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Optimal Powertrain Control of a Heavy-Duty Vehicle Under Varying Speed Requirements

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***Abstract*—Reducing the fuel consumption is a major issue in the vehicle industry. In this paper, it is done by formulating a driving mission of a heavy-duty truck as an optimal control problem and solving it using dynamic programming. The vehicle model includes an engine and a gearbox with parameters based on measurements in test cells. The dynamic programming algorithm is solved by considering four specific types of transitions: transitions between the same gear, coasting in neutral gear, coasting with a gear engaged with no fuel injection and transitions involving gear changes. Simulations are performed on a driving cycle commonly used for testing distribution type of driving. In order to make sure that the truck does not deviate too much from a normal way of driving, restrictions on maximum and minimum allowed velocities are imposed based on statistics from real traffic data. The simulations show that 12.7 % fuel can be saved without increasing the trip time by allowing the truck to engage neutral gear and make small deviations from the reference trajectory.**

I. INTRODUCTION

Manufacturers of heavy-duty trucks are constantly striving to reduce the fuel consumption of their vehicles. One way of doing so is by improving the hardware of the vehicle, e.g. by increasing the efficiency of the engine or the gearbox. Another way, which is the focus of this paper, is by improving the software. An example of such a solution is the look-ahead cruise controller, which takes future road topography into consideration when controlling the velocity. An average fuel saving of 3 % has been reported [1]. This kind of solution today mainly considers highway driving where the desired cruising velocity is constant.

Fuel efficient ways of driving is often found by formulating the driving mission as an optimal control problem, see [2] for an overview of such methods in automotive applications. In [3], the optimal control of a heavy-duty vehicle driving on a highway is found using dynamic programming (DP). The solution to an optimal control problem can either be precomputed and used as a trajectory to be tracked by a controller, or it can be found online while driving. Finding a solution using DP is often computationally heavy due to the curse of dimensionality, i.e., the fact that the computation time grows exponentially with the number of states and control signals. In order to make it real-time implementable, algorithms reducing the search space using heuristics based on energy and time consumption can be used [4]. In [5], such methods are used for solving a problem regarding the minimization of the fuel consumption in the vicinity of signalized intersections. Another way to reduce the amount of computations performed in the vehicle is to solve the optimal control problem in the cloud [6]. In [7], an optimal cruise control algorithm is developed using DP for a hybrid electrical vehicle having its velocity limited by a driving corridor.

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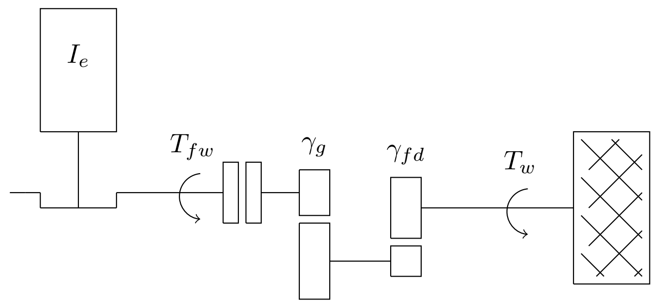


Fig. 1: The powertrain including from the left: engine, flywheel, gearbox, final drive and driving wheel.

The fact that an engine's efficiency depends on engine speed and torque is considered in the optimization problem in [8]. In [9], an optimization problem involving a vehicle model based on forces is solved by using Pontryagin's maximum principle. The same principle is used in [10], where gear changes are also considered. In [11], the concept of a driving corridor is used, i.e., an upper and a lower limit that the velocity of the vehicle must lie within. The same problem as in [9] is treated, but with more thorough simulations in order to investigate how variations in the driving corridor influence the energy consumption and trip time. The work in this paper extends [9] and [11] by including powertrain components such as an engine and a gearbox in the vehicle model. This is a step towards more implementable solutions, since the tractive force is replaced by the injected fuel and the gear selection is added as a control signal.

The main contribution of this paper is twofold. The first contribution is the application of DP to a driving mission of a heavy-duty truck with varying requirements on the allowed

TABLE I: Parameters related to the truck and the environment.

Parameter	Value
m - truck mass	26 000 kg
r_w - wheel radius	0.5 m
P_{max} - maximum power	250 kW
c_d - air drag coefficient	0.5
ρ - air density	1.292 kg$\cdot$ m^{-3}
A - truck cross-sectional area	10 m^2
c_r - rolling resistance coefficient	0.006
γ_{fd} - final drive ratio	2.92
ω_{idle} - idle engine speed	500 RPM
ω_{min} - mimimum engine speed	800 RPM
ω_{max} - maximum engine speed	2400 RPM
I_e - moment of inertia engine	4 kg$\cdot$m^2
I_w - moment of inertia wheels	92 kg$\cdot$m^2
τ_g - time of a gear change	1 s
e_{ideal} - ideal specific energy	48 MJ/kg
γ_c - energy to fuel efficiency	0.45

speed where the boundary conditions are set based on real traffic data. The second contribution is the use of a vehicle model with a fuel map and gearbox losses derived from experiments. This allows the solution to have realistic gear changes and the possibility to coast using neutral gear.

II. Vehicle model

The model of the powertrain of the truck can be seen in Fig. 1 and each part will be described in the following subsections. Parameters related to the truck and the environment are given in Table I.

A. Chassis dynamics

The dynamics of the truck are given by:

$$\frac{dv}{dt} = \frac{1}{mc_m}\Big(\frac{T_w}{r_w} + F_{env} - F_b\Big) \tag{1}$$

where v and m are the velocity and mass of the truck, T_w is the torque at the wheels, r_w is the wheel radius and F_b is the force at the brakes. The variable c_m represents a mass factor in order to take the moment of inertia of the wheels and engine into account and is given by

$$c_m = \frac{mr_w^2 + I_w + (\gamma_g\gamma_{fd})^2 I_e}{mr_w^2} \tag{2}$$

where I_w and I_e are the moment of inertia of the wheels and engine respectively and γ_g and γ_{fd} are the transmission ratios of the current gear and the final drive respectively. If neutral gear is engaged, then $\gamma_g = 0$. The sum of the environmental forces F_{env} in (1) is given by

$$F_{env} = -mg\cos(\alpha)c_r - \frac{1}{2}\rho A c_d v^2 - mg\sin\alpha \tag{3}$$

where g is the gravitational constant, α is the road slope, c_r is the coefficient for the rolling resistance, ρ is the air density, A is the truck frontal area and c_d is the air drag coefficient.

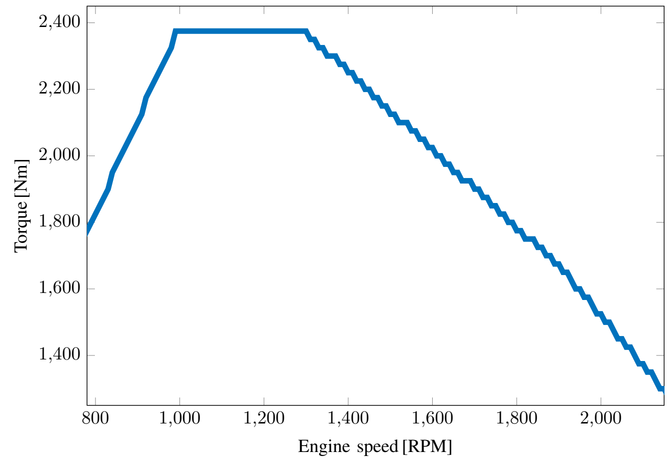


Fig. 2: Maximum flywheel torque $T_{max}(\omega)$ given as a function of engine speed in revolutions per minute.

B. Engine

The flywheel torque T_{fw} is given by

$$T_{fw} = T_e - T_d \tag{4}$$

where T_e is the engine torque created by the combustion in the engine and T_d is the drag torque created by friction in the engine. The flywheel torque T_{fw} is limited such that $T_{fw} \leq T_{max}(\omega)$ where $T_{max}(\omega)$ is a function of engine speed ω that can be seen in Fig. 2. The engine drag torque is a function of engine speed and is taken from previous experiments in a test cell. In the simulations in this paper, it is approximated by the function

$$T_d = d_0 + d_1\omega \tag{5}$$

where d_0 and d_1 are found by a linear least square fit to the experimental values. The minimum possible flywheel torque is negative and is obtained when no fuel is injected to the engine. In this case, the engine torque $T_e = 0$ and the flywheel torque is created solely by the engine drag torque. The fuel flow $\dot{m}_f$ is given in kg/s by a fuel map in the form of a look-up table as a function of engine speed and flywheel torque such that

$$\dot{m}_f = \dot{m}_f(\omega, T_{fw}). \tag{6}$$

This look-up table is taken from previous experiments in test cells. Two specific parts of the fuel map are of special interest. The first is when neutral gear is engaged and is referred to as *coasting in neutral*. In this case, the engine torque should be enough to overcome the engine drag torque in order to keep the engine speed at the constant value ω_{idle}. The fuel consumption is in this case

$$\dot{m}_f = \dot{m}_f(\omega_{idle}, 0). \tag{7}$$

The second case of special interest is when a gear is engaged but no fuel is injected to the engine. This is referred to as *coasting with gear* and the fuel flow is:

$$\dot{m}_f = 0. \tag{8}$$

C. Gear engaged

The flywheel torque is transmitted through the gearbox with a resistive torque in the gearbox T_{gb} such that

$$T_w = \left(T_{fw} - T_{gb}(\omega, T_{fw})\right) \gamma_g \gamma_{fd}. \tag{9}$$

The torque T_{gb} is given as a function of engine speed and flywheel torque found through experiments. It can be approximated by a linear plane such that

$$T_{gb}(\omega, T_{fw}) = k_1 \omega + k_2 |T_{fw}| + k_3 \tag{10}$$

for some constants k_1, k_2 and k_3. The gearbox has 14 gears and four different planes are used to model the losses. The different planes correspond to having high or low range and whether the direct gear is used or not. The engine speed, which is given by the engaged gear and the velocity of the truck, is limited such that

$$\omega_{min} \leq \omega \leq \omega_{max}. \tag{11}$$

For any velocity of the truck, the set of feasible gears is thus given by the gears satisfying (11).

D. Neutral gear

When neutral gear is engaged, the engine speed is constant with the value ω_{idle}. In order to keep this value, $T_{fw} = T_e - T_d = 0$ must hold. Since only a very small torque is transmitted through the gearbox, the gearbox losses are set to $T_{gb} = 0$ when using neutral gear.

E. Gear changes

A transition from one gear to another is modelled to take 1 s, during which no fuel is injected to the engine. The energy loss due to the engine drag torque is calculated using (5). When a gear shift is performed such that the engine speed is increased, i.e., either from a higher to a lower gear or from neutral gear to any gear, the rotational energy in the engine is increased. The energy required to increase the engine speed to the new value is given by the difference in rotational energy

$$\Delta E_\omega = \frac{I_e(\omega_2^2 - \omega_1^2)}{2} \tag{12}$$

for two engine speeds ω_1 and ω_2. The cost in terms of energy needed in order to increase the engine speed can be related to a fuel cost by a constant e such that $m_f e = \Delta E_\omega$. The constant $e = e_{ideal}\gamma_c$ is the specific energy of diesel calculated from the ideal specific energy of diesel e_{ideal} times an average combustion efficiency of the engine γ_c.

III. Problem Formulation

The state variables, control variables and environmental variables are all discretized with steps of $\Delta s = 10$ m. The positions $0, \Delta s, 2\Delta s, ..., (N-1)\Delta s$ define the position of the *stages*. The velocity of the truck is discretized with steps of $\Delta v = 0.1$ m/s. The model (1) is discretized using Euler forward such that

$$x_{k+1} = F_k(x_k, u_k) \tag{13}$$

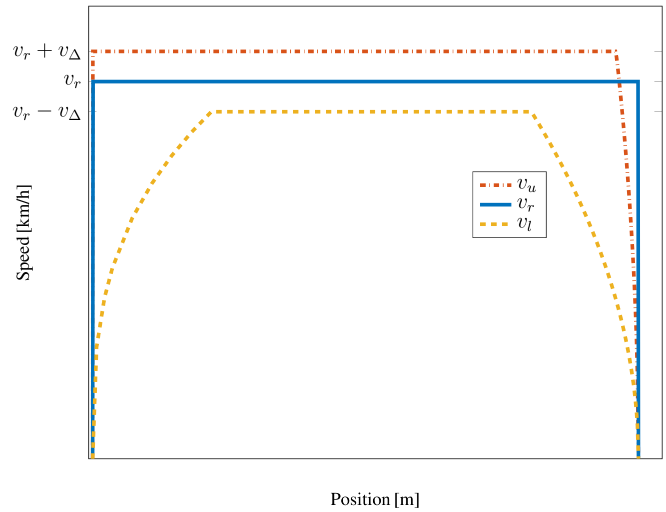


Fig. 3: The driving corridor created from one section of the reference speed trajectory.

where the subscript k indicates the stage and the state variable x_k is given by

$$x_k = \begin{bmatrix} v_k & g_k \end{bmatrix}^T \tag{14}$$

where g_k is the currently engaged gear. The control variable u_k is given by

$$u_k = \begin{bmatrix} T_{e,k} & g_{d,k} & F_{b,k} \end{bmatrix}^T \tag{15}$$

where g_d is the control gear. If the control gear is different from the current gear, i.e., $g_{d,k} \neq g_k$, then a gear change is requested. The velocity is at all positions bounded by the driving corridor such that $v_{l,k} \leq v_k \leq v_{u,k}$, where v_l and v_u are the lower and upper bound of the driving corridor. As discussed in [11], the driving corridor is created from a reference speed trajectory v_r and from real truck operation data. The main idea of using traffic data is to restrict the truck to decelerate in a way that does not deviate too much from a normal way of driving. Two inputs are needed in order to create the driving corridor: the maximum and minimum allowed deviation from the reference speed trajectory v_Δ and the number of standard deviations n_Σ the truck is allowed to deviate from the mean deceleration when v_r is decreased. The creation of the driving corridor for one section of the driving cycle can be seen in Fig. 3. The lower limit v_l is not set based on traffic data during the acceleration phase. Instead, it is set using 60 % of maximum power, which was found empirically to be a good value during simulations.

The objective of the optimization problem in this paper is the minimization of the weighted sum

$$\sum_{k=1}^{N-1} m_{f,k} + \beta t_k \tag{16}$$

where β defines the weighting between fuel and the time t.

IV. Dynamic programming solution

A. Theoretical background

The problem formulated in this paper is solved by using DP introduced in [12]. The main idea is to use the principle of optimality, i.e., the fact that regardless of what the initial state is, the remaining control must be optimal with respect to the states resulting from this control. The optimal cost-to-go $J_k(x_k)$ is defined as the cost for taking the truck from the state x_k at the current stage to the final stage. The transition cost from one stage to the next is denoted by $\zeta_k(x_k, u_k)$. The algorithm becomes:

1) At the last stage N of the driving mission, the velocity v_N is fixed while the gear g_N is only restricted by the requirements for the engine speed (11). The optimal cost-to-go is equal to zero for all feasible states at this stage.
2) For $k = N-1, ..., 1$, the optimal cost-to-go is given by

$$J_k(x_k) = \min_{u_k}\{\zeta_k(x_k, u_k) + J_{k+1}(F_k(x_k, u_k))\}. \quad (17)$$

3) The solution constitutes of the control sequence u_k for $k = 1, ..., N-1$.

The DP algorithm in this paper considers four different types of transitions between stages described in the following subsections.

B. Same gear transitions

For each velocity at the start stage, the set of feasible gears is given by the constraints on the engine speed. Then, for each velocity for which a gear is feasible also at the target state, the required flywheel torque T_{fw} and/or braking force F_b required for this transition is calculated using (13). The optimal cost-to-go and the control signals are given by

$$J_k(x) = \min_{u_k}\{\zeta_k(x_k, u_k) + J_{k+1}(F_k(x_k, u_k))\} \quad (18)$$

where $\zeta_k(x_k, u_k)$ is given by

$$\zeta_k(x, u) = (\dot{m}_{f,k} + \beta)t_k. \quad (19)$$

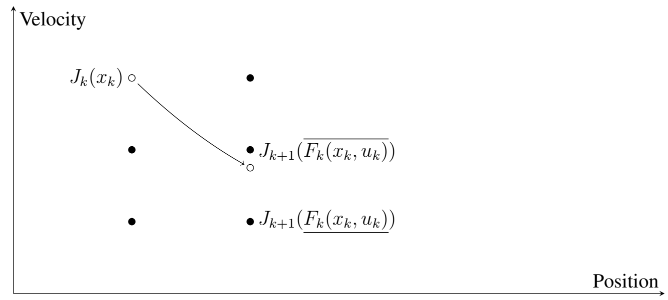
The fuel mass flow $\dot{m}_{f,k}$ is given by (6) and the evolved time is given by

$$t_k = \frac{2\Delta s}{v_k + v_{k+1}}. \quad (20)$$

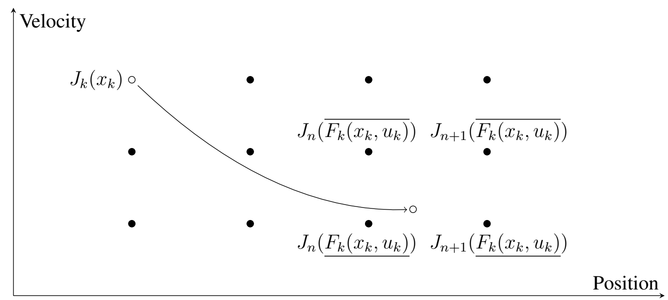
C. Coasting in neutral

A transition to a state where neutral gear is engaged is referred to as coasting in neutral. This is performed during a full interval Δs. The resulting velocity at the next stage is in most cases not equal to any of the beforehand discretized velocities. The cost-to-go at the end of one step of coasting in neutral is therefore given by a linear interpolation of the two surrounding values of the discretized velocities. The control signals are $g_{d,k} = 0$ and $T_{e,k} = T_d(\omega_{idle})$ and the braking force $F_{b,k}$ is only applied in order to avoid overspeeding. The optimal cost-to-go at stage k is given by

$$\begin{aligned} J_k(x) = \min_{u_k \in U_k} &\{\zeta_k(x_k, u_k) \\ &+ \epsilon J_{k+1}(\overline{F_k(x_k, u_k)}) + (1-\epsilon)J_{k+1}(\underline{F_k(x_k, u_k)})\} \end{aligned} \quad (21)$$



(a) Transition using coasting in neutral or coasting with gear.



(b) Transition involving a gear change.

Fig. 4: Transitions from stage k for which interpolation of the cost-to-go and control signals are necessary.

where $\overline{F_k(x_k, u_k)}$ and $\underline{F_k(x_k, u_k)}$ are the state $F_k(x_k, u_k)$ with its velocity rounded to the nearest higher and lower value respectively and ϵ is given by the linear interpolation between the two states, see Fig. 4a. The transition cost $\zeta_k(x_k, u_k)$ is given by (19) with the fuel mass flow given by (7).

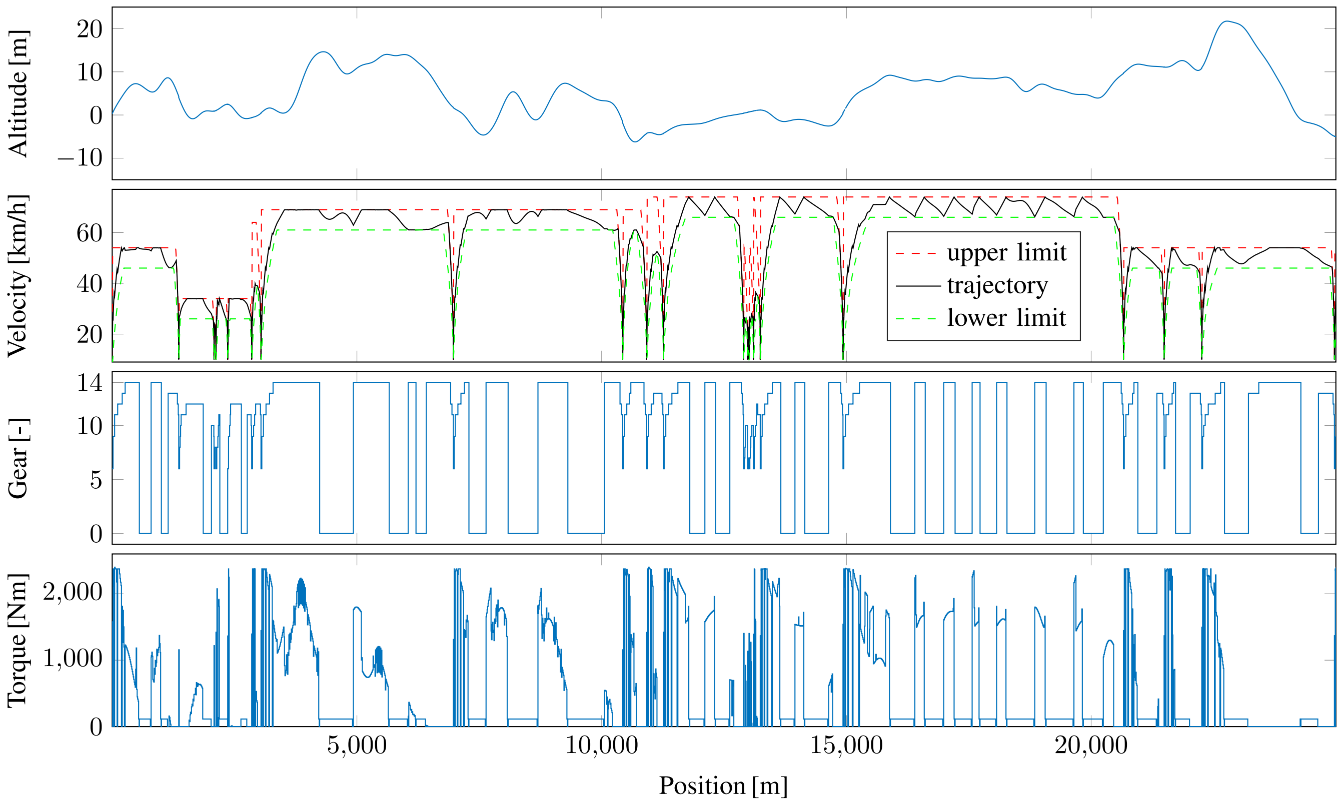
D. Coasting with gear

Transitions using coasting with gear are also performed during a full interval Δs, but with lower end velocity than when coasting in neutral, since $T_{fw} < 0$ in this case. The optimal cost-to-go is calculated for each feasible velocity and gear using (21) and (19) with the fuel mass flow given by (8). The control signals are $g_{d,k} = g_k$, $T_{e,k} = 0$ and the braking force $F_{b,k}$ is only applied in order to avoid overspeeding.

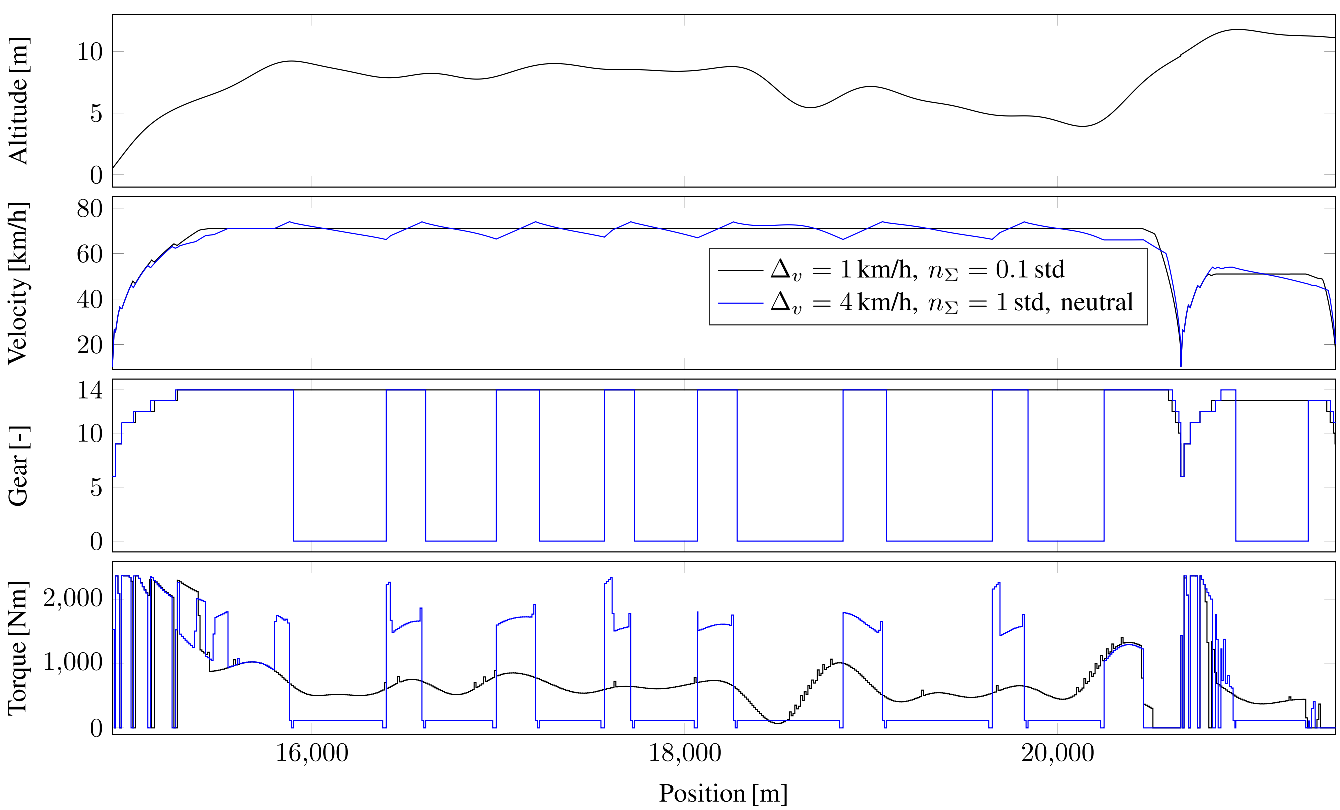
E. Gear changes

A gear shift always starts at a position belonging to a stage, lasts for τ_g, and might therefore end between two stages. During a gear shift, the truck rolls with open clutch, i.e., with zero flywheel torque. If the truck reaches the next stage in less than τ_g, the residual time of the gear change is calculated until two stages are found between which the gear change is completed. The cost-to-go is then given by a bilinear interpolation between the four surrounding points in terms of position and velocity, which can be seen in Fig. 4b. The optimal cost-to-go from a specific state x_k ending between stage n and $n+1$ is given by

$$\begin{aligned} J_k(x) = \min_{u_k}&\{\zeta_k(x_k, u_k) \\ &+ \epsilon_1 J_n(\overline{F_k(x_k, u_k)}) + \epsilon_2 J_n(\underline{F_k(x_k, u_k)}) \\ &+ \epsilon_3 J_{n+1}(\overline{F_k(x_k, u_k)}) + \epsilon_4 J_{n+1}(\underline{F_k(x_k, u_k)})\} \end{aligned} \quad (22)$$



(a) Simulation using $\Delta_v = 4$ km/h and $n_\Sigma = 1$ std.



(b) Trajectory using a narrow corridor compared to using a wider corridor and the ability to coast in neutral gear.

Fig. 5: Simulation results showing altitude, velocity trajectories, gear and engine torque.

where ϵ_i are constants defined by the bilinear interpolation and $\sum_{i=1}^{4} \epsilon_i = 1$. The control gear is any gear not equal to either the current gear nor neutral gear, $T_{e,k} = 0$ and the braking force $F_{b,k}$ is only applied in order to avoid overspeeding. The transition cost to the position where gear change is finished becomes

$$\zeta_k(x_k, u_k) = \sum_{i=k}^{n} (\dot{m}_{f,i} + \beta) t_i + \gamma \Delta E_\omega \tag{23}$$

where t_i is given by (20) for $i = k, ..., n-1$ and t_n is the time consumed at the incomplete last stage such that $\sum_{i=k}^{n} t_i = \tau_g$. The fuel mass flow is zero for an upshift, since the decrease in engine speed is used for overcoming the engine drag torque. When changing from a higher gear or from neutral gear, the fuel mass flow in (23) is set by $\dot{m}_f(\omega_k, 0)$.

F. Solution

The steps in IV-A are applied to IV-B - IV-E. If no feasible solution from a state is found, the cost-to-go is set to infinity. The control vector u_k is stored for each feasible state at each stage. When the backward DP-algorithm is finished, a forward algorithm using the optimal control u_k for the current state is applied. Using the control u_k, the truck might end up in between two discretized velocities during gear changes, coasting in neutral or coasting with gear. If the current state is between discretized states with the same gear, interpolation is used in order to find the control at this state. If the current state is between discretized states with different gears, the control belonging to the nearest one in terms of velocity is chosen.

V. SIMULATION RESULT

Simulations were performed using Matlab on a 66 km driving cycle developed for testing distribution driving. The driving cycle includes an altitude profile and a reference speed trajectory. The cycle contains a total of 53 stops, at which the truck is starting and stopping with a velocity of 10 km/h, since the dynamics during very small velocities are out of scope for this paper. The intention of the simulations is to investigate how the fuel consumption is affected by a wider driving corridor and by the possibility to use neutral gear. In order to fairly compare simulations with different driving corridors, the value of the time penalty parameter β is adjusted such that the simulations have the same trip time. A simulation with a narrow driving corridor $\Delta_v = 1$ km/h and $n_\Sigma = 0.1$ is used as a benchmark. The resulting trajectory using a wider driving corridor with the possibility to use neutral gear can be seen in Fig. 5a for the first 25 km of the cycle. A trajectory from the same simulation is in Fig. 5b shown together with a trajectory from the benchmark solution with the same trip time. The resulting fuel consumption for trajectories with the same trip time, is summarized in Table II.

VI. DISCUSSION

By using the truck model and problem formulation in this paper, the optimal control in terms of gear selection and engine torque can be found using DP. The simulations show potential

TABLE II: Resulting fuel consumption for five different simulations with the same trip time.

Driving corridor	Neutral allowed	Fuel savings
$\Delta_v = 1$ km/h, $n_\Sigma = 0.1$ std	no	0 %
$\Delta_v = 2$ km/h, $n_\Sigma = 1$ std	no	5.0 %
$\Delta_v = 4$ km/h, $n_\Sigma = 1$ std	no	8.0 %
$\Delta_v = 2$ km/h, $n_\Sigma = 1$ std	yes	9.2 %
$\Delta_v = 4$ km/h, $n_\Sigma = 1$ std	yes	12.7 %

fuel savings of up to 12.7 % without increasing the trip time. It can be seen in Fig. 5a that if allowed, coasting in neutral will be used frequently. Even on flat regions, the truck will switch between high torque and coasting in neutral, which is known as Pulse-and-Gliding [8]. Since the truck may only deviate within the driving corridor, the frequency at which neutral gear is engaged depends on the width of the driving corridor. The computation time of the proposed algorithm has not been considered in this paper but will be investigated in future work. This can be done by either taking means to make the algorithm real-time implementable or to use the derived solutions as trajectories for an online controller to track.

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