on the other hand, is a significantly decreased energy consumption for the adjusted controller compared to its original. The results from the simulations are shown in Figure 3.10.

3.4.6 Optimal fan control

To further evaluate the presented controllers, they are compared to the result of optimized fan usage calculated with dynamic programming. Data such as heat flows, vehicle speed and engine speed are recorded during a drive where the fan is disengaged. These data are then used as inputs to a cost function in the dynamic programming, described in Section 2.4. The one state cooling system model is used both in the dynamic programming and with the Look Ahead controller.

To be able to get a fair comparison between the Look Ahead fan controller and the optimal solution, they are bound to begin at the same temperature. The optimal solution is then only constrained by an upper and lower temperature limit, a maximum fan speed and a maximum final temperature. It is obvious that the

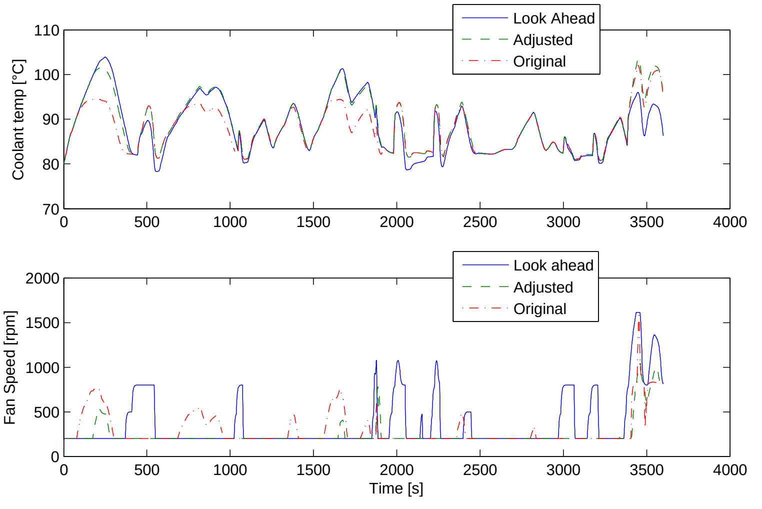

Figure 3.10. Coolant temperature after the retarder and fan usage for two variants of the conventional controller, one with an adjusted temperature map, and for the Look Ahead fan controller. Data is collected from a simulation at the Koblenz-Trier route. See Table 3.4 for the key values associated to this simulation.

Look Ahead fan controller behaves similar to the optimal solution. There are a number of differences though.

First, the optimal solution implies that a *bang-bang*-control[4] (the control signal takes either its maximum or minimum value) is most energy efficient, while the Look Ahead controller controls the fan speed more smoothly. Second, the Look Ahead controller has a lower fan speed compared to the optimal solution at most occasions. This is however a matter of calibration, and can easily be adjusted in the Look Ahead calibration file. Finally, the optimal solution suggests no fan engagement at $T \approx 2000\ s$ (see Figure 3.11) when the retarder is engaged. This causes no difference in energy consumption though.

Neither the optimal solution, nor the Look Ahead controller, engages the fan during drive in any uphill slopes at the simulated route. This is however a consequence of the absence of a thermostat in the one state cooling system model, causing a significantly lower mean coolant temperature. No conclusions about how the controller behaves during drive in uphill slopes can thus be drawn, besides the assuring fact that it does not engage the fan during this particular drive.

Both the optimal solution and the Look Ahead controller engage the fan during retarder braking. The Look Ahead controller does this through its override function, independent of coolant temperature. This is obviously efficient, since the optimal

[4]It is similar to a *bang-bang*-control, but according to [1], the system equation has to be linear against the control signal in order to acheive *bang-bang*, and that is not the case

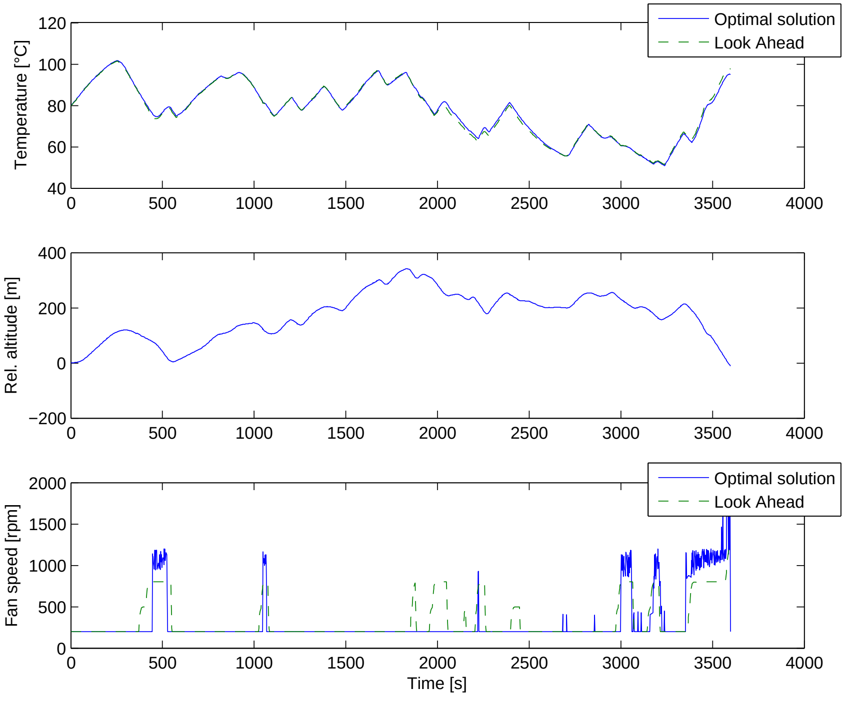

Figure 3.11. The optimal solution is presented and compared to the Look Ahead ditto during a one hour drive at the Koblenz-Trier route, with an ambient temperature of 30 °C. The noise is an effect of chosen calculation technique.

solution suggests the same behaviour. This is on the other hand not necessarily true in a real truck, since the thermostat will prevent the temperature to drop too low by directing the coolant flow through the bypass pipe. There is then no point to drive the fan, and the Look Ahead controller has a minumum temperature criteria that has to be satisfied to use the override function. This critera is however disengaged in this certain application, since lower temperatures must be considered as *normal*.

The noisy signal is an effect of chosen calculation setup. Euler's method of discretizising signals tends to cause this behaviour. This would be prevented if a more advanced discretizising method would be used. The reason that the fan speed does not exceed approximately 1200 *rpm* is that the cost function regard that speed as the previously mentioned *comfort speed*, thus penalizing speeds over that limit. It can however be necessary to exceed that speed if the cooling need is great, which is the case at $T \approx 3500$ *s* (see Figure 3.11).

There are also a couple of things that has to be mentioned about the result. There is no fan speed dynamic taken in regard, which means that instantaneous changes of fan speed can be suggested. Nor is the torque that is used by the fan compensated for[5]. The one state cooling model (see Equation 2.3.4), used in the calculations, contains some rather significant simplifications, most important the lack of a thermostat regulating the coolant flow through the radiator. This causes significantly lower minimum temperatures compared to the true system and the presented solution can thus not be directly applied. Conclusions can be drawn from the fundamental behaviour though.

Despite the mentioned problems, it must be pointed that the Look Ahead fan controller behaves very similar to the optimal solution. If a more detailed cooling system model would be used to calculate the optimal solution, the accordance would probably be even better since the Look Ahead fan controller is calibrated against the model presented in Chapter 2.

[5] The fan consumes either kinetic energy or engine power when it is engaged. This is not compensated for in the dynamic programming.

Chapter 4

Summary, Conclusions and Further Work

4.1 Summary

This thesis is a study of how cooling fan controlling can be improved by using road topography information. Two such controllers are presented, one that uses information available in vehicles produced today, and one that combines GPS-information with digital topographical maps to use information about the road ahead of the vehicle.

This study contains the following:

- A summary of other studies, related to this thesis is presented
- Two cooling system models are presented. One of them is more detailed and has a good accordance to real truck behaviour. The other model is a highly simplified one state model. They are both validated with good and fairly good accuracy respectively.
- The energy consumption caused by a conventional fan controller is estimated
- Two new fan control strategies are presented. One uses information about current inclination and changes thereof, and one uses information about topography ahead of the truck. These strategies are evaluated against each other and against conventional controllers.
- A study on optimal fan controlling is conducted, using the one state cooling system model mentioned above. The results are used to further evaluate the Look Ahead fan controller.

Simulations show that significant energy savings can be obtained, especially during warm conditions and hilly roads. Compared to conventional fan controllers, energy consumption can be cut by three quarters. Moreover, this is possible without any hardware redesign.

4.2 Conclusions

Table 4.1 (same as Table 3.2 on Page 34) shows some especially interesting key values, used to compare the different controllers. It is obvious that the improved controllers are more energy efficient compared to conventinal controllers. There are however some drawbacks, such as increased temperature tops and larger temperature variance.

Key value		Conventional	Look Down	Look Ahead
Mean temperature	[°C]:	87.8	88.1	88.4
Min/max temperature	[°C/°C]:	80.0/102.1	76.4/101.4	77.6/103.9
Temperature variance	[-]:	23.5	33.9	36.3
Fan engagements (in uphill slopes)	[-/s]:	6/670	2/125	0/0
High speed fan run (above comfort level)	[s]:	10	75	55
Energy consumption	[MJ]:	1.14	0.48	0.39

Table 4.1. The table presents a number of key values, used to compare the controllers. The values are collected during a drive at the Koblenz-Trier route with an ambient temperature of 25 °C.

The increased amplitude of the temperature cycles means a higher material wear. This is caused by the more restrictive fan usage during drive in uphill slopes when the engine emits large amounts of heat energy. The effects of these higher temperatures have not been part of this study but have to be taken in consideration when evaluating the results. It is hence hard to make any conclusions about the overall performance.

If effects such as hardware wear are neglected though, it is obvious that the improved controllers are better than conventional controllers. They use significantly less energy and have a more logical behaviour. They allow the retarder to be used for a longer time, thus saving the ordinary brakes, by preventively engaging the fan. There is one drawback attached this gain though, namely the increased time above fan comfort speed level. The conclusion must however be that it is more important to prolong the possible retarder usage, both for safety and economic reasons.

The easiest way to reduce energy consumtion caused by the fan is to allow higher coolant temperatures, i.e. raise the mean coolant temperature. This can be acheived by simply adjust the fan speed map in a conventional fan controller. One has to be aware that the margins are reduced though, thus obtaining a less robust controller. When using the Look Ahead controller, the temperature is for certain raised, but what is ahead of the truck is on the other hand known and can be compensated for.

When compared to an optimal solution, generated with dynamic programming and an one state cooling system model, the Look Ahead controller appears to behave quite similiar. One has to be aware of the simplifications mentioned in Section 3.4.6 but the result indicates that the Look Ahead controller has a near optimal behaviour.

To summarize the advantages and disadvantages with the presented controllers, they are presented as a list:

- \- Higher hardware wear, caused by an increased amplitude of the temperature cycles
- \- Fan is more often driven above the *comfort speed level*
- \+ Prolonged possible retarder usage
- \+ More logical fan behaviour
- \+ No hardware redesign is necessary to implement the controllers
- \+ The energy (and thus fuel-) consumption caused by the fan is reduced

All together, it must be concluded that the presented controllers in most aspects are better than conventional dittos. If only the fan clutch could be redesigned, allowing the fan be completely decoupled, energy consumption caused by the fan could be reduced to practically nil.

4.3 Further work

There are work left to do, mainly to get more robust controllers, but also to further cut unnecessary fuel consumption caused by bad fan controlling.

4.3.1 Robustness

As the ambient temperature is increased, the cooling performance in the radiator is decreased. This causes more rapid temperature rises, and there is a need of greater margins. To prevent overheating caused by the more reductive fan usage, ambient temperature should be a parameter in the fan speed map generator. It should also be studied how low ambient temperatures affect the cooling need in the system. It could for example be unnecessary to use the override function when driving in downhill slopes if the ambient temperature is below the freezing point.

In the present implementation of the improved controllers, the different distances that affect fan engagement probability is based on absolute distances. This could cause problems in ascents with great inclination when the vehicle can not hold its cruise speed, or if the driver deliberately drives slow. It should thus be investigated if a time based distance, derived from vehicle speed, is more suitable.

Downhill slopes should be divided into two different classifications: one where the retarder is expected to be engaged and one without retarder usage expectancy. This could then be used to determine whether the override function should be activated or not.

There are several units that can demand fan engagement, but only the engine and retarder have been taken in regard in this thesis project. The study should be extended to include the other demanders as well.

4.3.2 Hardware

No hardware modifications has been considered in this thesis project. It is however a fact that the fan clutch slip causes almost all of the energy losses when the new controllers are used. A study concerning this is presented [7]. It concludes that significant savings can be obtained if the cooling fan is electrically driven.

A controllable waterpump would also increase the degrees of freedom. It should be investigated how these mentioned modifications could be used together with the presented controllers.

It should also be investigated if the increased hardware wear, caused by larger temperature variations, creates a need to use other materials, or other hardware layouts. If the flow through the radiator can be made more homogeneous for example, material tensions can be decreased.

4.3.3 Other auxiliaries

The study should be extended to include other auxiliaries as well. The air compressor is probably the easiest auxiliary to proceed with. It does not demand any hardware reconfigurations and a similar implementation could be used. It is already investigated in [7] what gains can be obtained by using electrically driven auxiliaries. A study should be conducted in which these conclusions are combined with the results in this thesis project and translated to actual controllers.

Appendix A

Road state flowcharts

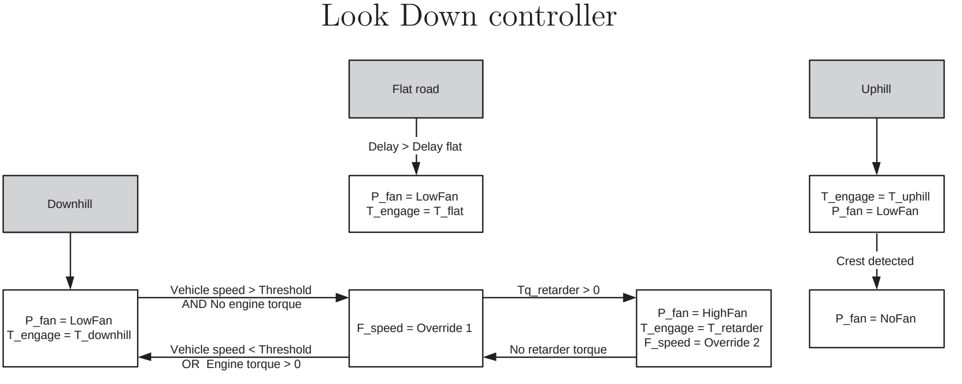

Figure A.1. A more detailed description of the Look Down stateflow design. The grey boxes correnspond to one of three top level states, also seen in Figure 3.1. *T_engage* is the *fan engagement temperature* mentioned in Sections 3.2 & 3.3. *T_uphill*, *T_downhill*, *T_flat* and *T_retarder* are temperature parameters, predefined in the calibration file. *P_fan* corresponds to the probability for fan engagement, also mentioned in Sections 3.2 & 3.3. *NoFan*, *LowFan* and *HighFan* represent the three main probability levels for fan engagement. *F_speed* is connected to the override function, *Override 1* and *Override 2* are consequently the two different override fan speeds, predefined in the calibration file. *Threshold* is the vehicle speed associated to the override function, also predefined in the calibration file. *Delay* is a time delay and *Delay flat* its threshold. Finally, *Tq_retarder* is retarder torque, which is non zero when the retarder is engaged.

Look Ahead controller

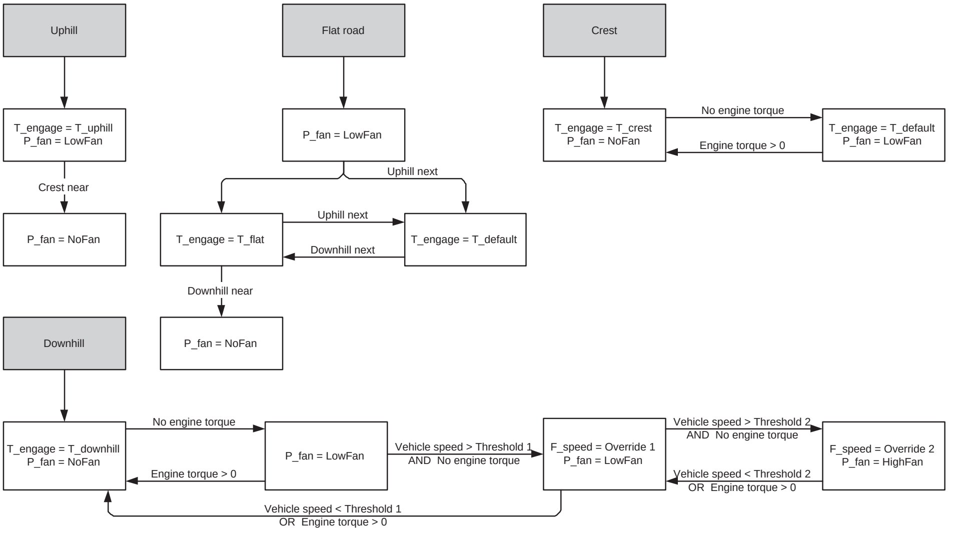

Figure A.2. A more detailed description of the stateflow design. The grey boxes correspond to one of the four top level states and thus the four different road classifications. *T_engage* is the *fan engagement temperature* mentioned in Sections 3.2 & 3.3. *T_uphill*, *T_downhill*, *T_crest*, *T_flat*, and *T_default* are temperature parameters, predefined in the calibration file. *P_fan* corresponds to the probability for fan engagement, also mentioned in Sections 3.2 & 3.3. *NoFan*, *LowFan* and *HighFan* represent the three main probability levels for fan engagement. *F_speed* is connected to the override function, *Override 1* and *Override 2* are consequently the two different override fan speeds, predefined in the calibration file. *Threshold 1* and *Threshold 2* are the two vehicle speeds associated to the override function, also predefined in the calibration file.

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