CONTROLLING GEAR ENGAGEMENT AND DISENGAGEMENT ON HEAVY TRUCKS FOR MINIMIZATION OF FUEL CONSUMPTION

Anders Fröberg [*] Lars Nielsen [*]
Lars-Gunnar Hedström [**] Magnus Pettersson [**]

[*] *Linköpings Universitet, SE-581 83 Linköping, Sweden*
email: froberg@isy.liu.se, lars@isy.liu.se
[**] *Scania, SE-151 87 Södertälje, Sweden*

Abstract: There is a potential to save fuel for heavy trucks by storing kinetic energy in the vehicle when driving downhill, because the speed adds kinetic energy to the vehicle which can be used after the downhill slope to propell the vehicle. This behavior can be even more utilized by disengaging the gear to reduce the friction in the driveline and thus increase the speed even more. Two different control strategies to choose when to disengage the gear is presented: One that uses instantaneous inclination and one predictive control scheme that uses look ahead information of the road topology. Simulation results show that gear disengagement in downhills can reduce the fuel consumption about 3%.

Keywords: Driveline Control, Dynamic Programming, Model Predictive Control.

1. INTRODUCTION

In order to save fuel it can be beneficial to increase speed when driving downhill to build up kinetic energy that can be used driving uphill. Typical cruise controllers used in heavy trucks allow the speed to vary between specified limits such that the speed is high in hollows and low on crests. If the slope is high when driving downhill the fuel injection can be cut off and no fuel is consumed. However, for small slopes the engine friction can be so high such that the speed is decreased when going into fuel cut-off mode. In these cases some fuel has to be injected to overcome the engine friction. Then it can be beneficial to disengage the gear so that the powertrain friction is reduced and the speed can be increased or maintained with only idle fuel flow. Further, if the road profile ahead is known, using e.g. GPS or collected data, then further improvements can be done. Here dynamic programming is used to make the trade off between going into fuel cut off and disengaging the gear. In this paper, two strategies will be developed, simulated, and evaluated to explore the potential fuel savings.

2. TRUCK MODEL

The truck is modeled with standard equations for a stiff driveline (Kiencke and Nielsen, 2000; Pacejka, 2002) as summarized here. See Section 6 for notations.

The dynamics of the engine inertia is modeled as

$$J_e\dot{\omega}_e = T_e - T_t \tag{1}$$

where T_e is the engine torque, which includes negative values representing e.g. negative torque during fuel cut off or if present an exhaust brake. Transmission and final drive are modeled as stiff rotational components with constant efficiencies.

$$\omega_e = i_t i_f \omega_w \tag{2}$$

$$T_t i_t \eta_t i_f \eta_f = T_w \tag{3}$$

The wheels are modeled as rolling wheels with brakes

$$J_w \dot{\omega}_w = T_w - F r_w - T_b \quad (4)$$

$$v = r_w \omega_w \quad (5)$$

The vehicle motion is described by

$$F = m\dot{v} + F_{air} + F_r + mg\sin(\alpha) \quad (6)$$

where the air and rolling resistance is

$$F_{air} = \frac{1}{2} c_d A \rho v^2 \quad (7)$$

$$F_r = mgc_r\cos(\alpha) \quad (8)$$

Negative values of α indicates a downhill slope and positive values indicates an uphill slope. If Equations (1)-(8) are combined the result is

$$c_1\dot{v} = c_2T_e + c_3T_b + c_4v^2 + c_5\cos(\alpha) + c_6\sin(\alpha) \quad (9)$$

where c_i are lumped model parameters.

The control input to the engine is the injected amount of fuel per engine stroke δ. The resulting engine torque is mapped as a function of δ and engine speed ω_e.

$$T_e = T_e(\delta, \omega_e) \quad (10)$$

The fuel consumption is computed as

$$m_f(\delta(t), \omega_e(t)) = \int \delta(t) \frac{\omega_e(t)}{2\pi} \frac{n_{cyl}}{n_r}\, dt \quad (11)$$

3. CONTROL STRATEGIES

Cruise controllers in heavy trucks normally allow the speed to increase some above the setpoint when driving downhill (Sandberg, 2001). If the speed increases even though the engine does not deliver any torque, the brakes are not applied until a speed limit defined by the cruise controller is reached. The typical speed interval allowed is 5-10 km/h. In this work, the cruise controller is implemented as two PI-controllers, one controlling the fueling and one controlling the brakes.

Two strategies will be developed in the following: one instantaneous strategy in Section 3.1, and one strategy with look ahead in Sections 3.2-3.5.

3.1 Using Instantaneous Inclination, II-strategy

As mentioned above, there are possibilities to enhance the cruise controller to save fuel. For example a gyro, an accelerometer, or a GPS and 3D map can be used to obtain information about the inclination. This instantaneous inclination can be utilized, and the rationale behind the algorithm below is as follows: Consider driving downhill with the gear engaged in such a small slope that the engine has to deliver some torque for the vehicle to maintain speed. In such a slope it can be possible to disengage the gear, and thereby lowering the driveline friction sufficiently much, such that the speed can be maintained or even increased. The increase in kinetic energy that is stored in the vehicle leads to lower fueling some distance after the downhill slope and thereby the overall fuel consumption can be reduced. When the gear is disengaged the engine has to be run in idle mode to deliver power supply to auxiliary systems such as power steering, and hence consumes a certain amount of fuel. In downhills with inclination so high that the engine does not has to deliver any torque to maintain speed it is always beneficial to go into fuel cut-off mode.

Following this idea, the model presented in Section 2 is used to derive the inclination angles for when it is beneficial to disengage the gear. When the gear is disengaged it is seen from Equation (9), setting $T_e = 0$, $T_b = 0$, that for inclination angles

$$\tilde{\beta} \in \{\tilde{\beta} : c_4v^2 + c_5\cos(\tilde{\beta}) + c_6\sin(\tilde{\beta}) \geq 0\} \quad (12)$$

the speed will be maintained or increased. The boundary for the set (12) (using equality in (12), then becomes

$$\beta = \arcsin\left(\frac{c_4v^2}{\sqrt{c_5^2 + c_6^2}}\right) - \arctan\frac{c_5}{c_6} \quad (13)$$

On the other hand, if the inclination is too high it was earlier stated that fuel cut off was beneficial. The following model for the engine friction when it is being dragged

$$T_{ed} = d_1\omega_e + d_2 \quad (14)$$

with both $d_i < 0$, is used. Together with Equation (9) it is seen that for inclination angles

$$\tilde{\gamma} \in \{\tilde{\gamma} : c_2(d_1\omega_e + d_2) + c_4v^2 + c_5\cos(\tilde{\gamma}) + c_6\sin(\tilde{\gamma}) \geq 0\} \quad (15)$$

the speed will be maintained or increased even when the gear is engaged. The boundary for the set (15) can be expressed as

$$\gamma = \arcsin\left(\frac{c_2(d_1\omega_e + d_2) + c_4v^2}{\sqrt{c_5^2 + c_6^2}}\right) - \arctan\left(\frac{c_5}{c_6}\right) \quad (16)$$

3.2 Lookahead

As stated above it is possible to have knowledge about the upcoming road profile. This can be used to make a more intelligent choice, than the method described in Section 3.1, on when to disengage the gear. To use the extra information about the upcoming road profile and find the optimal control strategy a model predictive control scheme is used (Back *et al.*, 2004; Terwen *et al.*, 2004).

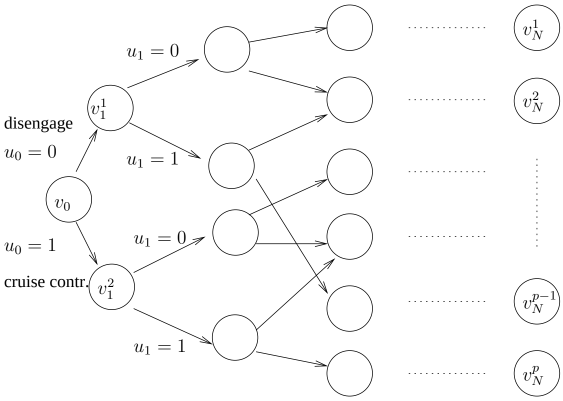

Fig. 1. An example of a dynamic programming transition graph for a prediction horizon of N samples. From each state there are two choices, either disengage the gear or use the cruise controller over the next sample interval. The cost for each transition is the amount of fuel that is needed to go between the corresponding states, i.e. change the speed from v_i^l to v_{i+1}^m on the length of one sample interval.

3.3 Formulation of the optimization problem

Since the altitude information of the road profile is given as a function of position, the model (9) is reformulated with a change of variables from time to position according to

$$\frac{dv}{dt} = \frac{dv}{ds}\frac{ds}{dt} = v\frac{dv}{ds} \tag{17}$$

which introduced in (9) gives

$$c_1 \frac{dv}{ds} = \frac{1}{v}(c_2 T_e + c_3 T_b + c_4 v^2 + c_5 \cos(\alpha) + c_6 \sin(\alpha)) \tag{18}$$

Using this model the choices of whether to disengage the gear or not can be represented by a transition graph as depicted in Figure 1. The sample distance Δ was chosen to the same as the distance between the samples of the altitude. The cost for a transition between state v_k to state v_{k+1} is computed as

$$g_k(v_k, (\delta_k, u_k)) = \begin{cases} m_f(\delta_k, \omega_e), & u_k = 1 \\ m_{f,idle} & u_k = 0 \end{cases} \tag{19}$$

where $u_k = 1$ denotes that the gear is engaged and $u_k = 0$ denotes that the gear is disengaged. δ_k and ω_e is computed from a simulation of the model using the standard cruise controller, and m_f is computed from Equation (11). The fuel consumption at idle is $m_{f,idle}$.

The following optimization problem is general, but since the aim is dynamic programming the notation from (Bertsekas, 2000) is used. Let $\mu_k(v_k) = (\delta_k, u_k)$. Consider the class of all admissible control laws

$$\pi = \{\mu_0, \ldots, \mu_{N-1}\} \tag{20}$$

that maps states v_k into controls. Given an initial state v_0 and an admissible control law, the states v_k are defined by the equation

$$v_{k+1} = f_k(v_k, \mu_k(v_k)) \tag{21}$$

where $f_k(\cdot)$ is defined by a discrete approximation method of (18), e.g. Eulers method or a Runge-Kutta method, see (Hairer *et al.*, 2000). If the cost for an end state is $g_N(v_N)$ the total cost for π starting at v_0 is

$$J_\pi(v_0) = g_N(v_N) + \sum_{k=0}^{N-1} g_k(v_k, \mu_k(v_k)) \tag{22}$$

The optimal control strategy to drive the distance corresponding to N samples is to find the way, π^*, through the transition graph with the lowest cost, which is given by

$$J_{\pi^*}(v_0) = \min_\pi J_\pi(v_0) \tag{23}$$

3.4 Design Considerations

The design criterion (22) is defined by the fuel consumption (19), but the final cost $g_N(v_N)$ remains to be designed. Sometimes this may need consideration. If all end states are assigned the same cost it is in most cases optimal to end up in the state with lowest velocity, because the fuel required to reach that state is lower than for all other states (of course except if braking is considered). However, if the end state is in a downhill slope, it can be beneficial for the total cost of the whole driving scenario if the end state for the current optimization has a higher velocity. The fuel consumption for reaching such an end state has to be compared to the velocity of that state in some way, and here the idea is that the final states are assigned a cost corresponding to a fuel equivalent of the higher kinetic energy they corresponds to. This comparison is made with an efficiency model of the truck.

3.5 Determining the Reachable State Space

Using dynamic programming to optimize (22) means backward calculation in a transition graph, and here this graph has the following characteristics. In the problem considered, there are natural bounds on the velocity. Too high speed can not be accepted for, e.g., safety or regulation reasons. The driver will probably not allow the speed to decrease below a certain limit. To find the upper limit for the speed the truck model is simulated with maximum fueling which results in a speed sequence $(v_{sf}^0, \ldots, v_{sf}^N)$. The upper limit is then taken as $\min(v_{sf}, v_{max})$, where v_{max} is the highest speed allowed. As lower limit $v_{min} = \min(v_c, v_{set})$ is chosen, where $(v_c^0, \ldots, v_c^N)$ is the speed sequence obtained with a standard cruise controller and v_{set} is a set-point speed chosen by the driver.

Even though the state space is restricted with an upper and lower bound one can have infinitely many states in between. Therefor an approximation is done such that if two states are very close to each other, $|v_1 - v_2| < \varepsilon$ for some small positive value ε, they are approximated to the same state. Following this procedure the problem grows linearly with prediction horizon, and the maximum number of states in each stage in the transition graph in Figure 1 is $(v_{max} - v_{min})/\varepsilon$.

4. SIMULATION RESULTS

To evaluate the control strategies described in Section 3 the truck model presented in Section 2 has been simulated with different road topologies. Both constructed test topologies as well as actual topologies from the highway E4 outside Linköping, Sweden have been used, see Figures 2-7. The constructed road topologies have been chosen such that they should show interesting properties of, and differences between, the two proposed control strategies and an ordinary cruise controller.

The standard cruise controller and the controller using instantaneous inclination, II-strategy, were sampled with 10 Hz. The look ahead controller was sampled each 25 meters, and the prediction horizon was 10 samples corresponding to 250 meters.

The gap between β and γ defined by Equations (13) and (16) is narrow, typically in the order of 0.1 degree for trucks weighing around 20-60 tons and speeds around 85 km/h. In Figure 2 the inclination is between β and γ. The II-strategy disengages the gear in the downhill which can be seen in the second subplot from above, where 0 means disengaged gear. There is thus two periods around position $s = 600$ and $s = 800$ where this happens. Thereby the speed increases, as can be seen in the lowest subplot as the solid line. The behavior of the standard cruise controller is seen in the same subplot as dashed line. It does not disengage so the speed is in these periods (around $s = 600$ and $s = 800$) decreasing instead of increasing as for the II-strategy. In subplot 3 from above the fuel savings, about 1.5 %, for the II-strategy, is seen as the difference between the curves in the same period ($s = 600$, $s = 800$).

In Figure 3 the inclination is outside the operating range of the II-strategy and it gives the same result as the standard cruise controller. If the plot of gear engagement/disengagement is studied closely it is seen that the II-strategy does in fact disengage the gear at 4 positions even though

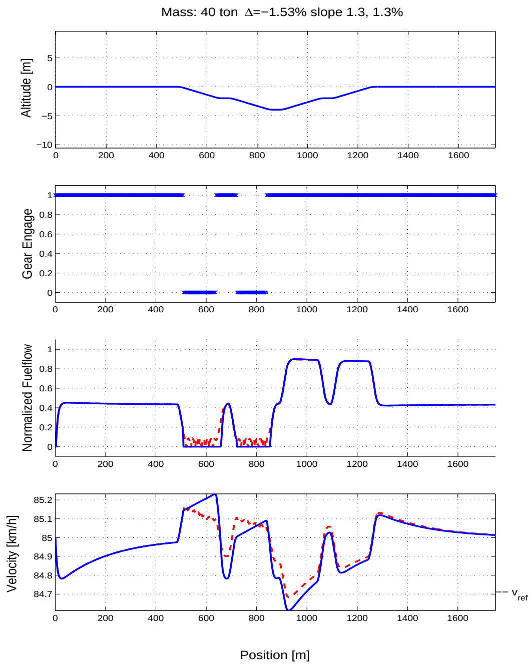

Fig. 2. Simulations of a 40 ton truck. The dashed line represents a standard cruise controller and the solid line the II-strategy. The inclination in the slopes is $\mp 1.3\%$ and the fuel saving is 1.53%.

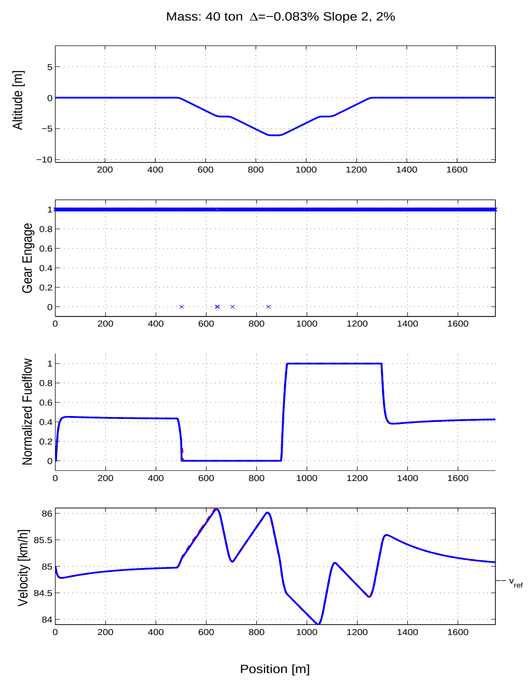

Fig. 3. Simulations of a 40 ton truck. The dashed line represents a standard cruise controller and the solid line the II-strategy. The inclination in the slopes is $\mp 2\%$ and the fuel saving is 0.083%.

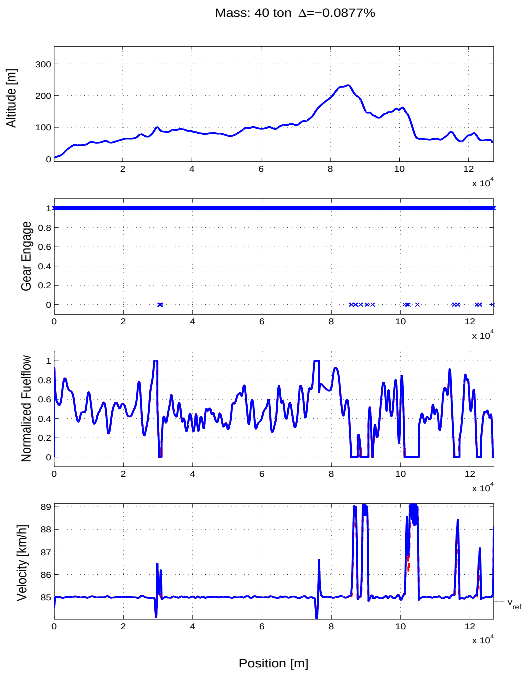

Fig. 4. Simulations of a 40 ton truck. The dashed line represents a standard cruise controller and the solid line the II-strategy. The road data comes from the highway E4 outside Linköping and the fuel saving is 0.088%.

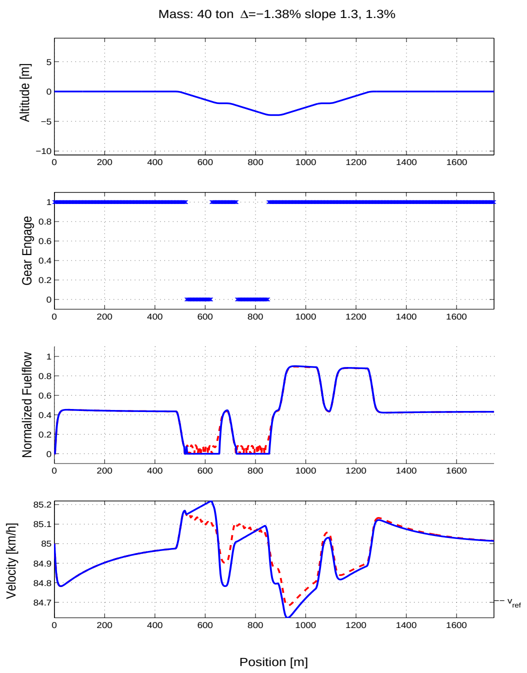

Fig. 5. Simulations of a 40 ton truck. The dashed line represents a standard cruise controller and the solid line the look ahead strategy. The inclination in the slopes is $\mp 1.3\%$ and the fuel saving is 1.38%.

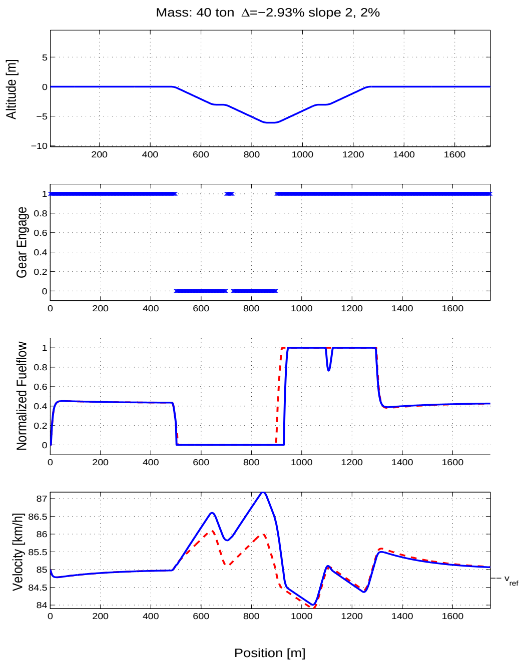

Fig. 6. Simulations of a 40 ton truck. The dashed line represents a standard cruise controller and the solid line the look ahead strategy. The inclination in the slopes is $\mp 2\%$ and the fuel saving is 2.93%.

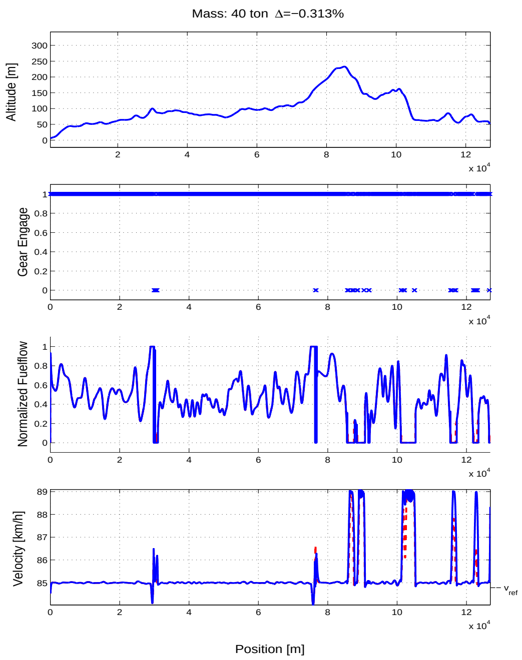

Fig. 7. Simulations of a 40 ton truck. The dashed line represents a standard cruise controller and the solid line the look ahead strategy. The road data comes from the highway E4 outside Linköping and the fuel saving is 0.31%.

the inclination is outside the range $[\beta, \gamma]$. This is because the algorithm calculates the instantaneous inclination as an interpolation between two consecutive samples. In Figure 4 the II-strategy is illustrated on a real road profile, and subplot 2 shows that the gear is seldom disengaged. Only in the last third of the driving scenario there are downhill slopes where the strategy can be used and the total reduction of fuel consumption is small. In Figure 5 it is seen, for this road profile, that the look ahead strategy works almost as the II-strategy in Figure 2. It would however be expected that the look ahead strategy should give a lower or at least equal fuel consumption as the II-strategy, and the reason for not being so is that the look ahead strategy only changes its control signal each 25 m. Driving in 85 km/h this is roughly 0.1 times the sampling frequency of the II-strategy. On the other hand, for the test profile in Figure 6 it is seen that the look ahead strategy disengages the gear not only in the downhill slopes but also in the flat sections between and after the downhill. Compared to the standard cruise controller the kinetic energy is increased in the downhill. Because of this the look ahead strategy can start to inject fuel later after the downhill, (subplot 2 around position $s = 900$ m), and still keeps the same speed as the standard cruise controller. The reduction of fuel consumption is almost 3 %. In Figure 7 it is seen that the look ahead strategy disengages the gear in the downhills of the third part of the driving scenario. Compared to the II-strategy in Figure 4, for the same real road profile, the gear is, in the look ahead case, disengaged during approximately three times as long. The reduction of fuel consumption is higher than for the II-strategy, but is still modest over the total distance.

5. CONCLUSIONS

Simulations shows that fuel consumption can be decreased with up to approximately 3% for some driving scenarios by disengaging the gear when driving downhill, and thereby increasing the vehicle's kinetic energy. This increase leads to lower fueling directly after the downhill. As mentioned in Section 3.5 the size of the optimization problem is linear in prediction horizon, and the case presented can easily be run well under real time.

If the vehicle is equipped with an automated manual transmission, see e.g. (Pettersson and Nielsen, 2000), or an automatic clutch, no extra hardware in the powertrain is needed to implement the control strategies presented. Since only the control software has to be changed the implementation cost is expected to be reasonably low.

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6. NOMENCLATURE

A	Vehicle cross sectional area
c_d	Air drag coefficient
c_i	Lumped vehicle model parameters
c_r	Rolling resistance coefficient
d_i	Engine drag torque parameters
F	Vehicle Forces
g	Gravitational acceleration
i	Gear ratio
J	Inertia
m	Vehicle mass
m_f	Fuel mass
n_{cyl}	Number of cylinders
n_r	Revolutions per engine cycle
q_{hv}	Fuel heating value
r_w	Wheel radius
T	Torque
v	Vehicle speed
α	Inclination
η	Efficiencies
Δ	Sample distance
δ	Injected amount of fuel per engine stroke
ρ	Air density
ω	Rotational speed

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