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3D ROAD GEOMETRY BASED OPTIMAL TRUCK FUEL ECONOMY

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ABSTRACT

This paper investigates the benefit of using a 3D road geometry based optimal powertrain control strategy in reducing the fuel consumption of heavy trucks. The optimal control, which applies a sequential quadratic programming (SQP) method, is designed to predict the optimal truck velocity trajectory, based on the road geometry with the consideration of fuel consumption and travel time. The fuel consumption baseline is developed based on an engineering drive cycle. Computer simulations of a Class 8 truck are conducted with Intermap real 3D road geometry. Simulation results show that the optimal control strategy is able to reduce the fuel consumption with equal or even shorter travel time, when compared to the defined baseline.

NOMENCLATURE

g gear number
h integration step size (m)
m truck mass (kg)
m_f fuel flow rate (g/kW-h)
N engine speed (rpm)
P power (kW)
t travel time (sec)
u throttle position
v longitudinal truck velocity (m/s)
v_{cyc} drive cycle velocity (m/s)
s longitudinal truck position (m)
S prediction horizon (m)

1 INTRODUCTION

Nowadays, heavy trucks consume a high percentage of the US's highway fuel usage, nearly 15-20 percent in 2005. Manufacturers, therefore, are interested in making trucks more and more fuel economic. The major fuel losses of the moving truck are from air drag, rolling resistance, and road slope. Especially, the fuel losses resulting from the road slope can be significant in heavy vehicles such as long haul trucks.

Several studies have been conducted to design predicted powertrain controllers to reduce fuel consumption based on the road information. DaimlerChrysler developed a Predictive Cruise Control (PCC) system to reduce fuel consumption of heavy trucks [1]. PCC calculates the optimal vehicle speed trajectory according to the road information with respect to fuel consumption. PCC can achieve up to 5% fuel reduction for a selected vehicle and a designated road profile. In [2], the author designed the Model Predictive Cruise Control (MPCC) in

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heavy trucks by using road topography. MPCC can determine the throttle input, break levels, and gear selection to reduce fuel consumption. Scania Inc., as stated in [3], developed an Expert Cruise Controller (ECC) using look ahead road information. ECC implements different control strategies for different road section types, and therefore highly relies on the accuracy of the road map. Generally speaking, these prior works focused mainly on the design of road map based optimal cruise controllers to minimize fuel consumption. The comparison baseline for fuel consumption is developed from the operation of a conventional cruise controller, and the road information was generated from GPS measurements.

This work, unlike previous studies, will emphasize two different points. First, the optimal control strategy is designed to perform a real drive cycle but not track a constant cruise speed. The resulting fuel consumption is then compared with the baseline, the normal drive cycle fuel consumption. In order to keep a constant speed on the hilly road, cruise controller might require large throttle change and fuel consumption, which may not be a realistic baseline. On the contrary, by performing a real drive cycle, the optimal control strategy is compared to the real truck driving condition, and therefore its gain of fuel economy will be more realistic. Second, the road maps applied are commercial GIS road geometries. They are part of a commercial data set that will be consistently accurate nationwide - and not generated to support some specific research, as in [1] and [2].

This paper is organized as follows. In Section 2, problem formulation is given, while Section 3 deals with the drive cycle and baseline development. In Section 4, the system modeling and control system design are provided. Section 5 shows some simulation results and analysis. The conclusion and future work are discussed in Section 6.

2 PROBLEM FORMULATION

In order to investigate the possibility and the gain by applying 3D road geometry to minimize the truck fuel consumption, the main objectives of this paper are to: develop a realistic drive cycle and fuel consumption baseline, with respect to the specific road geometry; design an optimal powertrain control strategy to perform the drive cycle, with minimizing fuel consumption; and evaluate the performance of the designed system by various road geometries to obtain more representative results.

3 ROAD GEOMETRY and BASELINE

In this research, the 3D road geometry is provided by Intermap Technologies Corp. , in order to obtain the best truck operations and maximal fuel performance. The fuel consumption baseline for the specific road section and drive cycle should be defined to evaluate the performance of the designed optimal control strategy.

3.1 Road geometry

The road geometry provided by Intermap is a 3D road vector, with accurate longitudinal, x, lateral, y positions and elevation, z. In this work, only the longitudinal road but not the horizontal road curvature is taken into account. Therefore, 2D road slop can be calculated based on the road vector by Equation (1):

$$\phi(k+1) = \frac{z(k+1)-z(k)}{x(k+1)-x(k)} \tag{1}$$

where ϕ is the road slope and k is the sampling point.

3.2 Drive cycle and baseline

A drive cycle constitutes a series of vehicle speeds as a function of time on a specific road section. For the fuel economy research, the definition and selection of the baseline, which is fuel consumption for a normal drive cycle, is critically important. In prior works, ([1]- [3]), the drive cycle is defined as a constant speed, and the fuel consumption baseline is then calculated from a PI cruise controller to perform this drive cycle. This baseline is generic because the cruise control is highly engaged in the real-time truck driving. However, when driving on the hilly road, the cruise controller might require large throttle variation and fuel consumption, which is a bad truck operation case. Thus, with the comparison of this baseline, the gain of fuel economy obtained by the optimal controller may not be realistic.

Consequently, different drive cycles should be taken into account. The current practices in drive cycle development are mainly segment-splicing, Monte Carlo simulation, and an “engineering” approach. The first approach is real-driving data based, and the second approach is simulated from a realistic driving behavior model. The third approach, on the other hand, is defined by a designer to implement some truck feasibility testing [4]. In this research, the third method is applied for the drive cycle development, and the first method will be applied in future work. If the drive cycle developed by the first method is applied, the function of the optimal control strategy will then be compared to the real truck driving condition, or even better, experienced drivers’ behavior, and therefore, the gain of the fuel economy is more meaningful.

4 TRUCK MODEL and CONTROL SYSTEM

Two different control strategies are designed to perform the developed drive cycle. An optimal control, called ‘Optimal Controlled truck’, is designed to predict the optimal truck velocity along the drive cycle based on the road geometry with the consideration of fuel consumption and travel time. Additionally, a sliding mode controller is designed to accurately perform the drive cycle, and resulting fuel consumption is defined as the baseline. This system is called ‘Normal truck’ in this work.

4.1 Heavy truck modeling

A simple Class 8 truck longitudinal model is considered, which includes the engine, driveline, wheel, and truck dynamics. A tire model is not used, and therefore a no-slip condition is assumed. The developed truck model was validated by TruckSim 5.0 in previous work [5].

Longitudinal dynamics To describe the longitudinal motion of a vehicle, the dynamics are derived from the loads on the vehicle. Longitudinal vehicle dynamics typically include many losses such as rolling resistance, air drag, and road grade or slope. The developed model has one degree of freedom and was derived using the equation of motion.

$$m\frac{dv}{dt} = F_w - F_s - F_{rr} - F_a \tag{2}$$

where F_w is wheel friction force, $F_s = mg\sin\phi$ is longitudinal force due to road grade, F_{rr} is rolling resistance force, and F_a is air drag force.

Engine and fuel consumption For simplicity, the engine model is designed based on a rectangular engine map, as shown in Figure 1. This map shows a steady-state relation between the current engine speed and the maximum engine torque. Interpolated from this map, the maximum engine torque can be formulated as a function of the engine speed. Meanwhile, with the introduction of the normalized throttle position, the desired engine torque can be further calculated from:

$$T_m = f(N) \tag{3}$$
$$T_e = T_m u \tag{4}$$

where T_m and T_e are maximum and desired engine torque, u is control input (throttle position), and $f()$ is a function of engine speed.

The truck fuel consumption is calculated depending on a Brake Specific Fuel Consumption (BSFC) map as shown in Figure 2, where the fuel flow rate δ can be interpolated as a function of engine speed and engine power, as written in Equation (5).

$$\delta = h(T_e, P) \tag{5}$$
$$\frac{dm_f}{dt} = \frac{P\delta}{3600} \tag{6}$$

where $h()$ is a function of engine speed and power, P is engine power, and $\frac{dm_f}{dt}$ is fuel consumption time rate with unit g/sec . It should be noted that when the engine output power is zero or driving down-hill, the BSFC map is set to have zero fuel consumption.

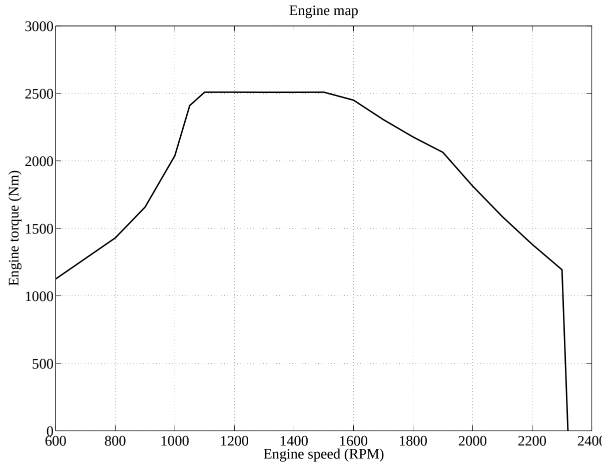

Figure 1. Engine map

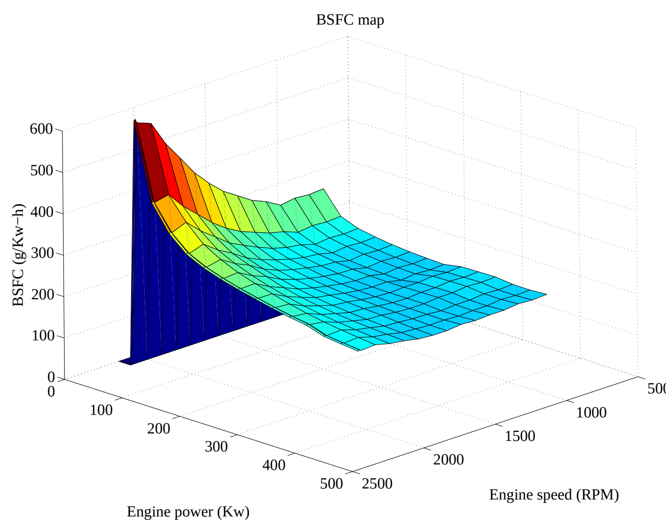

Figure 2. Engine BSFC map

Driveline The driveline is assumed stiff such that the engine rotational rate can be calculated by:

$$J_e\dot{\omega}_e = T_e - T_c \tag{7}$$

where T_c is the external load from clutch. If the clutch is stiff and the transmission, final drive, and wheel are considered together with the longitudinal forces as shown in Equation (2), the complete truck longitudinal model can be written as:

$$\frac{dv}{dt} = \frac{r}{J_w + mr^2 + \eta_d n_d^2 \eta_t n_t^2 J_e} \cdot (\eta_d n_d \eta_t n_t T_e - F_s r - F_{rr} r - F_a r) \quad (8)$$

where J_w and J_e are wheel and engine inertia (kg-m^2), and η_t, η_d, n_t, and n_d are efficiency and ratio of transmission and final drive, respectively. In this work, the braking is considered as 'negative' throttle position and equivalent negative engine toque, which will not impact the fuel consumption calcualtion, as BSFC map defines the nonzero fuel consumption only for the positive torque. Thus, the throttle is normalized within [-1, 1].

Because stated in [1] and [2], the road map is position dependent rather than time dependent, it is necessary to change Equation (5) and (8) to be differentiated with respect to position rather than time by substituting:

$$dt = \frac{1}{v} ds \quad (9)$$

The complete truck model is shown in Figure 3

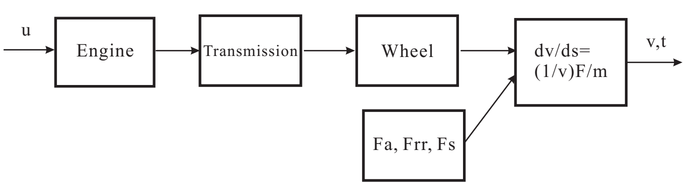

Figure 3. Truck longitudinal model

4.2 Optimal control strategy

A complete optimal control system for real-time implementation should include a map matching system to utilize the truck current position and the road information ahead, based on measurements from GPS, Inertial Navigation System Sensor (INS) and the 3D road geometry. In this work, the position data are presumed, and therefore, the road geometry information is directly available and accurate to the optimal control action. Truck parameters are assumed constant and unchanged as well.

An optimal powertrain control strategy is designed to predict the optimal truck velocity based on the road geometry with respect to fuel consumption and travel time. The control function is achieved by commanding proper throttle position, [-1,1] and transmission gear number $g \subset [1,\ 10]$. As mentioned previously, the negative throttle represents the function of braking.

The fuel minimization problem can be constructed to find the constrained minimum of a fuel consumption function. This problem is generally referred to as constrained nonlinear optimization. The goal is to find a sequence of control inputs, throttle and gear selection, to minimize the cost function. The cost function is the sum of fuel consumption and travel time over the entire road section. Additionally, an extra term J_{gear} is added to penalize frequent gear change. The cost function is shown below:

$$\begin{aligned} J_{total} &= J_{fuel} + J_{time} + J_{gear} \\ J_{fuel} &= Q \Sigma_{k=0}^{S-1} h \frac{dm_f}{ds} \\ J_{time} &= R \Sigma_{k=0}^{S-1} \frac{h}{v} \\ J_{gear} &= T \Sigma_{k=0}^{S-1} |g(k+1) - g(k)| \end{aligned} \quad (10)$$

where h is a constant integration step size, S is the prediction horizon, and $Q/R/T$ are weighting factors. Currently, these factors are determined by some rules: Q is chosen large to have fuel consumption weighted more than the other two; R is not set too small, which ensures a short travel time; T is selected based on the maximum gear change frequency. Additionally, the initial and final truck velocities are set to follow the drive cycle. The upper and lower bounds of truck velocity, throttle position, and gear number can be implemented into the optimization problem as well. Note that the equality constraint is the discrete time dynamic model as described in Equation (8) as position dependent. The complete optimization model is shown in Figure 4

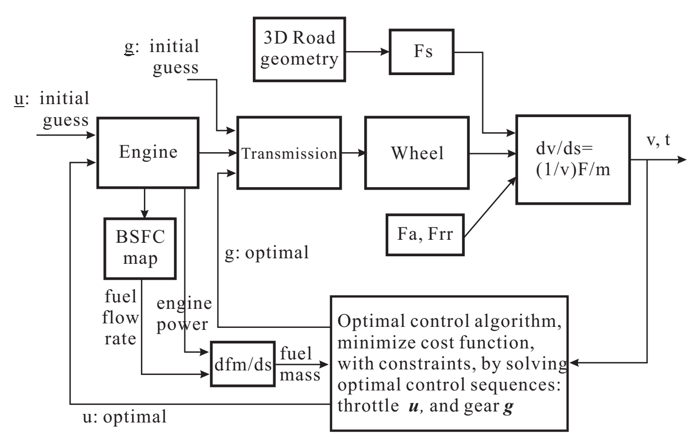

Figure 4. Fuel optimization problem model

The above process can be implemented and solved by the MATLAB function 'fmincon' with the collocation method, which uses sequential quadratic programming (SQP). This algorithm solves a quadratic programming (QP) subproblem at each iteration, which is based on a constrained direction determination and line search optimization with Quasi-Newton directions method.

In the implementation, the entire road map is separated into a series of sections with equal length, and the optimal solution is then obtained by iterative road section updates and calculations. The prediction horizon is selected as 4000m, and the integration step size is 40m. Thus, 'fmincon' solves for 100 throttle and gear positions, and calculates fuel consumption and travel time every 4000m, which takes only a short calculation time. By comparison, it can be found that with the selection of a larger horizon and smaller integration step, the gain of fuel economy is slightly improved, but the computation time is largely increased.

4.3 Sliding mode control

To accurately perform the drive cycle and generate the fuel consumption baseline, a sliding mode controller is used. The sliding surface can be simply chosen as $S_s = v - v_{cyc}$, and the desired throttle position is calculated by converting the desired acceleration to the desired engine torque [6]:

$$u_{des} = \frac{T_e}{T_m} - K(v - v_{cyc}) \tag{11}$$

where v_{cyc} is the drive cycle velocity. The gear selection is determined based on the engine speed, current gear position, and the throttle position.

5 SIMULATION RESULTS and ANALYSIS

The developed truck model and control systems are implemented into MATLAB and SIMULINK. The simulation results are shown and analyzed to evaluate the performance of the designed road geometry based optimal powertrain control strategy.

5.1 Simulation setup

The simulation setup, including selection of road profile, drive cycle, and truck parameters, is described below.

Road geometry Intermap's 3D road geometry acquired in California is shown in Figure 5, where route number 2 and 3 will be used in this research. The details of these two road profiles are provided in Table 1, where the mean, maximum, and minimal road slopes are listed as percentage. The σ in this table represents the standard deviation. Road geometries longer than 100km will be taken and applied in future research.

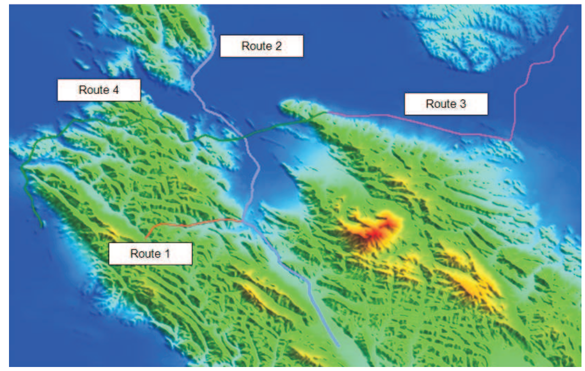

Figure 5. Overview of Intemap's road geometry

Table 1. Intermap road profiles analysis

route	length (m)	slope(%) mean	max	min	σ
2	48000	0.29	4.72	-3.73	1.32
3	36000	-0.21	2.96	-4.33	1.06

Drive cycle An engineering drive cycle is also developed for this research. It is based on a truck fuel consumption and emission test, which specifies the desired distance for the truck to accelerate from a starting to an ending speed, over any road grade, and then calculates the required fuel. For this research, the drive cycle is defined as the truck accelerates from the starting speed, 24.4m/s (88km/h), to the ending speed, 33.3m/s (120km/h), over routes 2 and 3, which are shown in Figure 6.

Truck parameters Important truck parameters, used in the simulation, are listed in Table 2.

Table 2. Class 8 truck parameters

mass (kg)	gears	J_e (kgm^2)	J_w	r (m)	c_{rr}
31752	10	3.95	277.6	0.508	0.007

5.2 Results and analysis

The main advantage of the simulation is that the designed system is tested with two different real road geometries. Thus,

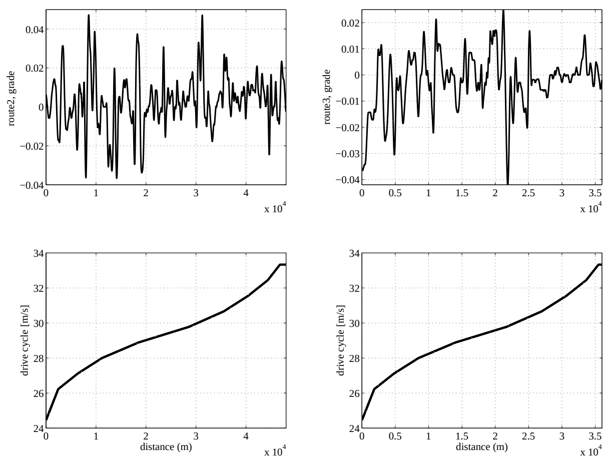

Figure 6. Road slope and drive cycle, routes 2 and 3

the simulation results are more realistic in evaluating the system performance. The comparison of fuel consumption in liter (F) and travel time (T) from the optimal controlled truck (OC) and the normal truck baseline (NO) is shown in Table 3, where the positive percentage value represents savings.

Table 3. Comparison of fuel consumption and travel time

Route	F_{NO}	F_{OC}	Diff(%)	T_{NO}(s)	T_{OC}	Diff(%)
2	22.32	21.68	2.87	1633	1646	-0.79
3	13.26	12.68	4.30	1215	1210	0.41

Route 2 The comparison of velocity, throttle, and gear position from Normal and Optimal Controlled trucks is shown in Figure 7, where the road elevation is given on the top plot. It can be seen that both systems perform the drive cycle, where the Normal truck performs accurately and the Optimal Controlled truck has varied speed along the drive cycle. The maximum variations are within a range of [-2, 2]m/s. It can be calculated based on Tables 1 and 3, the average speed is around 29m/s both for Routes 2 and 3. As stated in [7], trucks generally increase speed by up to about 5 percent on downgrades and decrease speed by 7 percent or more on upgrades as compared to their operation on level. Thus, the 2m/s speed variation, which is around 6 percent of the average truck speed, would be acceptable for real world driving, and comfortable for truck drivers. Meanwhile, as can be seen in Figure 7, the Optimal Controlled truck has smoother throttle change as well as less negative throttle positions than the Normal truck, which shows a good truck operation situation and reduces the need of frequent braking on the hilly road.

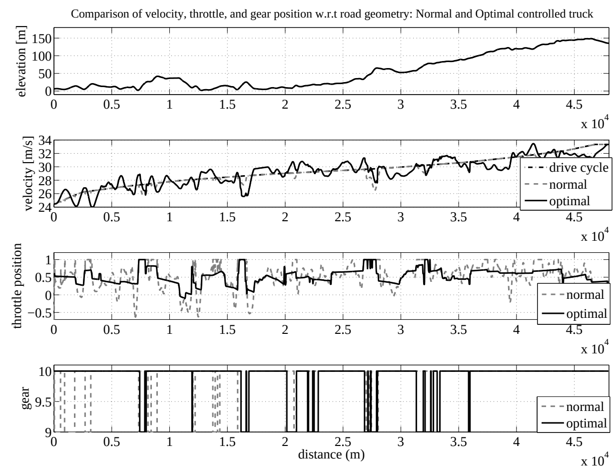

Figure 7. Comparison of velocity, throttle, and gear position: Normal vs. Optimal, Route 2

Figure 8 shows a section of the entire Route 2 driving, which illustrates the following basic functions of the optimal control strategy:

- Decelerate the truck before a sag curve and then gain the velocity by the potential energy on the sag curve, which can be seen at 1.22-1.23e4m, where the throttle position is reduced in front of a sag curve;
- Accelerate the truck before a crest curve to reduce the speed loss and the need for full throttle on the crest, which can be seen at 1.19-1.22e4m. The throttle position is kept large before but not throughout the crest curve, where the Normal truck keeps full throttle;
- Down-shift early than Normal truck to obtain the maximum torque to reduce the speed loss before climbing a large crest, which can be seen at 1.19-1.22e4m;
- Up-shift early to reduce engine speed before coasting a large sag curve.

Furthermore, the distribution of fuel consumption position rate (g/m) for two control strategies over the driving cycle are shown in Figure 9. The statistical analysis shows, by varying the throttle and gear positions, the optimal control strategy keeps 91 percent of all travel points distributed over the range [0, 0.6]g/m, where Normal truck has 83 percent distribution.

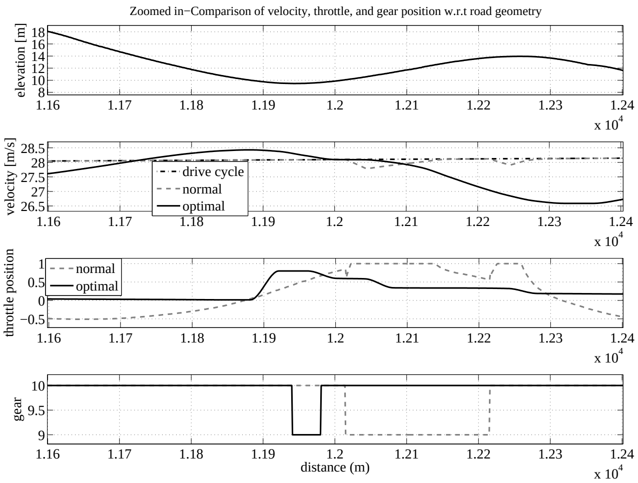

Figure 8. Zoomed-in: Comparison of velocity, throttle, and gear position, Route 2

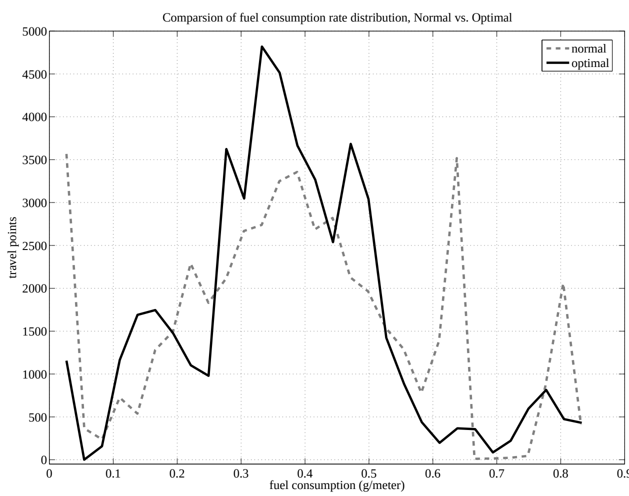

Figure 9. Comparison of fuel consumption position rate distribution: Normal vs. Optimal, Route 2

Route 3 By testing the designed control system on another real road geometry, Route 3, a good performance is obtained as well. The comparison of velocity, throttle, and gear position is shown in Figure 10, where the change of throttle is smooth and the need for negative throttle is small.

A section of Route 3 is shown in Figure 11. Here, the optimal control functions can again be identified, i. e. decelerating before a sag at 1.95e4m, accelerating in front of a crest at 1.98-

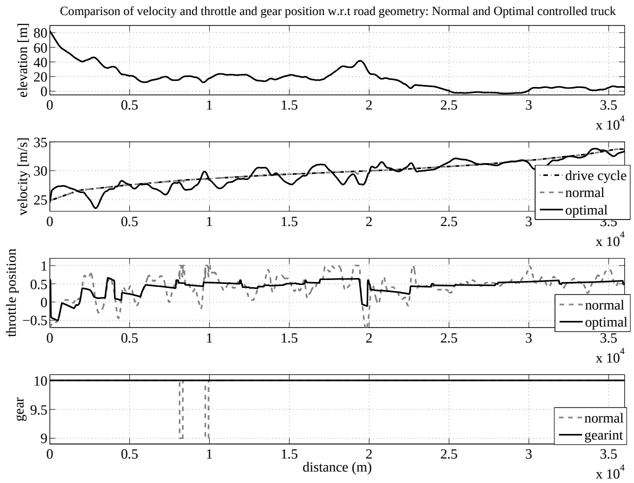

Figure 10. Comparison of velocity, throttle, and gear position: Normal vs. Optimal, Route 3

2.00e4m, etc. The comparison of fueling distribution is in Figure 12. It can be calculated that the optimal control strategy keeps 97 percent of travel points distributed over the range [0, 0.5]g/m, where Normal truck has 86 percent distribution.

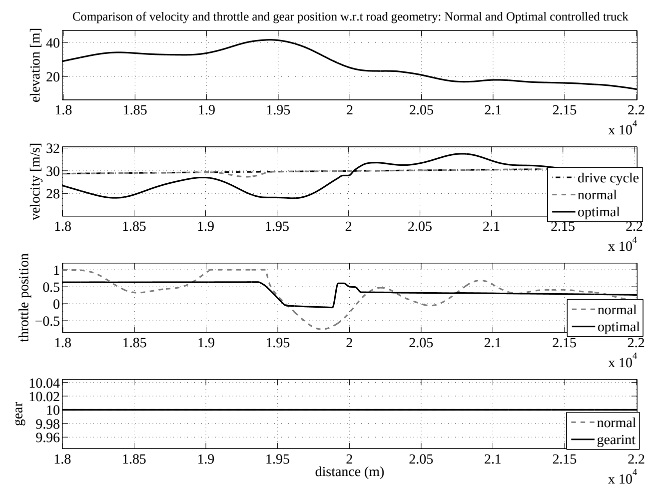

Figure 11. Zoomed-in: Comparison of velocity, throttle, and gear position, Route 3

Moreover, for the Route 3 test, a more promising point can be observed. As shown in Table 3, there is a travel time reduc-

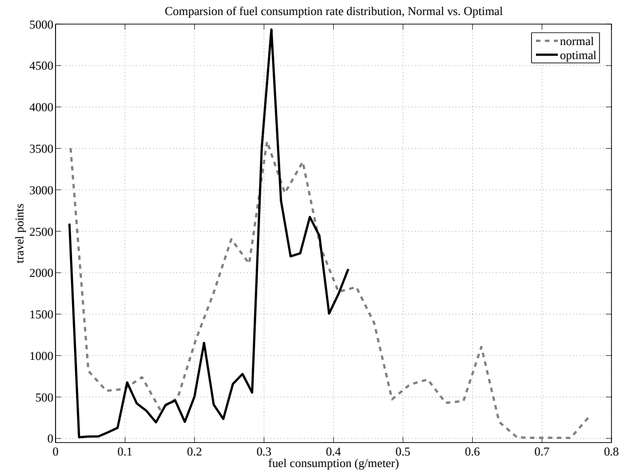

Figure 12. Comparison of fuel consumption position rate distribution: Normal vs. Optimal, Route 3

tion corresponding to the saved fuel, by the function of the optimal control strategy. It is because Route 3 is a down-mountain terrain, which has negative mean road slope, as shown in Table 1. Thus, it can be expected that by applying the road geometry on a down-mountain route, it is possible not only to reduce fuel consumption but also travel time, because the faster speed can always be gained from the sag curves, if their locations and grades are accurately provided by the 3D road geometry.

6 CONCLUSION and FUTURE WORK

In this work, a 3D road geometry based optimal powertrain control strategy is designed. The fuel consumption baseline is developed based on an engineering drive cycle but not a constant speed. The optimal control strategy is designed to minimize fuel consumption and travel time based on the road geometry. Simulations are conducted with commercial GIS road geometries, and the results show that the designed control strategy is able to reduce fuel consumption with equal or even shorter travel time, when compared to the developed baseline.

Future work would consider developing a real-data based drive cycle and using the corresponding fuel consumption as the baseline. By doing this, the designed optimal control strategy is compared to experienced drivers' behavior and the fuel economy result is more meaningful. Analyzing the influence from the accuracy of road geometry to the gain of fuel economy will also be performed in future research. It leads to minimum map accuracy requirements necessary for using the optimal control strategy to gain fuel benefits. Additionally, the influence from traffic conditions on the control and system performance might be investigated in future work.

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