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Look-ahead control for heavy trucks to minimize trip time and fuel consumption

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ABSTRACT

The scenario studied is a drive mission for a heavy diesel truck. With aid of an on-board road slope database in combination with a GPS unit, information about the road geometry ahead is extracted. This look-ahead information is used in an optimization of the velocity trajectory with respect to a criterion formulation that weighs trip time and fuel consumption. A dynamic programming algorithm is devised and used in a predictive control scheme by constantly feeding the conventional cruise controller with new set points. The algorithm is evaluated with a real truck on a highway, and the experimental results show that the fuel consumption is significantly reduced.

1 INTRODUCTION

As much as about 30% of the life cycle cost of a heavy truck comes from the cost of fuel. Further, the average mileage for a (European class 8) truck is 150,000 km per year and the average fuel consumption is $32.5 \text{ l}/100\text{km}$ (Schittler, 2003). Lowering the fuel consumption with only a few percent will thus translate into significant cost reductions. These facts make a system which can reduce the fuel consumption appealing to owners and manufacturers of heavy trucks. The problem scenario in the present work is a drive mission for a truck where the route is considered to be known. It is, however, not assumed that the vehicle constantly operates on the same route. Instead, it is envisioned that there is road information on-board and that the current heading is predicted or supplied by the driver. In the current work, information about the road slope is exploited aiming at a fuel consumption reduction.

One early work (Schwarzkopf and Leipnik, 1977) formulates an optimal control problem for a nonlinear vehicle model with the aim at minimizing fuel consumption and explicit solutions are obtained for constant road slopes. A dynamic programming (DP) approach is taken from Monastyrsky and Golownykh (1993) to obtain solutions for a number of driving scenarios on shorter road sections. Inspired of some of the results indicated in these and other works it was shown in Chang and Morlok (2005); Fröberg et al. (2006) with varying vehicle model complexity that constant speed is optimal on a constant road slope within certain bounds on the slope. The result relies on that there is an affine relation between the fuel consumption and the produced work. Analytical studies of the situation when this relation is nonlinear were conducted in Fröberg and Nielsen (2007). Predictive cruise control is investigated through computer simulations in, e.g., Lattemann et al. (2004); Terwen et al. (2004), but constructing an optimizing controller that works on-board in a real environment puts additional demands on the system in terms of robustness and complexity.

In Hellström et al. (2006) a predictive cruise controller (CC) is developed where discrete DP is used to numerically solve the optimal control problem. The current paper is a continuation where an improved approach is realized and evaluated in actual experiments. One objective is to evaluate the order of fuel reduction that can be obtained in practical driving. The strategies to achieve fuel reductions may be intuitive, but only in a qualitatively manner. Another objective is therefore to find the optimal solution and thereby quantify the characteristics of the best possible strategy. The purpose is also to analyze controller behavior in real trial runs and evaluate potential benefits.

To perform this study a chain of elements is needed. Section 2 presents a vehicle model of standard type. Section 3 deals with the DP algorithm, and it is described how the problem characteristics are utilized to reduce the complexity, to determine penalty parameters, and efficiently compute the criterion by an inverse technique. In Section 4 the experimental setup is presented, and finally the quantitative evaluation concludes the study.

2 TRUCK MODEL

The modeling follows Kiencke and Nielsen (2005); Sandberg (2001), and the resulting model is then reformulated and adapted for the numerical optimization that is performed.

The engine torque T_e is modeled as

$$T_e(\omega_e, u_f) = a_e \omega_e + b_e u_f + c_e \quad (1)$$

where ω_e is the engine speed and u_f is the control signal which determines the fueling level.

The control u_f is assumed to be bounded by

$$0 \leq u_f \leq u_{f,max}(\omega_e) \quad (2)$$

where the upper limit $u_{f,max}(\omega_e)$ is modeled by a second-order polynomial in engine speed ω_e ,

$$u_{f,max}(\omega_e) = a_f \omega_e^2 + b_f \omega_e + c_f.$$

When a gear is engaged, the engine transmits a torque T_c to the clutch and

$$J_e \dot{\omega}_e = T_e - T_c \quad (3)$$

where J_e is the engine inertia and ω_e is the engine speed. The clutch, propeller shafts and drive shafts are assumed stiff and their inertia are lumped into one together with the wheel inertia, denoted by J_l . The resulting conversion ratio of the transmission and final drive is denoted by i and energy losses are modeled by an efficiency η . When a gear is engaged, this gives

$$\begin{aligned} \omega_e &= i \omega_w, \quad T_w = i \eta T_c \\ J_l \dot{\omega}_w &= T_w - T_b - r_w F_w \end{aligned} \quad (4)$$

where T_w is the torque transmitted to the wheel, T_b is the braking torque and r_w is the wheel radius. F_w is the resulting friction force.

When neutral gear is engaged, the engine transmits zero torque to the driveline. The driveline equations (3) and (4) then become

$$\begin{aligned} J_e \dot{\omega}_e &= T_e, \quad T_c = T_w = 0 \\ J_l \dot{\omega}_w &= -T_b - r_w F_w. \end{aligned} \quad (5)$$

The motion of the truck is governed by

$$m \frac{dv}{dt} = F_w - F_a(v) - F_r(\alpha) - F_N(\alpha) \quad (6)$$

where α is the road slope. The models of the longitudinal forces are explained in Table 1.

It is assumed that the transmission is of the automated manual type and that gear shifts are accomplished through engine control, see Pettersson and Nielsen (2000). A shift is modeled by a constant period of time τ_{shift} where the neutral gear is engaged

Table 1: Longitudinal forces.

Force	Explanation	Expression
$F_a(v)$	Air drag	$\frac{1}{2}c_w A_a \rho_a v^2$
$F_r(\alpha)$	Rolling resistance	$mgc_r \cos \alpha$
$F_N(\alpha)$	Gravitational force	$mg \sin \alpha$

before the new gear is engaged. The number of the currently engaged gear is denoted by g . The ratio i and efficiency η then becomes functions of the integer g . The control signal that selects gear is denoted by u_g . Neutral gear corresponds to gear zero, equivalent with a ratio and efficiency of zero.

The vehicle velocity v is

$$v = r_w \omega_w \quad (7)$$

where ω_w is the wheel speed of revolution and r_w is the effective wheel radius. Equations (3)-(7) are combined into

$$\frac{dv}{dt}(x, u, \alpha) = \frac{r_w}{J_l + mr_w^2 + \eta(g)i(g)^2 J_e} \left(i(g)\eta(g)T_e(v, u_f) - T_b(u_b) - r_w (F_a(v) + F_r(\alpha) + F_l(\alpha)) \right) \quad (8)$$

where

$$x = [v, g]^T \quad u = [u_f, u_b, u_g]^T \quad (9)$$

denote the state and control vector, respectively. The states are the velocity v and currently engaged gear g and the controls are fueling u_f , braking u_b and gear selector u_g .

The mass flow of fuel is determined by the fueling level u_f (g/cycle) and the engine speed ω_e (rad/s). The flow in (g/s) is then

$$\frac{n_{cyl}}{2\pi n_r} \omega_e u_f \quad (10)$$

where n_{cyl} is the number of cylinders and n_r is the number of crankshaft revolutions per cycle. Using (4) and (7) in (10) gives

$$\dot{m}(x, u) = \frac{n_{cyl}}{2\pi n_r} \frac{i}{r_w} v u_f, \quad g \neq 0 \quad (11)$$

whereas in the case of neutral gear, $g = 0$, the fuel flow is assumed constant and equal to an idle fuel flow \dot{m}_{idle} .

2.1 REFORMULATION

Models (8) and (11) are transformed to be dependent on position rather than time. Denoting traveled distance by s and the trip time by t , then for a function $f(s(t))$

$$\frac{df}{dt} = \frac{df}{ds} \frac{ds}{dt} = \frac{df}{ds} v \quad (12)$$

is obtained using the chain rule where $v > 0$ is assumed. By using (12), the models can be transformed as desired.

The approach in this work is numerical and therefore the model equations should be made discrete. The quantization step in position is constant and equal to h . The control signals are considered piece-wise constant during a discretization step. Denote

$$\begin{aligned} x_k &= x(kh), u_k = u(kh) \\ \alpha_k &= \frac{1}{h} \int_{kh}^{(k+1)h} \alpha(s) ds \end{aligned} \quad (13)$$

where the road slope α_k is set to the mean value over the discretization step. The trapezoidal rule is used to make the truck model (8) discrete. If a gear shift occurs during a step, a second-order Runge-Kutta method is used for a time step equal to the delay τ_{shift} to modify the initial values and the step length. The system dynamics is finally

$$x_{k+1} = f(x_k, u_k, \alpha_k) \quad (14)$$

where $f(x_k, u_k, \alpha_k)$ is given by (8).

The discretized problem is incorporated into the algorithm and thus affects the algorithm complexity. The simplest Euler method do, however, not yield satisfactory results due to truncation errors, see Hellström (2007). For this reason second-order methods were chosen.

3 LOOK-AHEAD CONTROL

Look-ahead control is a predictive control scheme with additional knowledge about some of the future disturbances, here focusing on the road topography ahead of the vehicle. An optimization is thus performed with respect to a criterion that involves predicted future behavior of the system, and this is accomplished through DP (Bellman and Dreyfus, 1962).

The conditions change during a drive mission due to disturbances, e.g., delays due to traffic, or changed parameters such as the vehicle mass. The robustness is increased by feedback and the approach taken here is therefore to repeatedly calculate the fuel-optimal control on-line. The principle is shown in Figure 1. At point A, the optimal solution is sought for the problem that is defined over the look-ahead horizon. This horizon is obtained by truncating the entire drive mission horizon. Only the first optimal control is applied to the system and the procedure is repeated at point B.

This section will first deal with the identification of the control objectives. Based on these, a suitable criterion is devised and the tuning of its parameters is discussed. At the end, the DP algorithm is outlined.

3.1 OBJECTIVE

The objectives are to minimize the energy and time required for a given drive mission. The vehicle velocity is desired to be kept inside an interval

$$v_{min} \leq v \leq v_{max} \quad (15)$$

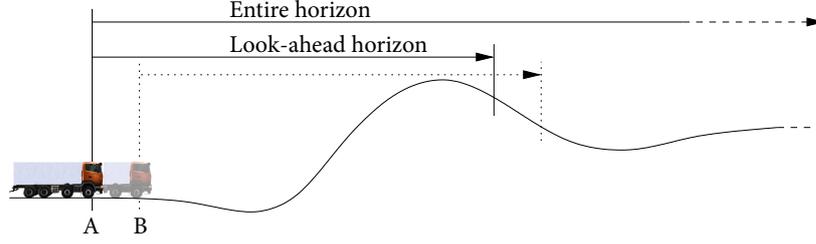


Figure 1: Illustration of the look-ahead control strategy.

where v denotes the vehicle velocity. These bounds are set with respect to the desired behavior of the controller. For example, the lower bound will be the lowest velocity the controller would deliberately actuate. The upper bound can be set by, e.g., safety reasons or legal considerations.

The brake system is assumed to be powerful enough to keep the upper bound in (15). On the other hand, the lower bound is not expected to be physically reachable on all road profiles. It is assumed though, that it is possible to keep a velocity, denoted by v_{lim} , which is positive at all times. If Equation (15) were to be used, it would not be certain to find any feasible solution. Therefore the constraints on the vehicle speed v are expressed as follows:

$$0 < \min \{v_{min}, v_{lim}(s)\} \leq v \leq v_{max} \quad (16)$$

3.2 CRITERION

The fundamental trade-off when studying minimization of energy required for a drive mission is between the fuel use and the trip time. The fuel use on a trip from $s = s_0$ to $s = s_f$ is

$$M = \int_{s_0}^{s_f} \frac{1}{v} \dot{m}(x, u) ds \quad (17)$$

where $\frac{1}{v} \dot{m}(x, u)$ is the mass flow per unit length as function of the state x and control u . The trip time T is simply

$$T = \int_{s_0}^{s_f} \frac{dt}{ds} ds = \int_{s_0}^{s_f} \frac{ds}{v}. \quad (18)$$

To weigh fuel and time use, the cost function proposed is

$$I = M + \beta T \quad (19)$$

using (17) and (18) and where β is a scalar factor which can be tuned to receive the desired trade-off.

Criterion (19) is then made suitable for discrete DP by dividing the look-ahead horizon into N steps of length h (m) and transforming the cost function. Denote

$$m_k = \int_{kh}^{(k+1)h} \dot{m}(x, u) ds, \quad t_k = \int_{kh}^{(k+1)h} \frac{ds}{v}, \quad a_k = |v_k - v_{k+1}| \quad (20)$$

and the cost function can be expressed as

$$J = \sum_{k=0}^{N-1} \zeta_k(x_k, x_{k+1}, u_k, \alpha_k) \quad (21)$$

where

$$\zeta_k = [1, \beta, \varepsilon] \begin{bmatrix} m_k \\ t_k \\ a_k \end{bmatrix}, \quad k = 0, 1, \dots, N-1 \quad (22)$$

and β, ε are scalar penalty parameters for controlling the properties of solutions. The difference in the criterion between neighboring discretization points is typically very small. In order to receive a smoother control, the term a_k is added with a small value of ε .

3.3 PENALTY PARAMETERS

The size of the factor ε is chosen for smoothing but still such that the term εa_k becomes considerable smaller than the others.

One way to determine the parameter β , i.e. the trade-off between fuel and time, is to study a stationary solution to the criterion in Equation (19). Assume that a gear is engaged and there exists at least one control \hat{u} , for which (2) holds and that gives a stationary velocity \hat{v} , i.e. $dv/dt = 0$. From the equations (1), (8), and Table 1 it is concluded that \hat{u} can be written as

$$\hat{u} = c_1 \hat{v}^2 + c_2 \hat{v} + f(\alpha) \quad (23)$$

where, for a given gear, c_1 and c_2 are constants and $f(\alpha)$ is a function corresponding to the rolling resistance and gravity, and thus being a function of the road slope α . With (11) and (12), the fuel flow is written as

$$\frac{1}{v} \dot{m}(x, u) = c_4 \hat{u}_f \quad (24)$$

where c_4 is the proportionality constant. The cost function (19) is thus

$$\hat{I}(\hat{v}) = \int_{s_0}^{s_f} \left(c_4 (c_1 \hat{v}^2 + c_2 \hat{v} + f(\alpha)) + \frac{\beta}{\hat{v}} \right) ds \quad (25)$$

where the integrand is constant with respect to s if constant slope is assumed. A stationary point to \hat{I} is found by setting the derivative equal to zero,

$$\frac{d\hat{I}}{d\hat{v}} = \int_{s_0}^{s_f} \left(c_4 (2c_1 \hat{v} + c_2) - \frac{\beta}{\hat{v}^2} \right) ds = 0. \quad (26)$$

Solving the equation for β gives

$$\beta = c_4 \hat{v}^2 (2c_1 \hat{v} + c_2) \quad (27)$$

that can be interpreted as the value of β such that a stationary velocity \hat{v} is the solution to (26). Note that the value of β neither depends on the vehicle mass m nor the slope α . The calculated β will thus give the solution \hat{v} of the criterion for any fixed mass and slope as long as there exists a control \hat{u}_f satisfying the bounds in (2).

3.4 PREPROCESSING

The ambition with the present work is a real-time algorithm and hence the complexity plays an important role. The subset of the state space over which the optimization is applied, the search space, is one determining factor for the complexity. If the search space is reduced without losing any solutions, obvious gains are made. A preprocessing algorithm is therefore developed with this aim.

Since DP is used in an predictive control setting, the current velocity can be measured and used for limiting the set of possible initial states. In order to handle terminal effects, the final velocities are also constrained. By using the model and traversing the horizon forward and backward before the optimization is started, the search space is downsized, see Hellström (2007).

The preprocessing algorithm gives, for each stage, an interval of velocities which are to be considered. For every stage the interval $[v_{lo}, v_{up}]$ is discretized in constant steps of δ . This makes up a set V_k ,

$$V_k = \{v_{lo}, v_{lo} + \delta, v_{lo} + 2\delta, \dots, v_{up}\}. \quad (28)$$

3.5 DP ALGORITHM

To summarize, the optimal control problem at hand is the minimization of the objective,

$$\min_{u, g} \sum_{k=0}^{N-1} \zeta_k(x_k, x_{k+1}, u_k, \alpha_k)$$

where ζ_k is given in (22). The system dynamics is given by

$$x_{k+1} = f(x_k, u_k, \alpha_k) \quad k = 0, 1, \dots, N-1$$

according to (14). The constraints are

$$0 < \min\{v_{min}, v_{lim}(kh)\} \leq v_k \leq v_{max} \quad \forall k$$

according to (16). Due to the predictive control setting, the initial state x_0 is given.

With a given velocity, only a subset of the gears in the gearbox is feasible. If the operating region of the engine is defined with bounds on the engine speed $[\omega_{e,min}, \omega_{e,max}]$, it is easy to select the set of feasible gears. Only gears with a ratio that gives an engine speed in the allowed range are then considered. In a state with velocity v , the set of usable gears G_v is thus defined as

$$G_v = \{g \mid \omega_{e,min} \leq \omega_e(v, g) \leq \omega_{e,max}\} \quad (29)$$

where $\omega_e(v, g)$ is the engine speed at vehicle velocity v and gear number g .

Braking is only considered in the algorithm if the upper velocity bound is encountered. Braking without recuperation is an inherent waste of energy and therefore braking will only occur when the velocity bounds would otherwise be violated. This reduces the complexity since the number of possible control actions lessens.

Table 2: Truck specifications.

Component	Type	Characteristics
Engine	DC9	Cylinders: 5 Displacement: 9 dm ³ Max.torque: 1,550 Nm Max.power: 310 hp
Gearbox	GRS890R	12 gears
Vehicle	-	Total weight: 39,410 kg

A state x is made up of velocity v and gear number g . The possible states in stage k are denoted with the set S_k and it is generated from the velocity range V_k given in (28) and the set of gears G_v given in (29). This yields

$$S_k = \{ \{v, g\} \mid v \in V_k, g \in G_v \}. \quad (30)$$

At a stage k , feasible control actions $u_k^{i,j}$ that transform the system from a state $x^i \in S_k$ to another state $x^j \in S_{k+1}$ are sought. The control is found by an inverse simulation of the system equations, see e.g. Fröberg and Nielsen (2008). Here this means that for a given state transition, x_k to x_{k+1} , the control u_k is solved from (14). Interpolation is thereby avoided. If there are no fueling level u_f and gear u_g that transforms the system from state x^i to x^j at stage k , there are two possible resolutions. If there exist a feasible braking control u_b it is applied, but if there is no feasible braking control the cost is set to infinity.

The algorithm is outlined as follows:

1. Let $J_N(i) = 0$.
2. Let $k = N - 1$.
3. Let

$$J_k(x^i) = \min_{x^j \in S_{k+1}} \{ \zeta_k^{i,j} + J_{k+1}(x^j) \}, \quad x^i \in S_k.$$

4. Repeat (3) for $k = N - 2, N - 3, \dots, 0$.
5. The optimal cost is J_0 and the sought control is the optimal control set from the initial state.

4 TRIAL RUN

The experiments are performed on the highway E4 between the cities of Södertälje and Norrköping in Sweden, see Figure 4. The truck used is a Scania tractor and trailer, see Figure 2. The specifications are given in Table 2.

Following in this section, the experimental setup and road slope data are presented. The last part of the section will present some results from the trial runs that have been undertaken.



Figure 2: The vehicle used in the experiments.

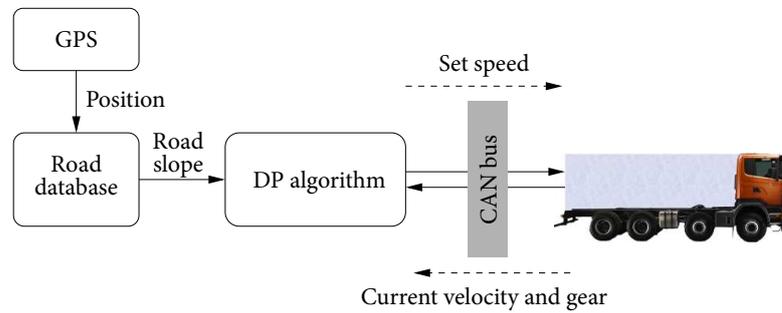


Figure 3: Information flow.

4.1 SETUP

The information flow in the experimental setup is shown schematically in Figure 3. Due to adjustments for the demonstrator vehicle, gear shifts were not directly controlled by the algorithm. This is handled by including a prediction model of the gear control system and take it into account when calculating the running costs. In the optimization algorithm, a shift that is not predicted is assigned an infinite cost. As depicted in Figure 3 the algorithm controls the vehicle by adjusting the set speed to the conventional CC. The fueling level is therefore only controlled indirectly. The standard CC available from Scania is used, which is basically a PI-regulator. All communications are done over the CAN bus.

The algorithm parameters used are stated in Table 3 and the penalty factors are shown in Table 4. The factors are adjusted in order to receive a stationary solution in the middle of the desired velocity interval (15).

All software run on a portable computer with an Intel Centrino Duo processor at 1.20 GHz and 1 Gb RAM. With the stated parameters, a solution on a road stretch of level road is calculated in tenths of a second on this computer.

The truck has a legislative speed limiter at 89 km/h. Propulsion above this limit is not possible. When the truck accelerates due to gravity above 89 km/h, the brake control

Table 3: User parameters

Parameter	Function	Value
h	Step length	50 m
N	Number of steps	30
$h \cdot N$	Horizon	1500 m
δ	Velocity discret.	0.2 km/h
v_{min}	Min. allowed vel.	79 km/h
v_{max}	Max. allowed vel.	89 km/h

Table 4: Penalty factors

Factor	Penalizes	Value
	Fuel use	1.0
β	Time use	6.2
ε	Velocity changes	0.1

system is activated at a set maximum speed. In the trial run this limit is set to be 91 km/h.

4.1.1 DATABASE

The slope in front of the vehicle for the length of the look-ahead horizon is needed to be known. It is expected that such data will be commercially available soon. But for now, the road slope along the trial route is estimated off-line prior to any experiments. This is done by aid of a non-stationary forward-backward Kalman filter (Sahlholm et al., 2007). The estimated slope and elevation are shown in Figure 4. The measurements were obtained at 20 Hz from a GPS unit. The filter inputs are vertical and horizontal velocity of the vehicle, elevation and the number of reachable GPS satellites.

4.2 PERFORMANCE

In total, five comparative trial runs were made. All runs were done in light to moderate traffic, and each consisted of one drive with look-ahead control and one with standard cruise control. The algorithm parameters, see Table 3 and 4, were the same for all runs. The trip time thus became about the same for all drives with the look-ahead control. The set point for the CC was chosen in order to achieve a trip time close to the one obtained with look ahead.

4.3 OVERALL RESULTS

The relative changes in fuel consumption and trip time (Δ_{fuel} , Δ_{time}) are shown in Figure 5 and Figure 6 for each direction on the trial road. A negative value means that the look-ahead controller (LC) has lowered the corresponding value. The set point for the CC increases along the horizontal axis. The left-most result is maybe the most convincing since it reduces both fuel use and trip time in both directions.

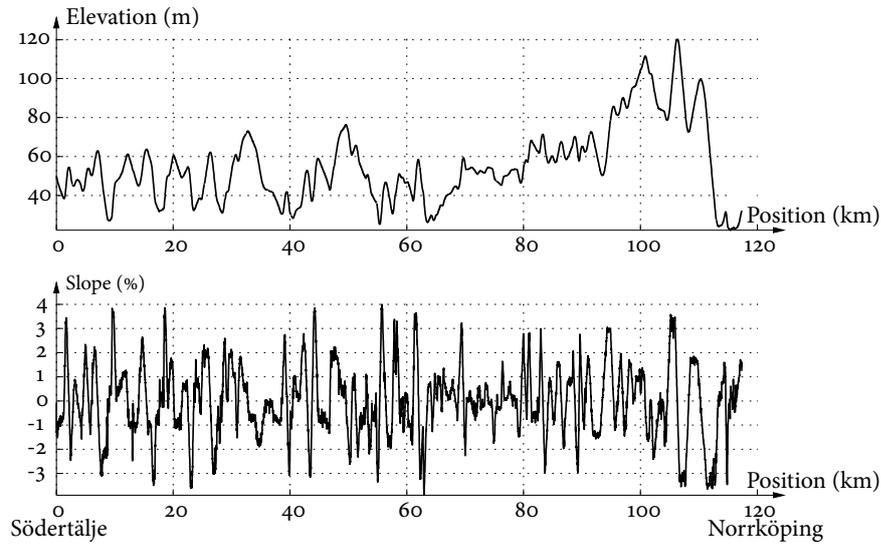


Figure 4: Estimated road topography.

The average results in both directions that are made with the same set speed are also calculated. For these mean values the fuel consumption was lowered with 3.53%, from 36.33 l/100km to 35.03 l/100km , with a negligible reduction of the trip time (0.03%) in comparison with the CC. Also interesting to note is that the mean number of gear shifts on this route decreases from 20 to 12 with the LC.

4.4 CONTROL CHARACTERISTICS

With the intention to give a representative demonstration of more detailed controller characteristics, two road segments have been chosen. The first is a 2.5 km segment close to Södertälje and is named *the Järna segment*. The second one is a 3.5 km segment about halfway on the trial route and called *the Hället segment*.

Each figure, see, e.g., Figure 7, is divided into four subfigures, all having the position as the horizontal axis. The road topography is shown at the top and the coordinates for the start and final position are also given on the horizontal axis. The next subfigure shows the velocity trajectories for the LC and the standard CC. The third part shows normalized fueling (acc) and auxiliary brake (brake) levels with thick and thin lines, respectively. At the bottom, both the engaged gear number and the fuel use are shown. Data related to the LC are displayed in solid lines and data associated to the CC are displayed with dashed lines in these figures. Above the figures, the time and fuel spent on the section are shown together with the relative change (Δfuel , Δtime) in these values between the two controllers. A negative value means that the value is lowered by the LC.

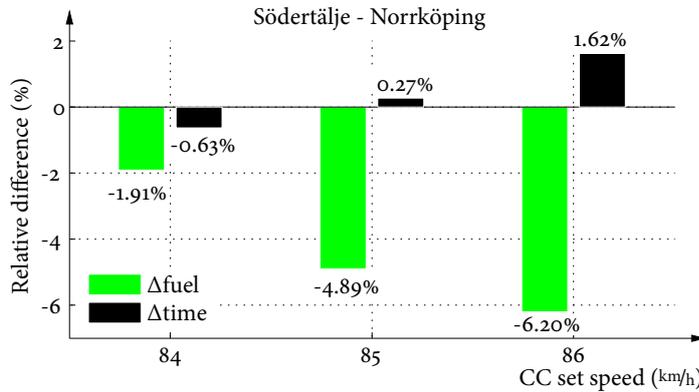


Figure 5: Trial run results on the road from Södertälje to Norrköping with varying cruise controller (CC) set speed.

4.4.1 THE HÅLLET SEGMENT

Figures 7 and 8 show the Hållet segment. In Figure 7, the LC accelerates at 500 m prior to the uphill that begins at 750 m. At the top of the hill at 1750 m, the LC slows down in contrast to the CC. The truck is thus let to accelerate by gravity alone. The CC will, however, use a non-zero fueling as long as the truck is going slower than the set point. The LC slow down reduces the need for braking later in the downslope and thereby the inherent waste of energy is lessened. From the fuel integral at the bottom, it is seen that the LC consumes more fuel the first 1.5 km owing to the acceleration, but then gains.

The trip in the other direction, see Figure 8, gives similar features. A gain of speed at 250 m and then a slow down at the top of the hill at 2250 m. In both directions, time as well as fuel are saved.

Note that the sections in Figure 7 and 8 are not exceptionally steep. The uphill and downhill slope is at most about 4% for short intervals. However, they become significant for the truck due to the large vehicle mass.

4.4.2 THE JÄRNA SEGMENT

In Figures 9 and 10 the Järna segment is shown. Figure 9 shows that the LC begins to gain speed at 200 m and thereby avoids the gear shift that the CC is forced to do around 1 km. At 1400 m, the LC slows down and lets the truck accelerate in the downslope.

In Figure 10, a drive in the other direction is shown. The LC accelerates at 500 m and starts to slow down at 1400 m. The slow down lessen the braking effort needed at about 2000 m.

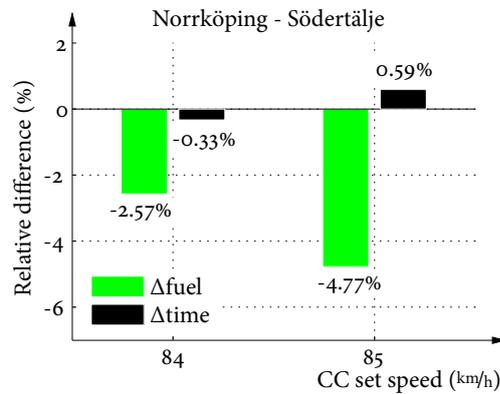


Figure 6: Trial run results on the road from Norrköping to Södertälje with varying cruise controller (CC) set speed.

5 CONCLUSIONS

The control algorithm was proven to perform well on-board in a real environment. Using the standard cruise controller as an inner loop and feeding it with new set points is advantageous considering robustness against model errors and disturbances.

The gearbox consists of a set of discrete gears and there is no propulsion force during a gear shift. Taking these facts into account renders a challenging optimization problem. A discrete dynamic programming algorithm is devised where the search space is reduced by a preprocessing algorithm. The algorithm computes a solution in tenths of a second on a modern laptop computer and this allows evaluation in a real environment on-board a truck.

The trial runs show that significant reductions of the fuel consumption can be achieved. A fuel consumption reduction of about 3.5% on the 120 km route without an increase in trip time was obtained. The mean number of gear shifts was reduced with 42% due to shifts avoided by gaining speed prior to uphill.

The look-ahead control mainly differs from conventional cruise control near significant downhills and uphills where the look-ahead control in general slows down or gains speed prior to the hill. Slowing down prior to downhills is intuitively saving fuel. There is, however, no challenge in saving fuel by traveling slower, so if the vehicle is let to slow down at some point, the lost time must thus be gained at another point. Accelerating prior to uphills is one way which, at least for shorter hills, gives a higher velocity throughout the hill and will reduce the need for lower gears. These strategies are intuitive but the crucial issue is the detailed shape of the solution and its actuation such that a positive end result is obtained, and this is shown to be handled well by the algorithm.

A final comment is that the controller behavior has been perceived as comfortable and natural by drivers and passengers that have participated in tests and demonstrations.

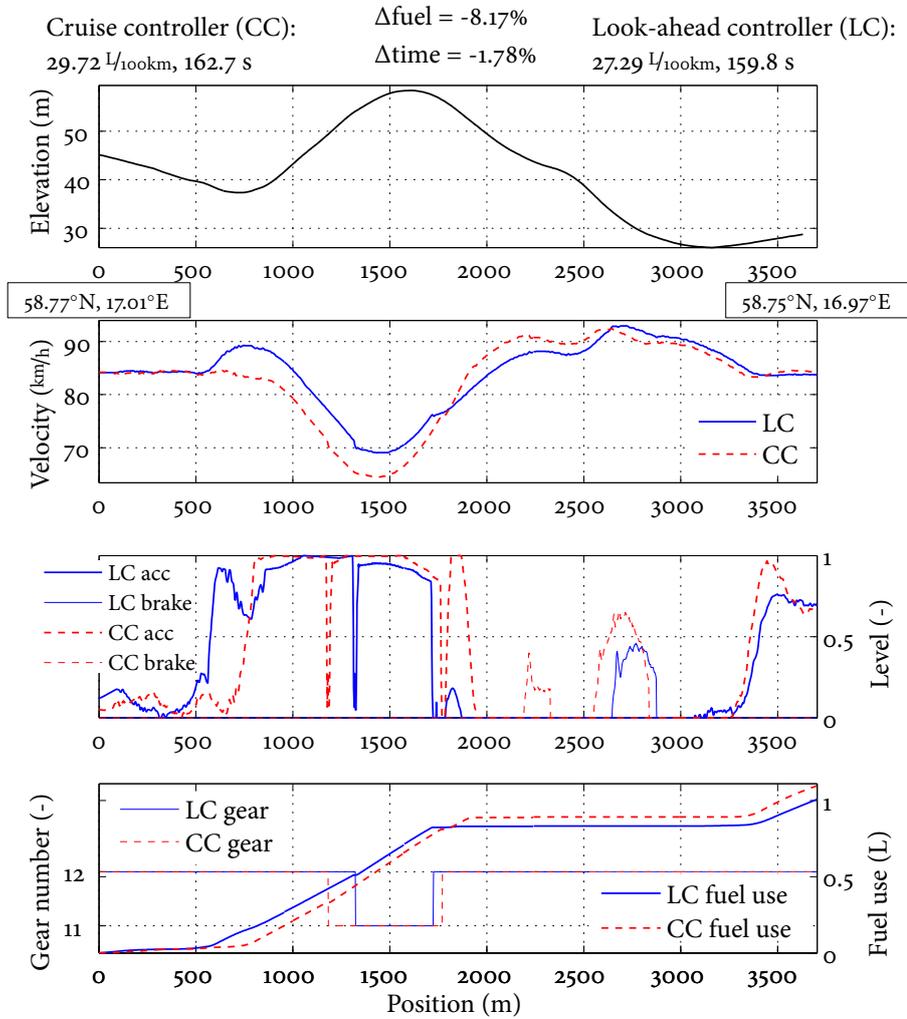


Figure 7: The Hället segment. The LC accelerates at 500 m prior to the uphill and slows down at 1750 m when the top is reached.

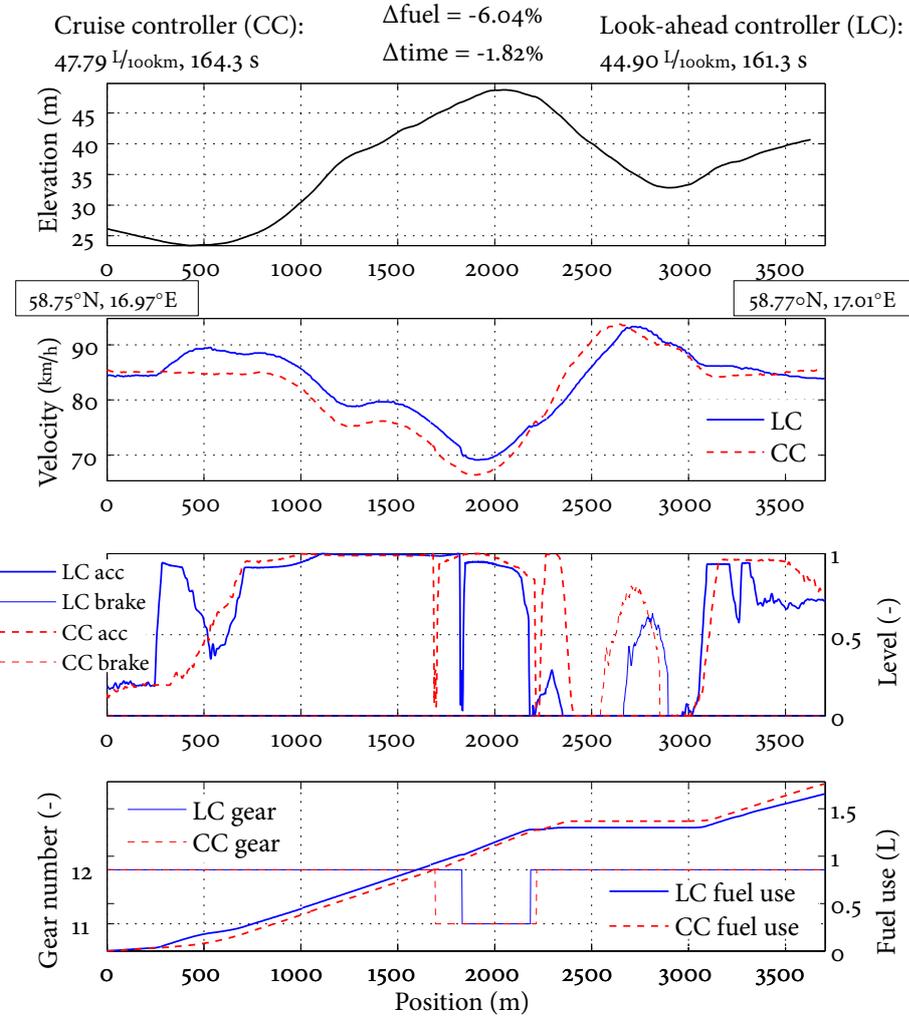


Figure 8: The Hället segment. The LC gains speed at 250 m prior to the uphill and slows down at 2250 m prior to the downhill.

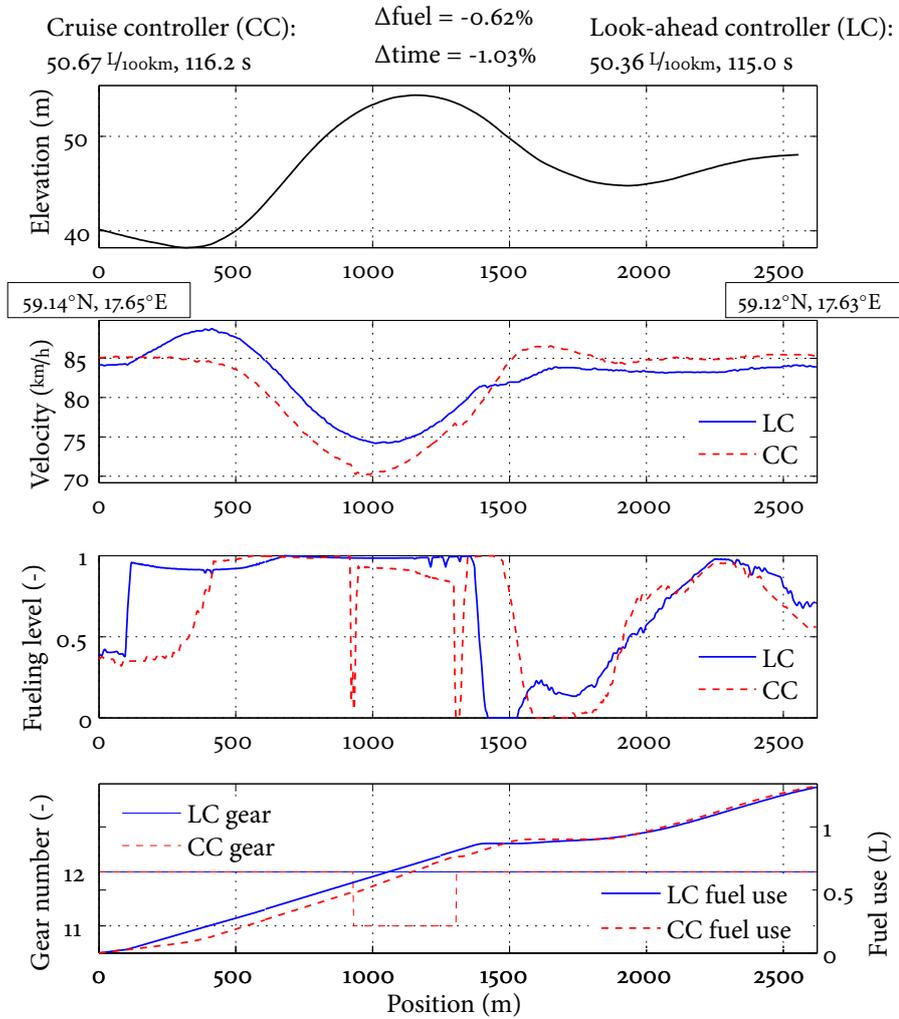


Figure 9: The Järna segment. The LC gains speed at 200 m prior to the uphill and avoids a gear shift. At 1400 m the LC slows down and the truck is let to accelerate in the downslope.

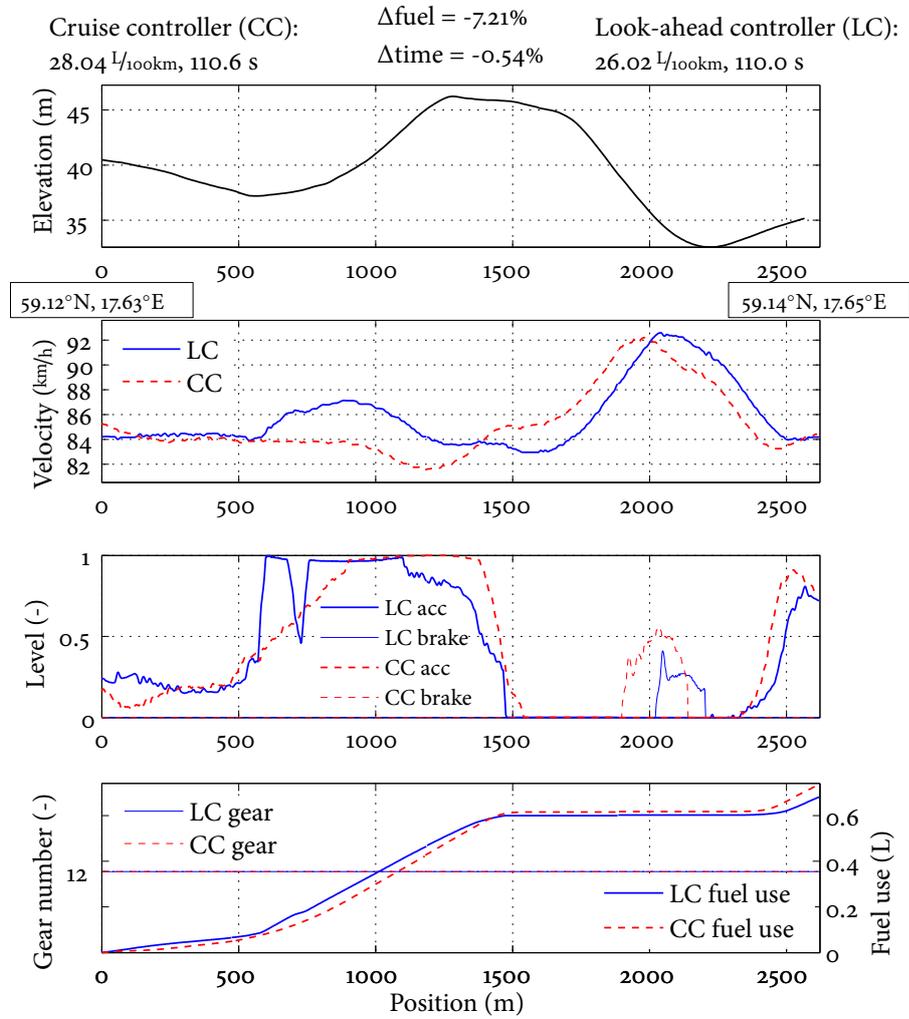


Figure 10: The Järna segment. The LC accelerates at 500 m and slows down at 1400 m thereby reducing the braking effort needed later on.

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