International Journal of Control
Vol. 79, No. 5, May 2006, 479–495

Taylor & Francis
Taylor & Francis Group

Modelling and control of auxiliary loads in heavy vehicles

N. PETTERSSON† and K. H. JOHANSSON*‡

†Scania CV AB, Södertälje, Sweden
‡Department of Signals, Sensors and Systems, Royal Institute of Technology, Stockholm, Sweden

(*Received 23 May 2005; in final form 21 November 2005*)

Improved control of the auxiliary units is a possible method to save fuel in heavy vehicles. Today, the auxiliaries are often mechanically driven by the engine, and are constrained to rotate with a fixed ratio to the engine speed. This mechanical constraint results in energy losses. In the paper, the benefits of driving the auxiliaries with electricity are evaluated. The output of an electrically driven auxiliary can at every time instant be controlled to match the actual need. Considered auxiliaries are electrical generator, water pump, cooling fan, air compressor, air conditioning compressor, oil pump and power steering pump. Upper limits on what fuel saving that can be achieved if the auxiliaries are redesigned are first discussed. Simulations indicate that the fuel consumption caused by the auxiliary units is in the range of 5–7% of the total consumption. A Modelica library used for simulation of the energy consumption of the auxiliary units is presented. The cooling system of the library is described in some detail. A case study on optimal control of the cooling system is then performed. Control actuators are the electrical generator, the cooling fan and the water pump. The fan and the pump are supposed to be electrically driven. The control design is based on a simplified model derived from physical principles. It is evaluated through simulations with external variables collected from experiments. The results show that significant energy savings can be obtained.

1. Introduction

The fuel economy is one of the most important properties of heavy vehicles. This is especially true for long haulage trucks. In Western Europe, and regions with a similar economical situation, the cost of fuels represents approximately one third of the total cost for the owner of long haulage trucks. The truck owner is running a business where the success is depending on low cost, and is well aware of this fact. When times come to replace the old truck, brands with good fuel economy are competitive. The truck manufacturing companies, of course, respond to this demand and put efforts into lowering the fuel consumption wherever it is possible when developing new models and components. Naturally, a good fuel economy must have its origin in a well tuned combustion process in the intrinsic engine. Substantial progress has been achieved over the years to increase the efficiency of the diesel engine. Simultaneously, all other losses in the vehicle have to be reduced. Examples of potential improvements are reduction of aerodynamic drag, rolling resistance and, the topic of this paper, better utilization of the energy consumed by auxiliary units such as pumps and fans driven by the engine.

1.1 *Motivation*

The importance of auxiliary units with high efficiency is often stressed and a fair number of proposals on various novel designs are found in the literature, but very little of this can be seen in the vehicles on the market today. Efficiency of the auxiliary units was pointed out as one of the prioritized areas in the plan for reducing fuel consumption and emission for trucks

*Corresponding author. Email: kallej@s3.kth.se

International Journal of Control
ISSN 0020–7179 print/ISSN 1366–5820 online © 2006 Taylor & Francis
http://www.tandf.co.uk/journals
DOI: 10.1080//00207170600587333

in the US Technology Roadmap for the 21st Century Truck Program (US Department of Energy 2000). Today auxiliaries are in general mechanically driven and effective means to control their output to the actual need are often lacking. Up to now, the energy savings possible with more advanced solutions could seldom make up for the increasing cost and complexity. However, as the price of fuel tends to increase, alternatives become more competitive. The application considered in the paper is heavy vehicles used for long haulage transport. Long-haulage vehicles are more sensitive to fuel cost than other types of heavy vehicles. Thus, efforts spent in developing more efficient systems will be most attractive for this type of vehicles. Even if the development may lead to more complex and expensive solutions, it might be worthwhile if the fuel saving is big enough. Furthermore, many of the truck manufacturers have the bulk of their production volume in the long-haulage category. The performance of the product line of the manufacturer is often optimized for the long-haulage trucks. That design is then reused with minor modification for other types of heavy vehicles like, for instance, construction trucks. Thus, efforts spent on improving the performance of the long-haulage trucks will also affect other categories of heavy vehicles.

Motivation for the presented evaluation of the potential benefits of driving the auxiliaries with electricity also comes from the requirements for dependability of heavy vehicles. The resistance to new components sets particular demands on how to employ new technologies for lowering the fuel consumption. If a significant change of an existing design is done, possibly years of development and thorough tests have to be performed before introducing the novelty on the market. The feasibility study in this paper relies on theoretical reasoning and simulations, where the objective is to prepare for a decision on whether more practical and, thereby, more costly development activities should be started. As much knowledge as possible of real-world operating conditions is put into the study.

1.2 *Contribution*

The main contribution of the paper is to analyse the potential benefits of driving the auxiliary units electrically. The energy consumption of auxiliary units of a heavy vehicle is presented. The studied auxiliary units are electrical generator, cooling fan, water pump, air compressor, air condition compressor, oil pump and power steering pump. With electrical drives the output of the auxiliaries can be continuously adjusted to the desired level. It is shown that the energy utilization can be improved, compared to a mechanical design with less control capability. However, the use of a common source of power also introduces additional interactions between the subsystems. In order to utilize the possibilities offered with the electrical drive and to handle the interactions in a sound way, the problem is analysed using optimal control. It is argued that the control performance can be significantly improved by the use of model-based estimation, onboard GPS receiver and digitized maps.

An important factor that makes it interesting to consider electrification of previously mechanically driven devices and redesigns of the electrical system in heavy vehicles is the recent development of hybrid electrical passenger cars. The hybrid technology can be an enabler for passenger cars, city buses and light-weight trucks with less environmental impact and increased performance. However, for heavy trucks and especially in long-haulage transport, the technology seems to be less suitable. Compared with other vehicles, the engine in long-haulage trucks typically operates in a narrower speed range and at a power output closer to the maximum capacity. Therefore the average efficiency of the engine over the drive cycle comes closer to the maximum than in other vehicle applications, and thus, the potential improvement with hybridization is lower. Nevertheless, the technology development can be of use even here, since power electronic components with specifications and pricing adjusted for automotive use are likely to be available on the market in a few years. Components intended for propulsion of passenger cars may with smaller modification very well be suitable for other tasks in heavy vehicles. The same reasoning is valid also for the design methodology of the hybrid vehicles, for instance, many of the considerations about control principles for hybrid vehicles are applicable in the control of electrical power in heavy vehicles.

1.3 *Related work*

Modelling and control of auxiliary units in heavy vehicles is a quite unexplored area. Related research, as described below, has been presented on some specific applications and for particular control problems. The study presented in this paper is focused on Scania vehicles and European conditions.

An overview of the energy consumption of auxiliary units for heavy vehicles in North America is given in Hendricks and O'Keefe (2002) and models of auxiliary loads in O'Keefe *et al*. (2002). Some prototypes of electrically driven auxiliaries intended for heavy duty vehicles are discussed in Pettersson (2002), Hnatczuk *et al*. (2000) and Lasecki and Cousineau (2003). Every automotive company has a need for model libraries to be used in various tasks like assessing impacts on fuel consumption. Some efforts at Scania are described in § 3 and in Sanberg (2001) and Bengtsson (2004). An example of a Modelica library developed at Ford Motor Company is presented in Tiller *et al*. (2003). General-vehicle libraries to be used in studies of different aspects of hybrid electrical vehicles have also been developed, e.g., Laine and Andreasson (2003). Energy-efficient control of auxiliary units is discussed in § 4. Similar problems, but for hybrid electrical cars, were considered by Daimler Chrysler and the University of Karlsruhe (Back *et al*. 2004). Prediction based on an onboard GPS receiver and digitized maps was discussed in Finkeldei and Back (2004).

1.4 *Outline*

The paper starts in § 2 with a survey over the auxiliary units as they are designed today and how they influence the overall fuel economy. The consumption of the individual subsystems is derived through simulations of simple models. The results illustrate the relative significance of the auxiliaries. In § 3, detailed models for simulation of the energy consumption of the auxiliary units are presented. Particularly the development of a Modelica library and modelling of the cooling system are discussed. Section 4 presents a case study on energy optimal control of the cooling system. Optimal control theory is employed to derive the control of an electrically driven water pump and cooling fan, and the control of the generator. Finally, in § 5 the results are summarized. The paper is based on the thesis (Pettersson 2004). Various versions of the results have been presented at conferences (Pettersson and Johansson 2003a, b, 2004).

2. Energy consumption of auxiliary units

This section aims at giving an overview of the auxiliary units and their influence on the overall fuel consumption for long haulage vehicles. The ambition is to give an estimate of the relative significance of the individual auxiliaries. The results may work as a guideline on where to put effort on new designs in order to decrease the fuel consumption and give estimates of what the potential of such efforts is.

The considered auxiliaries and their maximum and minimum power consumption in a Scania truck at 1400 rpm engine speed are shown in figure 1. The power consumptions are indicated for the full range of operation conditions (at the fixed engine speed). Note that for the water pump and the oil pump, the power consumption is constant at a constant engine speed. For the other auxiliaries, the consumption depends on several factors.

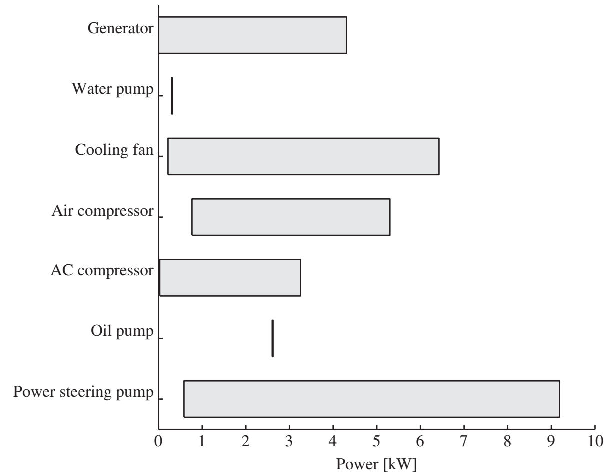

Figure 1. Summary of the auxiliary units with their minimum and maximum power consumption when operated at an engine speed equal to 1400 rpm.

In this section, we estimate through computer simulations what the influence of the auxiliaries are on the fuel consumption during a typical drive cycle. A suitable measure of the fuel consumption of the auxiliary units is defined and numerically derived through simulation of models of the auxiliary units. The models are not presented here due to space constraints. They are specified in detail in Pettersson (2004) and related to the models discussed in § 3.

2.1 *Drive cycle*

The considered drive cycles correspond to two roads that are representative for long-haulage traffic in Europe. Consequently they primarily consist of highway driving. The reference vehicle speed is mostly restricted by the set point of the electronic speed limiter, which is compulsory for the considered category of heavy vehicles within the European Union. The technical speed limit of the electronic device is defined to not exceed 89 km/h. It is assumed that the vehicle hold this speed on road sections where the legal limit for passenger cars is above the technical limit 89 km/h, even though the maximum allowed speed for articulated heavy vehicles is 80 km/h. In some sections of the roads, the legal limit is lower or it is impossible to hold the maximum allowed speed with the vehicle combination due to for instance the inclination or the curvature of the road. When going downhill and the retarder is used to control the speed, the reference speed is set to 6 km/h higher than the active limit when the speed is controlled with the cruise control acting on the engine. The speed limitations and the road inclinations used in the simulation are obtained from a data base of recorded road profiles available at Scania.

The first road section goes between Koblenz and Trier in Germany. The trip starts in Koblenz and returning to the starting point when Trier reached. Figure 2 shows road altitude (upper plot), vehicle speed (lower thick) and the required drive power (lower thin) for this drive cycle. The second road section runs between Södertälje and Gothenburg in Sweden. It starts in Södertälje passes Jönköping and ends in Gothenburg. Figure 3 shows the corresponding variables for this drive cycle. The two routes

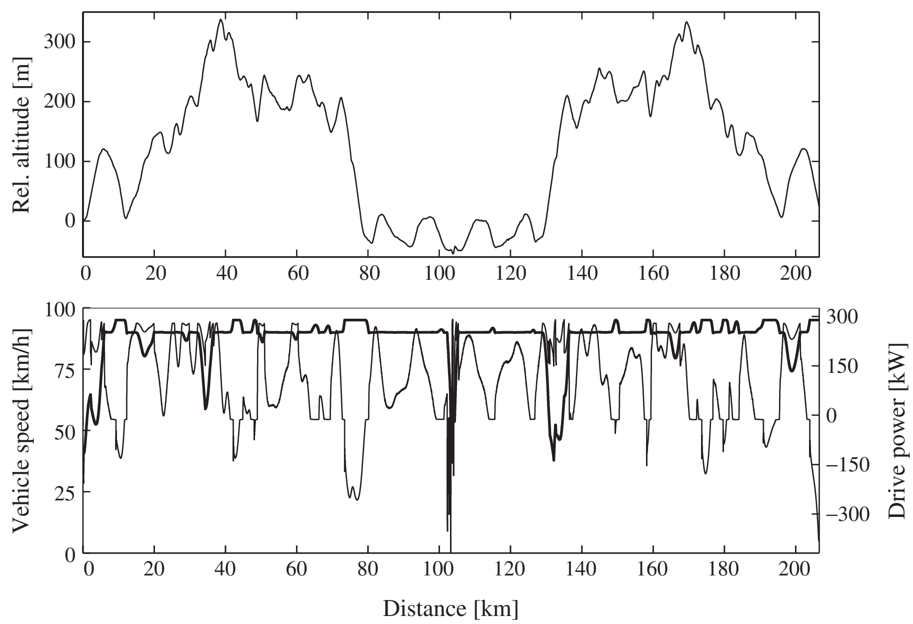

Figure 2. Road altitude (upper), and vehicle speed (lower, thick) and drive power (lower, thin) for the Koblenz–Trier route.

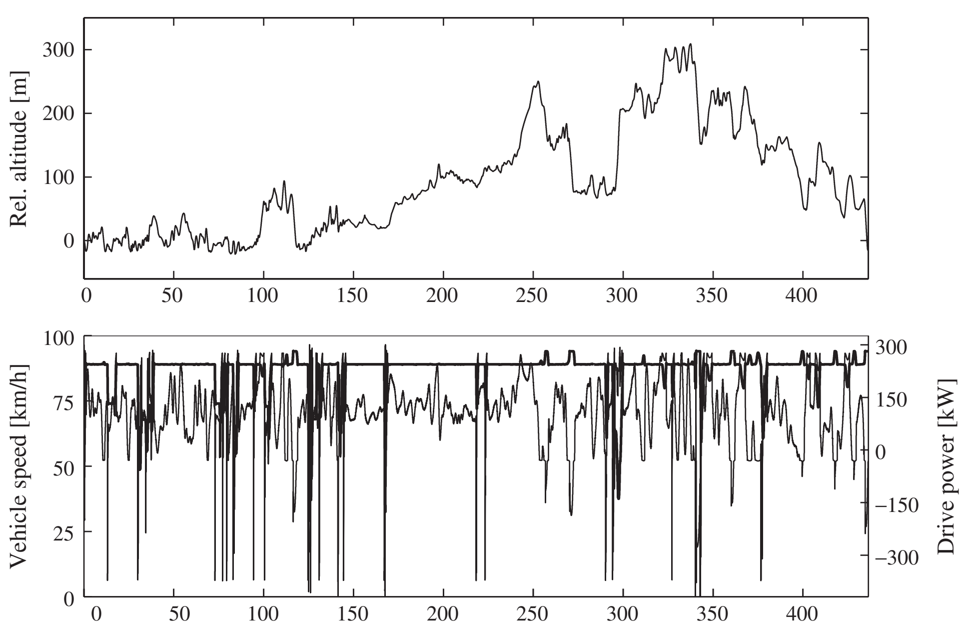

Figure 3. Road altitude (upper), and vehicle speed (lower, thick) and drive power (lower, thin) for the Södertälje–Gothenburg route.

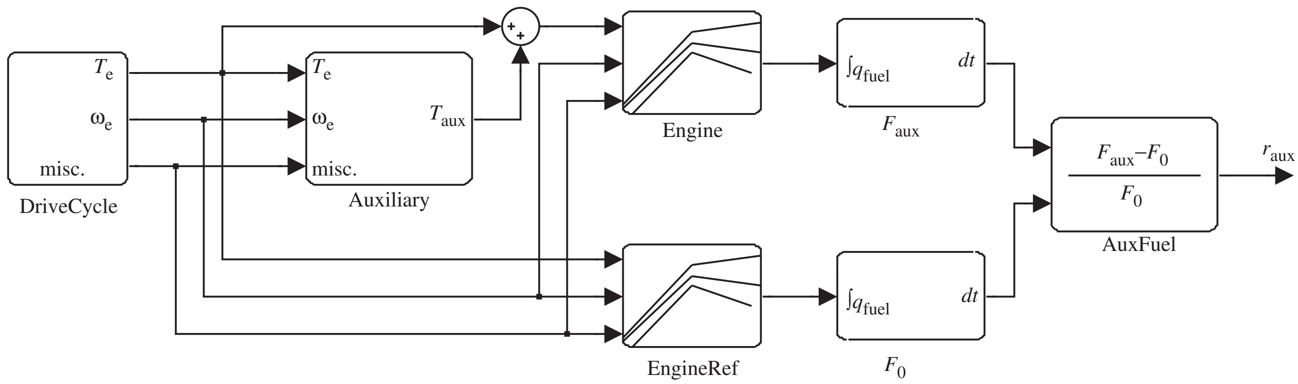

Figure 4. Block diagram illustrating the computation of the the fuel consumption of the auxiliary units.

differ from each other in the variation of road inclination. The first route contains some rather hilly sections mixed with more flat sections, while the second route is more even in the distribution of road inclinations. As a result, the total fuel consumption is higher on the German road than on the Swedish one.

2.2 *Computation of energy consumption*

To derive the energy consumption of an auxiliary, we introduce the power taken from the engine to drive that auxiliary at time t as $P_{aux}(t)$. The mean power consumption over a drive cycle with duration T is then equal to

$$\bar{P}_{aux} \triangleq \frac{1}{T}\int_0^T P_{aux}(t)\mathrm{d}t$$

This measure cannot be directly translated to how much the auxiliaries influence the overall fuel consumption. The correlation in time between the auxiliary and the engine load will influence how much the drive of an auxiliary unit costs in terms of fuel consumption. Let F_{aux} and F_0 denote the amount of fuel spent when driving with and without the auxiliary, respectively, and d the total distance travelled over a drive cycle. Then the mean fuel consumption for the auxiliary per travelled distance is equal to

$$\bar{f}_{aux} \triangleq \frac{F_{aux} - F_0}{d}.$$

We also consider a relative measure for the auxiliary part of the fuel consumption

$$r_{aux} \triangleq \frac{F_{aux} - F_0}{F_0}.$$

The computation for each auxiliary unit of the three measures $\bar{P}_{aux}$, $\bar{f}_{aux}$, r_{aux} introduced above is illustrated in the block diagram in figure 4. Models of the auxiliary units were developed in Simulink and are described in Pettersson (2004). The block labelled `Auxiliary` in figure 4 represents one instance of an auxiliary model. Inputs to that block are time trajectories describing the operation of the vehicle, which have been obtained from simulations of a complete vehicle model in advance. The output of the auxiliary block is an additional torque on the driveline, denoted T_{aux}, which is needed to run the auxiliaries. The sum of the torque for the auxiliary unit and the original engine torque is then fed to the inverse model of the engine, labelled `Engine`. Here the amount of fuel flow needed to create the total torque is calculated. The nominal engine torque is fed to a reference engine labelled `EngineRef`, to calculate the amount of fuel that would be consumed without the auxiliary load. The outputs of the engine models are finally used to compute $\bar{f}_{aux}$ and r_{aux}.

The technique applied here to compute the input that would yield a certain predefined state trajectory is sometimes referred to as an inverse, or backward, simulation in contrast to traditional, forward, simulations, where the state trajectory is the output. Here, forward and backward simulations are combined. Trajectories of the engine speed and torque are used to obtain the required input fuel flow in a backward manner while the internal states in the auxiliary model are simulated in a forward manner, cf., Wipke *et al.* (1999). The assumption that the auxiliary units do not influence the overall vehicle trajectories is a simplification. If the auxiliary load is large enough, it might actually cause a change in the gear shifting strategy. Such effects are assumed to be absent, since the auxiliary power is small in comparison to the total driveline power.

2.3 *Results*

The energy consumption of the auxiliary units is computed for the two drive cycles. The total part of the fuel consumption that can be derived from the considered auxiliary units is in the range 4.7% to 7.3%. The relative fuel consumption r_{aux} of the auxiliaries for the Koblenz–Trier route is between 0.40% and 1.40%. For the Södertälje–Gothenburg route it is between 0.45% and 1.7%. The reason that the auxiliaries represent a smaller part of the total fuel consumption in the German drive cycle than in the Swedish cycle is that the average load on the engine is higher in the German cycle due to the larger road inclinations. Therefore the auxiliary power represents a smaller fraction of the total driveline power. The relation between the studied auxiliaries does not change very much between the driving cases. The relation between the auxiliaries gives an insight into their relative influence on the overall fuel economy, see Pettersson (2004) for details. The total consumptions for the auxiliaries are summarized in table 1.

The measures are presented for the two routes and for an older and a more recent set of auxiliaries.

Table 1. Total consumptions for auxiliaries.

Drive cycle	Aux.	$\overline{P}_{aux}$ [kW]	$\bar{f}_{aux}$ [litre/100 km]	r_{aux} [%]
Koblenz–Trier	New	8.79	1.59	4.66
	Old	10.67	2.03	5.98
Södertälje–Gothenburg	New	8.36	1.79	5.62
	Old	10.51	2.32	7.31

3. Modelling of energy consumption

Vehicle models that can be used to evaluate energy control strategies of the auxiliary units are presented in this section. A model library developed in Modelica (Modelica Association 2002) is briefly described. The models are typically built from physical principles with parameters identified from various tests, resulting in so called grey-box models (Lennart Ljung 1999). As an example, we describe the modelling of the cooling system in some detail. It is shown how measurements from tests in a wind tunnel are used to tune the model. The model is then validated against data recorded from a dynamic drive cycle in the wind tunnel.

3.1 *Model library*

Figure 5 shows the composition of the vehicle model at the highest level of abstraction. The prime goal with the vehicle model is to serve as a tool for studying effects on the fuel economy from alternative designs of sub-systems. Flow of energy between various parts of the vehicle is the main considered physical quantity. To give an accurate estimate of the energy balance, the model covers the whole vehicle and describes processes involved in the energy conversion with a significant level of detail. Besides description of physical phenomena, it contains control software and various look-up tables.

The principal structure of the developed Modelica library at Scania is shown in figure 6. The library is organized after the parts of the truck, in contrast to the Modelica Standard Library (Modelica Association 2002), which is organized after engineering disciplines.

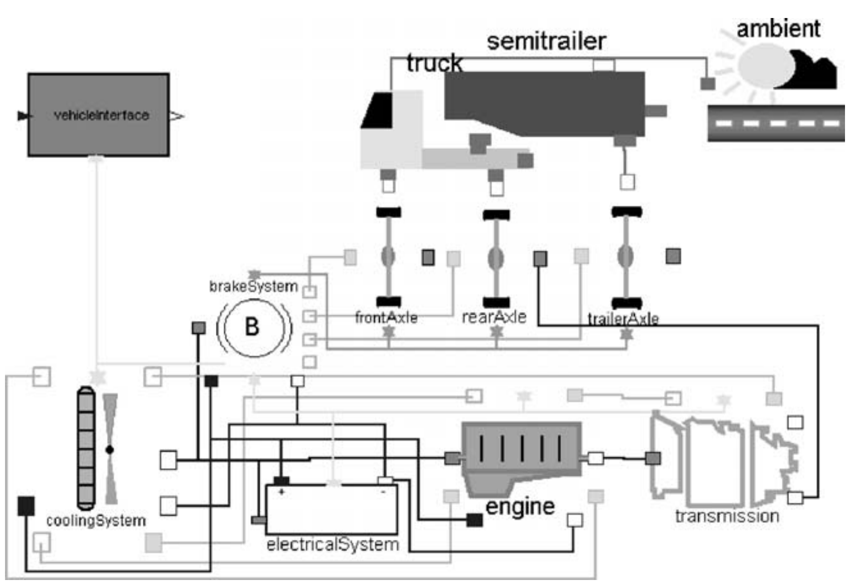

Figure 5. Modules of the vehicle model at the highest level of abstraction.

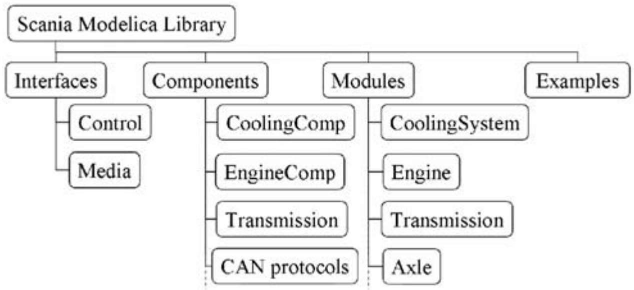

Figure 6. Structure of the Scania Modelica Library.

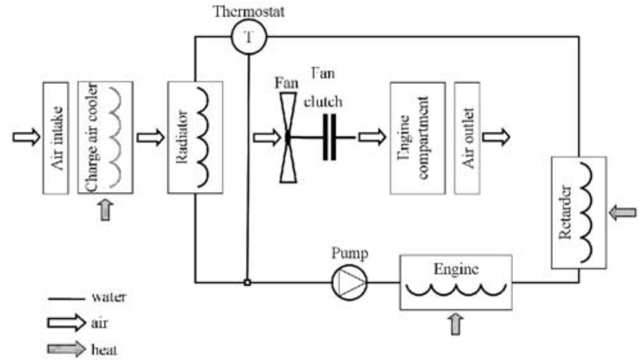

Figure 7. Components in the cooling system module. The model mainly consists of two adjoining flows of mass and energy: the flow of coolant fluid and the flow of air.

The Scania Modelica Library consists of four main branches: interfaces; components; modules and examples. The Interfaces branch contains classes describing connections between model components. One example is the CAN connector, used to mimic the information flow between control units in the truck. In the components branch, models of all physical parts needed to build up the complete model of a truck are gathered. The modules branch contains compound models, which can be put together for various types of simulations. Such working examples are found in the examples branch. As an illustration, we consider cooling systems components of the components branch. They build up a cooling system module.

3.2 *Cooling system module*

The cooling system is one of the modules of the Scania Modelica Library. Energy consumers in the cooling system are primarily the cooling fan and the water pump. In heavy vehicles, these units normally are mechanically driven. The model described next corresponds to the current design of a Scania truck where the water pump is driven directly from the crankshaft while the cooling fan is connected to the shaft via a viscous clutch enabling a passive speed control. The existing model structure allows for changing the model to describe other ways of driving and controlling these auxiliaries.

Figure 7 depicts the cooling system module. The model mainly consists of two adjoining flows of mass and energy: the flow of coolant fluid and the flow of air. The main part of the system is modelled using thermodynamic and hydraulic base classes, essentially following the principles in the ThermoFluid library of Tummescheit *et al*. (2000). The thermodynamic models are built up with alternating control volumes and flow models. Mass and energy balances are defined in the control volumes, while relations between the pressure drop and the flow are specified in the flow models. Series of control volumes and flow models are aggregated to form a composite model. A detailed description of the modelling of the cooling system is given in Pettersson (2004), including experiments and parameter estimations to obtain certain sub-models. See also Pettersson and Johansson (2003a,b). Some of the parameters in the sub-models represent basic quantities such as mass or volume, which are found in the technical specification of the components. Other parameters, typically describing the behaviour of the flow models, have to be estimated from data. For that reason, experimental data are collected from rig tests on individual components in a laboratory environment. For each component, an identification problem is solved trying to fit the predicted output from the models to the measurements. Table 2 summarizes which parameters that are identified and which data that are used. As a consequence of the well-controlled conditions for the laboratory experiments, the prediction errors are small for the identified models.

Some parameters of the sub-models are assigned as slack parameters, as indicated in the last column of table 2. They are adjusted to fit the behaviour of the total model to the measurements. The slack parameters are chosen based on engineering practice. Parameters that describe characteristics of other phenomena than what could be captured in a test of one component, or where no measurement data for the single component are available, are preferred as slack parameters. Another heuristic rule is to use only one slack parameter to achieve a certain correction when there are several possible parameters that can be adjusted to achieve the same effect.

3.3 *Validation results*

A validation of the total model is performed as a last step of the modelling and parameter identification.

Table 2. Components of the cooling system module in the Scania Modelica Library. Identified parameters and corresponding data sources are shown. Some parameters are used as slack variables in the model tuning.

Component	Characteristic	Data source	Slack
Pump	Pressure rise	Rig test	s
	Power consumption	Rig test	
Engine	Flow resistance	Rig test	
	Heat capacitance	Data sheet	s
	Heat emission to coolant	Rig test	
	Heat conductance	Rig test	
	Heat emission from charge air	Rig test	
Retarder	Flow resistance	Rig test	
	Heat capacitance	Data sheet	s
	Heat emission	None	
	Heat conductance	Rig test	
Thermostat	Opening characteristics	Rig test	
	Flow resistance	Rig test	
	Dynamic response	Rig test	
Radiator	Flow resistance coolant	Rig test	
	Flow resistance air	Rig test	
	Operating characteristics	Rig test	
	Heat capacitance	Data sheet	
Air intake	Pressure build-up	None	s
Charge air cooler	Flow resistance	Rig test	s
Fan	Pressure rise	Rig test	
	Power consumption	Rig test	
Fan clutch	Slip characteristics	Rig test	
Engine compartment	Flow resistance	Rig test	
Air outlet	Pressure build-up	None	s

Validation data are recorded during a dynamic drive cycle in a wind tunnel, where the load and speed of the dynamometer are programmed to follow a cycle corresponding to a specified road. In figure 8 simulation results are compared with measurements, when the dynamometer follows the profile of the road between Koblenz and Trier described in §2. The validation shows that the model is capable to capture the main dynamics of the cooling system quite well. It does not, however, describe the small oscillations observed in the measurements around 80°C. These oscillations most likely have their origin in the hysteresis of the thermostat, due to friction. The model of the thermostat is a rather rough approximation and do not give rise to the corresponding oscillations. Despite the observed differences, the model is sufficiently accurate to evaluate the energy consumption of the auxiliary units in the cooling system.

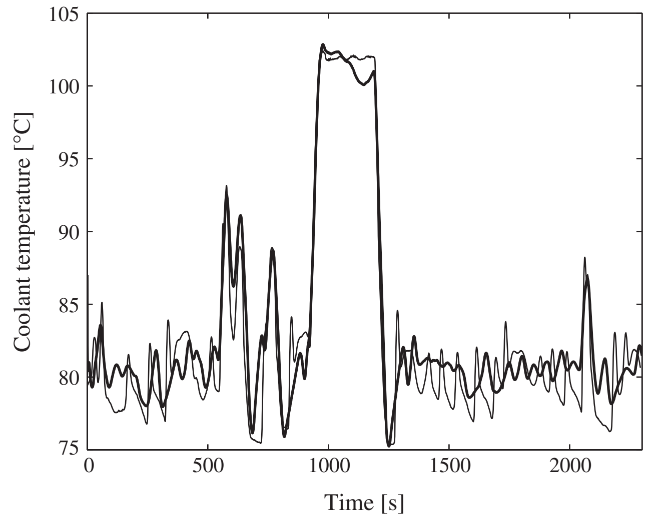

Figure 8. Simulated coolant temperature (thick) and measurements (thin) during a dynamic drive cycle between Koblenz and Trier. The model captures the main dynamics of the cooling system quite well.

3.4 *Discussion*

The modelling errors in the sub-models are very small. When the sub-models are assembled, phenomena that are not handled in the sub-models may play an important role. It may be effects from the interconnections between the components such as piping in the truck cab. Further, non-linearities may amplify small errors in the sub-models when these are connected and new feedback paths are closed. For example, it can be shown, using a simplified model of the cooling system, that the change of temperature of the coolant in steady state due to a small perturbation of the airflow is proportional to the squared inverse of the airflow. Thus, the simulated temperature will be very sensitive to modelling errors influencing the airflow. Further, for the pressure build-up due to the wind speed there exists no practicable experiment on a component level. Therefore, the result of the total model is verified through comparison with experimental data collected in a wind tunnel. In the wind tunnel, the vehicle is driven on a dynamometer with a defined load and speed of the engine and fans are used to simulate the wind speed.

The applied procedure of building a model consisting of sub-models with physical interpretation and performing sequential identification of parameters is illustrated in figure 9. Data collected from measurements on single components are used to estimate parameters of the sub-models. The experience from our study is that a good agreement between the simulated behaviour of the sub-models and the measurements do not guarantees accuracy of the total model.

Effects not included in the sub-models or small errors in the parameters of the sub-models may have a large influence on the behaviour of the compound model. Therefore the agreement between the actual system and the simulation output may be poor although the sub-models seem to be accurate. For that reason, it is necessary to compare output of the complete model with measurements. If a satisfactory result is not obtained, some of the parameters of the sub-models are assigned as slack variables and they are adjusted to tune the behaviour of the complete vehicle. When acceptable agreement is obtained, the complete model is validated against new measurements. The procedure allows for keeping the physical structure of the model, while the total behaviour of the model can be tuned to give a good fit to measurement data.

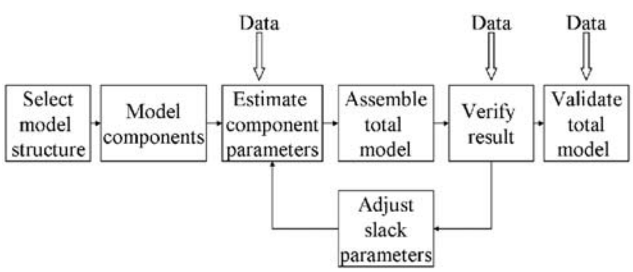

Figure 9. Applied procedure of sequential modelling and identification.

4. Energy optimal control of the cooling system

This section presents a study on energy optimal control of the cooling system. The configuration of the system is depicted in Figure 10. The mechanical drives of the water pump and the cooling fan are replaced with electrically driven systems, which enable a continuous control of the pump and the fan speeds, cf., figure 7. The electrical power required to run the drive systems and other electrical loads is produced with the generator, which is driven by the engine. An electrical buffer enables storage of energy. The controlled quantities are the temperature in the cooling system and the charge level of the electrical energy storage. Control inputs are the speed of the water pump and the cooling fan, and the electrical power of the generator. The objective is to minimize the fuel used to drive the input actuators. Control performance may significantly benefit from knowing about future dynamical behaviour of the system. Such information can be provided from model-based estimation together with the onboard GPS receiver and digitalized maps. In the context of this study, the heat emitted to the cooling system, the vehicle speed and the electrical loads are external influences that may be predicted. With a vehicle model, they can be numerically derived from the slope of the road and the predicted velocity.

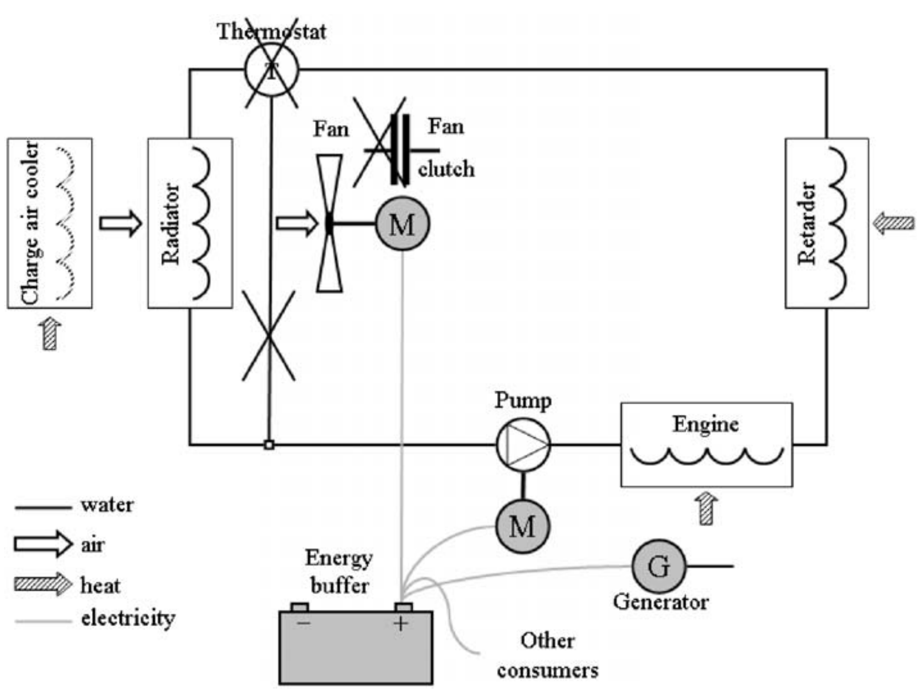

Figure 10. System configuration for the considered energy optimal control problem. Crossed-out components are the ones of the traditional cooling system that are excluded, cf., figure 7. Instead, novel components are added (grey) to enable improved overall energy control.

4.1 *Model*

A suitable system model is needed to derive the optimal controller. The Modelica model of the cooling system described in § 3 is too complex to be used for control design. However, it is a valuable starting point for finding a simplified model. The one presented next is of second order, with one state representing the temperature of the cooling system and the other the state of charge of the energy buffer. The control variables are the pump speed u_1, the fan speed u_2 and the power produced by the generator u_3. See Pettersson (2004) for detailed derivations.

Let the state x_1 represent the temperature in the cooling system minus the ambient temperature. Its dynamics is derived from the energy balance in the cooling system, that is, $\dot{x}_1$ is proportional to the net power flow into the system. Let v_1 represent the sum of the heat flows into the system. The power flow out of the cooling system is the heat transfer in the radiator, which can be factorized into the entry temperature difference denoted $\Delta\tau$ and the specific heat transfer ϕ_s. The model of the cooling temperature can thus be expressed as

$$\dot{x}_1 = -c_2 \Delta\tau \phi_s + c_1 v_1,$$

where c_1 and c_2 are scaling constants. (In the following, we use c_i to denote constants.) The entry temperature difference $\Delta\tau$ can be approximated as

$$\Delta\tau = x_1 - \frac{c_6 v_2}{u_2 + c_3 v_3},$$

where v_2 is the heat emission in the charge air cooler and v_3 the vehicle speed. The specific heat transfer ϕ_s can be approximated by a rational function. The resulting differential equation is then

$$\begin{aligned}\dot{x}_1 &= -c_2\left(x_1 - \frac{c_6 v_2}{u_2 + c_3 v_3}\right) \\ &\quad \frac{u_1(u_2 + c_3 v_3)}{c_4 u_1 + u_1(u_2 + c_3 v_3) + c_5(u_2 + c_3 v_3)} + c_1 v_1 \\ &\triangleq f_1(x_1, u_1, u_2, v_1, v_2, v_3),\end{aligned}$$

where u_1 and u_2 are control variables and v_1, v_2, v_3 are (uncontrollable) external variables. Figure 11 shows a validation of the model. The model state (thick solid line) is quite close to the corresponding state of the Modelica model (thin solid line) and the measurement recorded in wind tunnel (dotted line).

Let x_2 represent the state of charge above the lowest allowed level of the energy buffer. Introduce the notation ψ_n for the net power in the terminals of the electrical buffer. Then the model of the electrical system can be expressed as

$$\dot{x}_2 = \psi_n - c_7 \psi_n^2.$$

The net power ψ_n is the generated power minus power consumed by the pump, the fan and other electrical equipment. It is then possible to derive the following equation for the state of charge

$$\begin{aligned}\dot{x}_2 &= c_8 u_3 - c_9 v_4 - c_{10} u_1^3 - c_{11} u_2^3 \\ &\quad - c_7(c_8 u_3 - c_9 v_4 - c_{10} u_1^3 - c_{11} u_2^3)^2 \\ &\triangleq f_2(u_1, u_2, u_3, v_4),\end{aligned}$$

where v_4 represents electrical consumption of units other than the pump and the fan. To summarize, the model of the cooling system, including the electrical

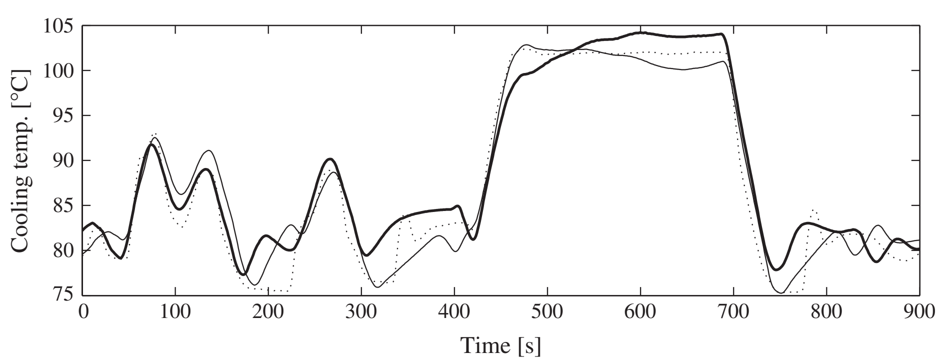

Figure 11. Coolant temperature obtained with the model used for control design (thick solid) compared with temperature obtained with the Modelica model (thin solid) and with measurements (dotted).

sub-system, is given by the second-order system

$$\left.\begin{aligned}\dot{x}_1 &= f_1(x_1, u_1, u_2, v_1, v_2, v_3)\\ \dot{x}_2 &= f_2(u_1, u_2, u_3, v_4).\end{aligned}\right\} \tag{1}$$

4.2 *Optimal control*

The control objective is to minimize fuel used to drive the generator, while keeping the temperature x_1 in the cooling system and the charge level x_2 in the energy buffer within specified limits. It is assumed that the time trajectories of v_1, v_2, v_3 and v_4 are known or can be predicted for some time ahead. In practice this prediction horizon is limited by the ability to accurately estimate the external variables from input data. Based on the model defined in the previous section, and knowledge of the future external variables, the optimal input trajectory is computed. The control objective of minimizing the fuel consumption is comparable to minimizing the power produced in the generator integrated over times when fuel is injected in the engine, based on the assumption that the both the combustion engine and the generator have a nearly constant efficiency. This is reasonable since the efficiency of the diesel engine in heavy vehicles varies with a few percentage points in the rpm range utilized in highway driving.

4.2.1 Optimal control problem. The considered prediction horizon spans from the present time t_i up to a final time $t_f = t_i + t_p$, where t_p is the prediction horizon. Let $\delta = \delta(t)$ be a binary weighting factor that equals one when fuel is needed to drive the vehicle forward, while it is zero when no fuel is injected in the engine. The objective can be formulated as

$$\min_{u_1, u_2, u_3 \in U} J(u) \triangleq \min_{u_1, u_2, u_3 \in U} \int_{t_i}^{t_f} \delta(t) u_3(t)\, \mathrm{d}t \tag{2}$$

subject to the dynamics (1), and the state constraints

$$0 \le x_{i_{\min}} \le x_i \le x_{i_{\max}} = 1, \quad i = 1, 2.$$

The set of admissible controls is equal to

$$U = \{u\colon 0 \le u_{i_{\min}} \le u_i \le 1,\ i = 1, 2, 3\}.$$

The optimization is performed in a receding horizon scheme where only an initial part of the calculated control input is applied. The length of the input trajectory that is applied in each control update is called the control horizon and spans from t_i to $t_i + t_c$, with $t_c < t_p$. When time $t_i + t_c$, is reached, the initial and final time is set to $t_i := t_i + t_c$, and $t_f := t_f + t_c$, and a new optimal control is derived. In order to force the control not to utilize the buffers in the end of the optimization interval, constraints on the final states are introduced

$$x_i(t_f) = \frac{x_{i_{\min}} + x_{i_{\max}}}{2}, \quad i = 1, 2. \tag{3}$$

4.2.2 Solution with inactive constraints. The Hamiltonian of the optimal control problem is equal to

$$H = \delta u_3 + \lambda_1 f_1(x_1, u_1, u_2, v) + \lambda_2 f_2(u, v),$$

where u and v denote the obvious vectors. To describe the general solution to the optimal control problem, we first consider the case when the state constraints are inactive. The adjoint variables should satisfy the differential equation

$$\begin{aligned}\dot{\lambda}_1 &= -\frac{\partial H}{\partial x_1} = -\lambda_1 \frac{\partial f_1}{\partial x_1}\\ \dot{\lambda}_2 &= -\frac{\partial H}{\partial x_2} = 0.\end{aligned}$$

Hence, λ_2 is constant. Note that the initial and final states are known, while the boundary conditions for the adjoint variables are not. The final value of the adjoint variable $\lambda(t_f)$ is arbitrary.

Let us first solve (2) with respect to u_3. For intervals when $\delta(t) = 0$, the optimal u_3, is dependent only on the sign of λ_2. It is straightforward to see that λ_2 should be negative. This gives the control

$$u_3 = \begin{cases} \mathrm{sat}^+(\mathrm{sgn}(-\lambda_2)), & \delta = 0\\ \mathrm{sat}^+\left(\dfrac{1 + c_8\lambda_2}{2c_7 c_8^2 \lambda_2} + \dfrac{c_9 v_4 + \psi_c(u_1, u_2)}{c_8}\right), & \delta = 1 \end{cases} \tag{4}$$

where $\mathrm{sat}^+(\alpha) = 1$, if $\alpha > 1$, 0 if $\alpha < 0$, and α otherwise. The function

$$\psi_c(u_1, u_2) = c_{10} u_1^3 + c_{11} u_2^3$$

denotes the power consumption of the pump and the fan. Note that when $\delta(t) = 1$, not only the sign of λ_2 is of importance. Here λ_2 should be chosen such that the constraints on x_2 are satisfied.

If u_3 does not saturate when $\delta = 1$, the Hamiltonian becomes

$$H = \begin{cases} \lambda_1 f_1(x_1, u_1, u_2, v) + \lambda_2[c_8 - c_9 v_4 - \psi_c(u_1, u_2) \\ \quad -c_7(c_8 - c_9 v_4 - \psi_c(u_1, u_2))^2], & \delta = 0 \\ \psi_c(u_1, u_2)/c_8 + \lambda_1 f_1(x_1, u_1, u_2, v) + h(\lambda_2, v_4), & \delta = 1, \end{cases} \quad (5)$$

where h is a known function. If $\delta(t) = 1$, $t_i \le t \le t_f$, the controls u_1 and u_2 that minimize H are obviously independent of u_3, λ_2 and x_2. If $\delta(t) = 0$, $t_i \le t \le t_f$, the minimizing controls u_1 and u_2 are independent of u_3 and x_2, but depends on the sign of λ_2 (which is known to be negative). This suggests that it is possible to separate the control of x_1 from the control of x_2: first u_1 and u_2 are derived from (5), then $\psi_c(u_1, u_2)$ is plugged into equation (4), giving u_3. However, if δ change in the prediction interval, λ_2 must be known when deriving the control of x_1. The optimal control problem for the cooling system corresponding to the Hamiltonian in (5) can thus be formulated as

$$\begin{aligned} \min_{u_1, u_2 \in U_c} J_c \triangleq \min_{u_1, u_2 \in U_c} \int_{t_i}^{t_f} & \{\delta(t)\psi_c(u_1, u_2) \\ & - (1-\delta)\lambda_2 c_8[\psi_c(u_1, u_2) \\ & + c_7(c_8 - c_9 v_4 - \psi_c(u_1, u_2))^2]\}\, \mathrm{d}t \end{aligned} \quad (6)$$

subject to the state equations and constraints. Due to the dependence of λ_2 in (6) and u_1, u_2 in (4), the control of x_1 and x_2 cannot be separated from each other. However, here the approach is to solve for the control of x_1 and x_2 in sequence, and iteratively update the parameters linking the controls together. The simplification this yields is considerable since only one-dimensional problems are solved at each step. The iteration scheme is as follows:

1. Set λ_2 to an initial guess (e.g., $-1/c_8$).
2. Solve the cooling optimisation problem in (6) with the current value of λ_2 to obtain u_1 and u_2.
3. Find a new λ_2 such that the constraints on x_2 are satisfied and apply equation (4) to obtain u_3.
4. Terminate if the current λ_2 is close enough to λ_2 used in 1 or if $\delta(t)u_3(t) = 0$, $t_i \le t \le t_f$, otherwise jump to 1.

Convergence properties of the iteration are discussed in Pettersson (2004).

4.2.3 Solution with active constraints. If the state constraints are active, the solution to the optimal control problem is somewhat more complicated, as discussed next. The adjoint variables can be discontinuous at time instances where the state constraints go from being active to non-active, or vice versa. Therefore the optimal trajectory x_1^* must be divided into constrained and unconstrained arcs. Consider the case when $\delta(t) = 1$, and assume that the optimal trajectory consists of the three parts illustrated in figure 12: an unconstrained arc, $x_1^*(t)$, $t_i \le t \le t_1$ ending on a constraint, say $x_{1_{\max}}$, a constrained arc, $x_1^*(t) = x_{1_{\max}}$, $t_1 \le t \le t_2$, ending at t_2 when the state leaves the constraint, and an unconstrained arc, $x_1^*(t)$, $t_2 \le t \le t_f$ ending at the final state value $x_1^*(t_f) = (x_{1_{\min}} + x_{1_{\max}})/2$. The optimal control that keeps $x_1^*(t) = x_{1_{\max}}$ is given by

$$\begin{aligned} \begin{bmatrix} u_1 \\ u_2 \end{bmatrix} & \\ & = \arg\min\{\psi_c(u_1, u_2) : u \in U_c, f_1(x_{1_{\max}}, u_1, u_2, v) = 0\} \\ & \triangleq g(x_1 \max, v). \end{aligned} \quad (7)$$

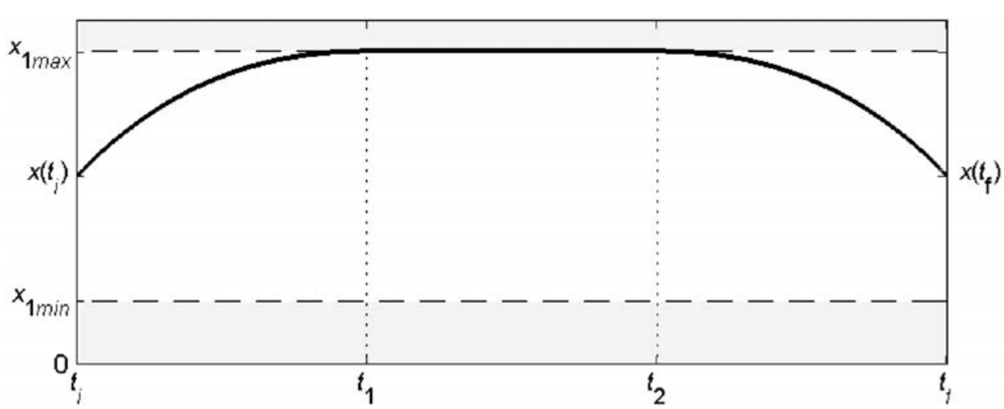

Figure 12. The optimal trajectory x_1^* is divided into constrained and unconstrained arcs.

In this case the cost function can be re-written as

$$\begin{aligned}
J_c &= \int_{t_i}^{t_f} \psi_c(u_1, u_2)\,\mathrm{d}t \\
&= \int_{t_i}^{t_1} \psi_c(u_1, u_2)\,\mathrm{d}t + \int_{t_1}^{t_2} \psi_c(g(x_{1_{\max}}), v)\,\mathrm{d}t \\
&\quad + \int_{t_2}^{t_f} \psi_c(u_1, u_2)\,\mathrm{d}t \\
&= \underbrace{\int_{t_i}^{t_1} \psi_c(u_1, u_2)\,\mathrm{d}t - \int_{t_i}^{t_f} \psi_c(g(x_{1_{\max}}), v)\,\mathrm{d}t}_{J_{c_1}} \\
&\quad + \underbrace{\int_{t_2}^{t_f} \psi_c(u_1, u_2)\,\mathrm{d}t - \int_{t_2}^{t_f} \psi_c(g(x_{1_{\max}}), v)\,\mathrm{d}t}_{J_{c_2}} \\
&\quad + \underbrace{\int_{t_i}^{t_f} \psi_c(g(x_{1_{\max}}, v))\,\mathrm{d}t}_{J_{c_0}}
\end{aligned} \tag{8}$$

Since J_{c_0} is independent of u, t_1 and t_2, the problem can be divided into two separate optimal control problems over unconstrained arcs. The times t_1 and t_2 are free parameters. Hence, the optimization can be performed over open time intervals. The condition on the final state (3) is now replaced by the condition that the corresponding Hamiltonian should be equal to zero when the state is entering or leaving the constraint. The discussion above can easily be extended to a general case when the state trajectory enters and leaves the constraint several times. The cost function then becomes

$$J_c = J_{c_1} + J_{c_2} + \sum_{k=3,5,7,\ldots,N} J_{c_k}(t_k, t_{k+1}) + J_{c_0}. \tag{9}$$

Consequently, the optimization can always be separated into independent problems over unconstrained arcs. In the numerical solution this is done iteratively, searching over the prediction horizon for trajectories that satisfy the condition that the Hamiltonian is equal to zero at the entry and exit to the constraints.

4.3 *Results*

The optimal control strategy is simulated with input data collected from wind tunnel experiments. The external variables v_1, v_2 and v_3 are measured on the truck when the load and speed of the dynamometer are programmed to follow trajectories corresponding to the specified roads. Simulations are performed over two road sections with altitudes depicted in figure 13. The first simulation runs over part of the road between Koblenz and Trier, discussed earlier. The second simulation runs over part of the road

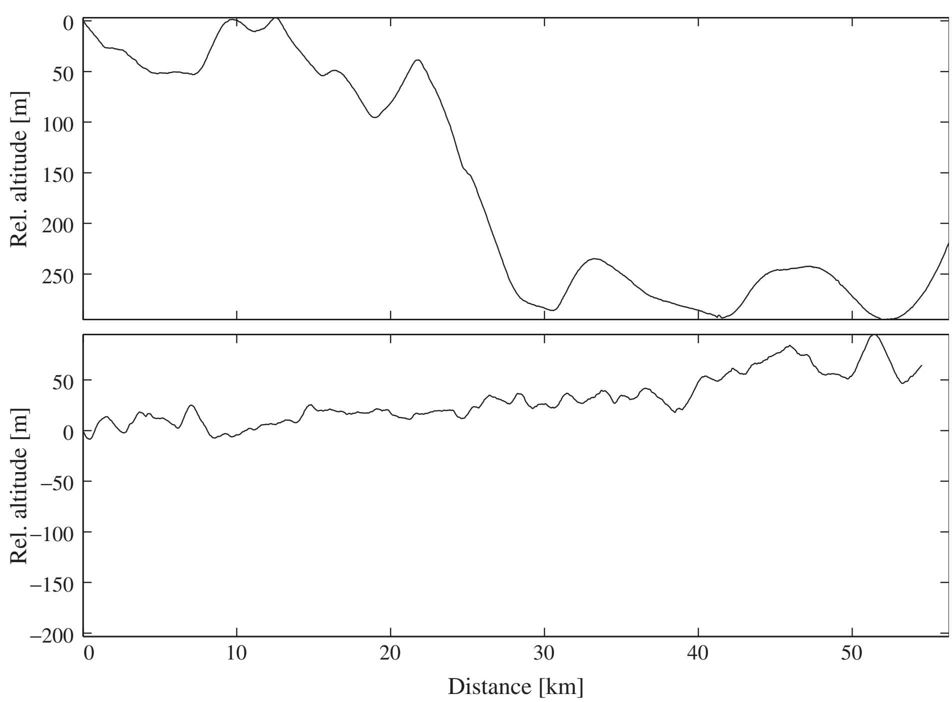

Figure 13. Road altitudes used to collect external data. The used part of the Koblenz–Trier road is shown in the upper plot and the used part of the Södertälje–Norrköping road in the lower.

between Södertälje and Norrköping in Sweden. The German route contains rather steep downhill slopes where a lot of heat is emitted to the cooling system from the retarder. The section of the road in Sweden contains long flat sections with some moderate uphill slopes where the engine produces heat that have to be cooled away. The prediction horizon is set to $t_p = 600\,\text{s}$ and the control horizon to $t_c = 100\,\text{s}$. It is assumed that the controller has exact knowledge of the external variables over the prediction horizon.

Simulation results are shown in figure 14 (Koblenz–Trier) and figure 15 (Södertälje–Norrköping). Simulations where optimal control is applied (thick lines) is compared with measurements on a traditional truck (thin lines). In the upper plot, the temperature obtained with optimal control (thick) and measured temperature (thin) is shown. The second plot shows optimal control of the pump (thick solid) and the fan (thick solid-stars) compared with the speed of the pump (thin solid) and the fan (thin solid-stars) in the traditional truck. The third plot shows simulated

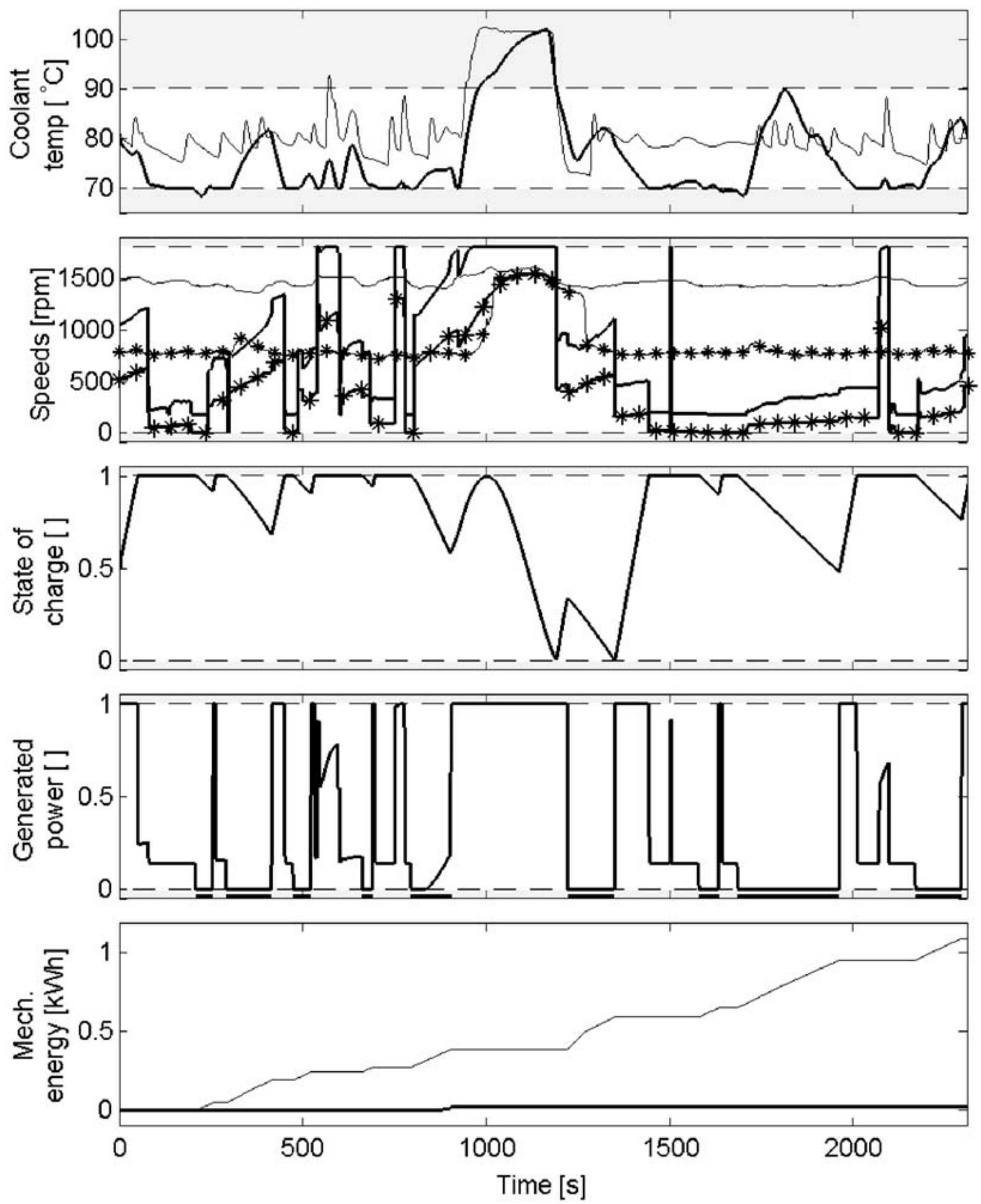

Figure 14. Simulation results for energy optimal control on the Koblenz–Trier road.

state of charge in the energy buffer. The fourth plot shows optimal control of the generator and indicates the intervals where $\delta(t) = 1$ as a bar on the time axis. The lowest plot shows the energy taken from the engine (when $\delta = 1$), with optimal control (thick) and with traditional control (thin). For the optimal control, the plotted energy consumption equals the cost function in (2) divided by the efficiency factor of the generator. The energy consumption in the traditional truck is calculated as the sum of the energy consumed in the pump, the fan including the clutch, and the generator. The generator is assumed to have the same efficiency as in the novel system where the optimal control is applied. Note that with the optimal control the energy saving is significant in both simulations.

The optimal control utilizes the admissible range of coolant temperature and state of charge. Therefore most of the electricity can be produced when $\delta(t) = 0$, that is, at times t when no fuel is injected in the engine and auxiliary loads can be added without any cost. This can be seen in the fourth plot of the figures, where the bar on the time axis indicates when $\delta(t) = 1$.

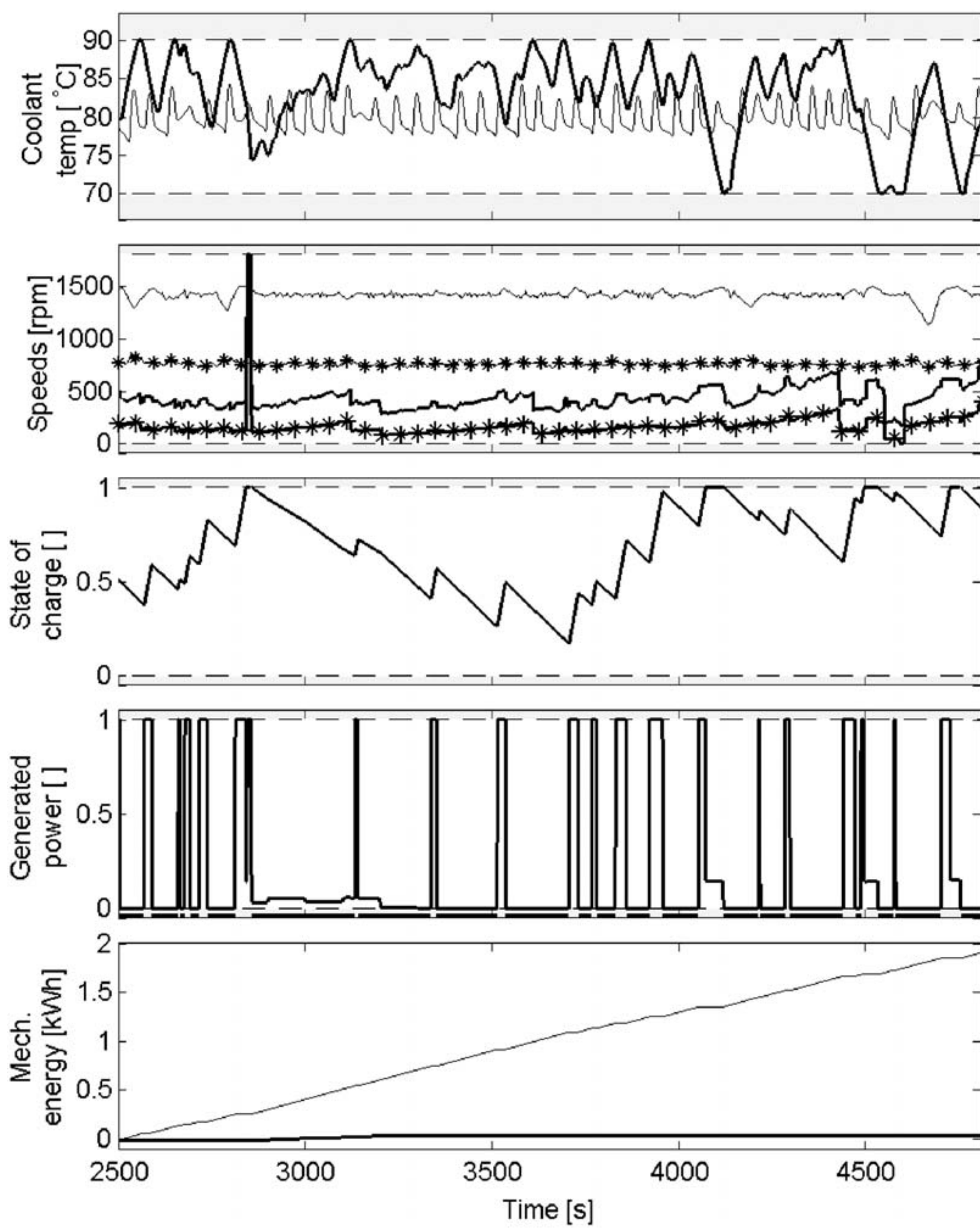

Figure 15. Simulation results for energy optimal control on the Södertälje–Norrköping road.

The variables δ and u_3 are simultaneously non-zero only in the interval 800 to 900 s in figure 14 and in the interval 350 to 800 s in figure 15. As a result, the accumulated cost to drive the auxiliary systems, shown in the lowest plot of the figures, increases only in these intervals. At all other times, the auxiliaries are run without cost.

Note that neither the optimal nor the traditional control is able to keep the temperature within the required limits in the first simulation (figure 14). With the optimal control, the electrical buffer is empty and the generator is saturated when that happens at about 1000 s, and thus, the temperature constraint cannot be satisfied. However it is notable that the optimal control solution prepares for this situation and lowers the temperature as much as possible before this occurs (the temperature is equal to $x_{1_{\min}} = 70°\mathrm{C}$ at $t = 900$ s). This can be done since the controller knows about future external influences.

5. Conclusions

Energy consumption of the auxiliary units in heavy vehicles were considered in the paper. The potential benefits of driving the auxiliaries electrically were investigated. With electrical drives the output of the auxiliaries can continuously be adjusted to the desired level and losses present in today's mechanical drives can be removed.

A simulation study over the auxiliary units as they are designed today and how they influence the overall fuel economy of a Scania vehicle indicated that the total part of the fuel consumption that can be derived from the considered auxiliary units is in the range of 4.7% to 7.3%. A model library, which can be used to evaluate novel drive systems and control principles for the auxiliary units, was developed in the modelling language Modelica. The library contains a mixture of models developed from physical principles and models fitted to collected data. Modelling of the cooling system was described in some detail. Simulation results showed good agreement with measurement data from tests in a wind tunnel. A case study on energy optimal control of the cooling and the electrical system was presented. Optimal control theory was employed to derive the control of the electrical generator, and the water pump and the cooling fan, which were both supposed to be electrically driven. The optimal controller gave a significant energy saving. The assumptions on the electrical components are preliminary, in order to give more precise estimates of the achievable energy saving, refined models need to be derived in future studies.

Acknowledgments

The authors are grateful for comments and contributions by Michael Blackenfeldt, Bo Egardt, Johan Lindström, Christer Ramdén, Erik Söderberg, Nils-Gunnar Vägstedt and Bo Wahlberg.

The work was partially supported by Scania CV AB, Swedish Programme Council for Vehicle Research, European Commission through the Network of Excellence HYCON, by the Swedish Foundation for Strategic Research through an Individual Grant for the Advancement of Research Leaders, and by the Swedish Research Council.

References

M. Back, S. Terwen and V. Krebs, "Predictive powertrain control for hybrid electrical vehicles", in *IFAC Symposium on Advances in Automotive Control*, Salerno, Italy, 2004.

P. Bengtsson, "Structuring of models intended for complete vehicle simulation", Master's thesis, Uppsala University, 2004.

US Department of Energy, Technology roadmap for the 21st century truck program. Technical report, DOE (2000).

E. Finkeldei and M. Back, "Implementing a MPC algorithm in a vehicle with a hybrid powertrain using telematics as sensor for powertrain control", in *IFAC Symposium on Advances in Automotive Control, Salerno*, Italy, 2004.

T. Hendricks and M. O'Keefe, "Heavy vehicle auxiliary load electrification for the essential power system program: Benefits, tradeos, and remaining challenges", SAE Paper 2002-01-3125, 2002.

W. Hnatczuk, M.P. Lasecki, J. Bishop and J. Goodell, "Parasitic loss reduction for 21st century trucks", SAE Paper 2000-01-3423, 2000.

L. Laine and J. Andreasson, "Modelling of generic hybrid electric vehicles", in *Proceedings of the 3rd International Modelica Conference*, 2003.

M.P. Lasecki and J.M. Cousineau, "Controllable electric oil pumps in heavy duty diesel engines", SAE Paper 2003-01-3421, 2003.

Lennart Ljung, *System Identification - Theory For the User*, 2nd ed., Upper Saddle River, NJ: Prentice Hall, 1999.

Modelica Association, *Modelica - A Unified Object-Oriented Language for Physical Systems Modeling*, Language Specification Ver. 2.0, Modelica, 2002. http://www.modelica.org

M. O'Keefe, T. Hendricks, J. Lustbader and A. Brooker, "Enhancements to NREL system analysis tools to improve auxiliary load modeling and air conditioner modeling for heavy vehicles", Technical report, NREL, 2002.

N. Pettersson, "Modeling and control of auxilary loads in heavy vehicles", Licentiate thesis, Department of Signals, Sensors and Systems, Royal Institute of Technology, Stockholm, Sweden, 2004.

N. Pettersson and K.H. Johansson, "Modelica library for simulating energy consumption of auxiliary units in heavy vehicles", in *Modelica Conference*, Linkoping, Sweden, 2003a.

N. Pettersson and K.H. Johansson, "Simulating energy consumption of auxiliary units in heavy vehicles", in *IFAC Symposium on System Identification*, Rotterdam, the Netherlands, 2003b.

N. Pettersson and K.H. Johansson, "Optimal control of the cooling system in heavy vehicles", in *IFAC Symposium on Advances in Automotive Control*, Salerno, Italy, 2004.

R. Pettersson, "Evaluation of energy savings in the truck engine achieved with control of the cooling flow", Master's thesis, Chalmers, 2002 (In Swedish).

T. Sandberg. "Heavy truck modeling for fuel consumption simulation and measurements", Licentiate thesis, Linkoping University, 2001.

M. Tiller, P. Bowles and M. Dempsey, "Development of a vehicle modeling architecture in modelica", in *Proceedings of the 3rd International Modelica Conference*, 2003.

H. Tummescheit, J. Eborn and F.J. Wagner, "Development of a modelica base library for modeling of thermohydraulic systems", in *Proceedings of Modelica Workshop*, 2000.

K.B. Wipke, M.R. Cuddy and S.D. Burch, "Advisor 2.1: A user-friendly advanced powertrain simulation using a combined backward/forward approach", Technical Report, NREL, 1999.