

Time-Resolved Estimation of Fuel Consumption Breakdown of a Heavy Duty Truck Under Actual Road Conditions

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ABSTRACT

Fuel economy and performance vary significantly with the vehicle design and configuration, road profile, and payload. The variation is more pronounced for heavy-duty trucks and understanding its origin is critical to maximizing fleet profitability. In this paper we demonstrate a method to continuously estimate fuel consumption breakdown over resistive forces while the vehicle is driven on a public highway. The method is fast, cost-effective, and capable of analyzing trip segments as short as one second. The method utilizes a non-linear Kalman filter and a vehicle dynamical model that has a coupled longitudinal and vertical motion.

The paper presents the breakdown of fuel consumption and an estimate of road grade profile obtained by driving a heavy-duty vehicle at the MnROAD research facility in Albertville MN. The road grade profile of the high-volume segment on Westbound Interstate 94 and the fuel consumption breakdown of the MnROAD heavy-duty test truck were estimated from recorded Control Area Network (CAN) signals and known vehicle parameters. The resulting estimates of the grade profile showed accuracy and repeatability within ± 0.15 degrees when compared to grade measured with a high precision differential GPS, thus confirming the accuracy of the estimation method. Also presented is a detailed breakdown of fuel consumption by grade resistance, rolling resistance, aerodynamic resistance, inertial forces, accessories, and driveline components. We discuss the implications of the results for vehicle fuel economy optimization, vehicle customization, and large scale grade profile determination.

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INTRODUCTION

Freight transportation is a significant contributor to the US economy. In 2011, US trucks delivered 67 percent of the domestic tonnage, consumed 52 billion gallons of fuel, and logged 397 billion miles over 2.6 million miles of paved roads [1, 2]. Freight transportation costs 603 billion dollars a year to US consumers and it directly impacts all segments of society [1, 3]. Even a few percent improvements in the efficiency of trucks can save billions of dollars in fuel costs and significantly reduce emissions. This paper aims at minimizing the vehicle operational cost by providing a breakdown of the fuel consumption, for example due to engine friction and air drag, under actual operating conditions. A detailed fuel and energy breakdown will enable fleets and manufacturers to compare fuel consumptions across multiple vehicle configurations and operating conditions and assess new efficiency technologies

[4]. Once the factors of excessive fuel consumption are identified, vehicle design, configuration and operation can be optimized for maximum fuel savings.

The interaction of freight trucks with the road consumes fuel and causes damage and deterioration to the road, thus increasing costs and decreasing safety. The friction between the road and the tires provides the stability and the tractive force that accelerates the vehicle, maintains its cruising speed, and decelerates it rapidly upon braking. The primary road characteristics affecting the frictional force and vibrational excitations are pavement, roughness, unevenness, and grade. Roughness, unevenness, and grade refer to variations in the road elevation profile in a length scale equivalent to tire contact size, to wheelbase, and to distances longer than vehicle length, respectively. Roughness and unevenness couple the longitudinal and vertical motion of the vehicle and generate transient dynamical contact forces that can stress the road and

potentially destabilize the vehicle. Dynamical loads also cost extra fuel. The vibrational energy dissipated in the tires and the suspensions, and the contact forces acting on the ground would appear as dynamical rolling resistance. Both the vertical and the longitudinal motion are eventually powered by engine torque that maintains the vehicle's cruise speed. Therefore, fuel economy, safety, and maintenance are related to road design and how vehicles interact with it. The method demonstrated in this paper can help set strategic priorities for vehicle design, road construction, pavement and maintenance, [3, 5].

In North America, heavy-duty trucks are custom built to match application and operator specifications. Vehicle customization endeavors to balance fuel economy and drivability; two important and competing goals. The fuel economy metric is defined as the kilometers per liters while the drivability metric is the reserve torque expressed in terms of equivalent percent grade at different operating conditions. Both metrics are a direct consequence of road properties. An optimal vehicle specification meets the desired drivability with the maximum fuel economy. The road grade profile is a significant factor in determining the minimum drivability requirements and consequently selecting an optimal powertrain configuration for fuel economy [3, 5]. Figure 1 shows the ability of the test truck, described in Appendix A, to overcome road grade at the top gear and at 105 km/h. Above three percent grade, the torque required for overcoming grade resistance and rolling resistance is greater than the engine peak torque at the top gear. At typical cruising speeds, when rolling resistance, aerodynamic drag, driveline and accessory losses are taken into account, the road grade limits the drivability to a less than 1.5 percent grade (shown as the Available Torque in Figure 1). Shifting to lower gears can provide the torque necessary to overcome grade resistance; however it will reduce fuel economy, lower maximum cruise speed, and increase engine wear because of operating near engine rated RPM. Thus, accurate road grade profile data were essential for configuring a vehicle that meets drivability requirements.

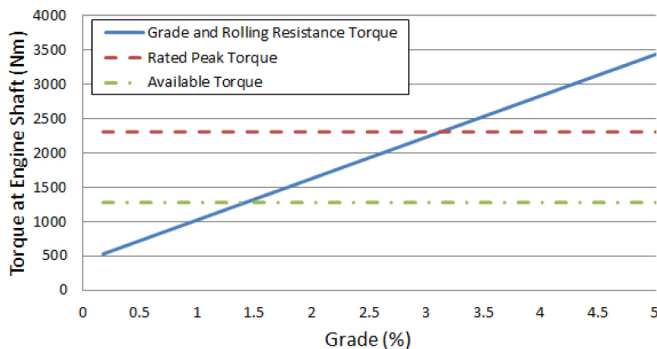


Figure 1. Solid line is the engine brake torque necessary to overcome the grade resistance and rolling resistance at the transmission top gear and at 105 km/h. The red dashed line is the engine peak brake torque of 2305 Nm. The green dot dashed line is the available torque which is the engine brake torque minus the accessory losses, aerodynamic drag, and driveline losses. At 105 km/h, the maximum grade is 1.4 percent. The percent grade is defined as $100 \cdot \tan(\chi_{road})$.

This paper also demonstrates how road grade estimation and fuel breakdown are related and how they are estimated simultaneously and consistently. The accuracy of the fuel consumption distribution over the rolling resistance, grade resistance and drag is established by comparing the resulting road grade profile to an independent road elevation measurement.

In the remainder of the paper, we present the experimental results and the theoretical model for vehicle and fuel consumption distribution over rolling resistance, aerodynamic resistance and individual powertrain components. We also present the equipment and track, data collection and analysis, and the results with a heavy duty vehicle.

VEHICLE MODEL

The vehicle dynamics utilized in the non-linear Kalman filter is a modified quarter car model that explicitly takes into account the coupling between the longitudinal and vertical motion. Quarter Car Model (QCM) is a common choice for the vehicle vertical motion where the road height profile is treated as a time dependent excitation [6]. The model is expressed in an inertial frame and the coupling between the road grade profile and vehicle is made explicit as shown in Figure 2. The definitions of the symbols in the following equations are given in Appendix B.

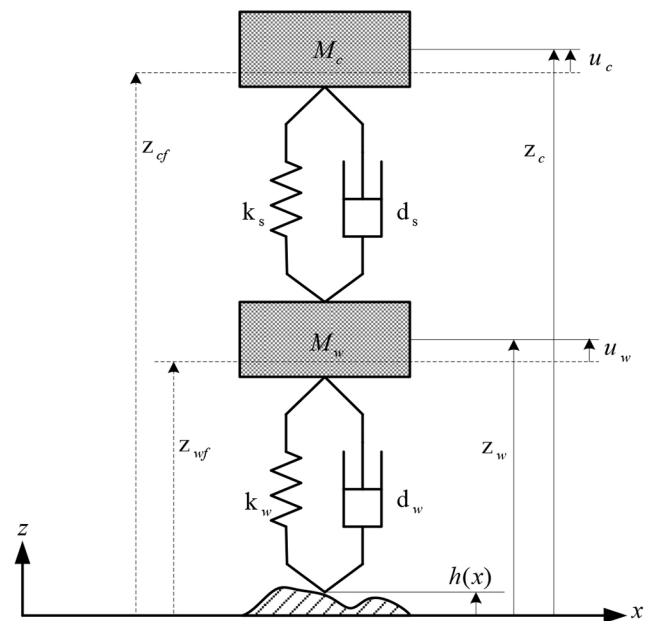


Figure 2. The quarter car model (QCM) accounts for the vertical and longitudinal motion coupling by expressing the road profile as a function of the longitudinal coordinate. QCM is chosen to estimate the dynamical contact forces.

The combined equations for vertical and longitudinal motion are:

$$\begin{aligned} M_w \ddot{z}_w = & -M_w g \cos(\chi_{road}) \\ & -k_w(z_w - z_{wf} - h) \\ & -d_w(\dot{z}_w - \dot{h}) \\ & -k_s(z_w - z_{wf} - z_c + z_{cf}) \\ & -d_s(\dot{z}_w - \dot{z}_c) \end{aligned} \quad (1)$$

$$\begin{aligned} M_c \ddot{z}_c = & -M_c g \cos(\chi_{road}) \\ & -k_s(z_c - z_{cf} - z_w + z_{wf}) \\ & -d_s(\dot{z}_c - \dot{z}_w) \end{aligned} \quad (2)$$

Under static conditions where there is no time dependence and ground height is zero:

$$h(0) = 0 \quad (3)$$

$$\begin{aligned} M_w g \cos(\chi_{road}) = & -k_w(z_{w0} - z_{wf}) \\ & -k_s(z_{w0} - z_{wf} - z_{c0} + z_{cf}) \\ = & -k_w u_{w0} - k_s(u_{w0} - u_{c0}) \end{aligned} \quad (4)$$

$$\begin{aligned} M_c g \cos(\chi_{road}) = & -k_s(z_{c0} - z_{cf} - z_{w0} + z_{wf}) \\ & -k_s(u_{c0} - u_{w0}) \end{aligned} \quad (5)$$

$$(M_c + M_w) g \cos(\chi_{road}) = -k_w u_{w0} \quad (6)$$

With respect to the static conditions, the model is expressed in terms of the spring displacements:

$$\begin{aligned} M_w \ddot{u}_w = & -M_w g \cos(\chi_{road}) \\ & -k_w(u_w - h) - d_w(\dot{u}_w - \dot{h}) \\ & -k_s(u_w - u_c) - d_s(\dot{u}_w - \dot{u}_c) \end{aligned} \quad (7)$$

$$\begin{aligned} M_c \ddot{u}_c = & -M_c g \cos(\chi_{road}) \\ & -k_s(u_c - u_w) - d_s(\dot{u}_c - \dot{u}_w) \end{aligned} \quad (8)$$

The ground contact force acting on the tire is

$$\begin{aligned} F_z = & -k_w(u_w - h) - d_w(\dot{u}_w - \dot{h}) \\ = & (M_w + M_c) g \cos(\chi_{road}) + M_w \ddot{u}_w + M_c \ddot{u}_c \end{aligned} \quad (9)$$

We assume that dynamical rolling resistance is accounted by the normal force variations. Then the dynamical rolling resistance is given by [7, 8]:

$$F_{RR} = -\mu_{RR} F_z \quad (10)$$

Typical values of μ_{RR} are between 0.005 and 0.010. The normal force F_z is the normal component of the gravitational force and the forces generated by the axle and suspension vibrations.

The contribution of the dynamical rolling resistance to fuel economy is usually computed as the average damping losses in the suspensions and tires where energy is dissipated after excitations by the road [9]. This approach is not suitable when the forces are time resolved, for example, when energy losses are computed in a time scale that is comparable to response time of the engine. Vibrational periods of suspensions are typically around one second during which a vehicle travels 30 meters at a cruising speed 105 km/h. In this paper, the dynamical rolling resistance contribution to fuel consumption is taken to be the work done on the tires by the instantaneous rolling resistance.

The grade force, the aerodynamic force, and total longitudinal resistive forces are:

$$F_{GR} = -(M_w + M_c) g \sin(\chi_{road}) \quad (11)$$

$$\dot{h} = \frac{\partial h}{\partial x} \frac{\partial x}{\partial t} = \tan(\chi_{road}) \dot{x} \quad (12)$$

$$F_{AR} = -\mu_{AR} (\dot{x} - \dot{x}_w)^2 \quad (13)$$

$$F_{xR} = F_{RR} + F_{GR} + F_{AR} \quad (14)$$

The driveline output force on the drive axle tires is:

$$F_{DL} = \mu_{DR} \gamma_c \gamma_{DL} F_{eL} \quad (15)$$

$$\tau_{DL} = r_{RR} F_{DL}$$

$$\gamma_{DL} = \gamma_{TR} \gamma_{RA} \quad (17)$$

$$r_{RR} = \frac{60 \gamma_{DL} \dot{x}}{2\pi RPM_e} \quad (18)$$

Thus, the vehicle center of mass motion becomes:

$$M_{eff} \ddot{x} = F_{DL} + F_{xR} \quad (19)$$

Equations (1), (2), (3), (4), (5), (6), (7), (8), (9), (10), (11), (12), (13), (14), (15), (16), (17), (18), (19) are integrated with fourth order Runge-Kutta method to propagate the system state in the Kalman filter.

FUEL CONSUMPTION BREAKDOWN

Fuel is the primary source of instantaneous power for overcoming forces acting on a moving vehicle. However, power may also be generated by the conversion of kinetic and potential energy of the vehicle. This section provides the expressions for distributing the power generated by the instantaneous fuel rates over the forces acting on vehicle. The fuel distributions of section Estimated Fuel Consumption Breakdown are computed by equations (20), (21), (22), (23), (24), (25), (26), (27), (28), (29), (30), (31), (32), (33), (34), (35), (36), (37), (38), (39), (40), (41), (42), (43), (44), (45), (46), (47), (48), (49), (50).

Upon combustion, the majority of fuel's chemical energy is consumed by the thermodynamic and frictional forces within the engine itself. The work done by the fuel combustion on the vehicle is either dissipated or converted into a recoverable source such as battery electrical energy, translational and rotational kinetic energy, and gravitational potential energy. Indicated torque is the mechanical torque generated by the fuel combustion. Brake torque is defined as the indicated torque minus the engine frictional losses, and it powers the engine accessories and the vehicle motion. The tractive torque is defined as the brake torque minus all the accessory, torque converter, and driveline losses, and it acts on the tires to drive the vehicle motion. The overall efficiency of the engine combustion and friction depends on the operating point.

When no fuel is injected into the engine, the power from all sources must be equal in magnitude to the power into the sinks:

$$\sum P_{Source} + \sum P_{Sink} = 0 \quad (20)$$

In (20), P_{Source} is a power source other than the fuel combustion and it is always positive. P_{Sink} is the power consumed by any force acting on the vehicle and it is always negative. Fuel is

injected into the engine when the total power consumption by sinks is greater than the total power generated from all the sources. The fuel mass injection rate is:

$$\dot{M}_{Fuel} = -\frac{1}{H_l} \left(\sum P_{Sink} + \sum P_{Source} \right) \quad (21)$$

In equation (21), the engine thermodynamic losses and engine friction losses are also treated as sinks. A vehicle cannot create fuel; therefore $\dot{M}_{Fuel}$ is always positive. When fuel is injected into the engine, the vehicle's tractive power is consumed proportionately by the sinks external to driveline, engine, accessories, and torque converter. We denote these external sinks by $Sink^*$.

$$\dot{M}_{Fuel, Sink^*} = \left(\frac{P_{Sink^*}}{\sum P_{Sink^*}} \right) \dot{M}_t \delta_+ (\dot{M}_t) \quad (22)$$

$$\begin{aligned} \dot{M}_t &= \dot{M}_{Fuel} - \dot{M}_{EF} - \dot{M}_{ET} \\ &\quad - \dot{M}_{AC} - \dot{M}_{TC} - \dot{M}_{DL} \end{aligned} \quad (23)$$

$$P_{Sink^*} \notin \{P_{Sink, EF}, P_{Sink, ET}, P_{Sink, AC}, P_{Sink, TC}, P_{Sink, DL}\} \quad (24)$$

Power consumed by the sinks does not necessarily come from the fuel combustion. For example, a vehicle may be accelerated by the gravitational force. In this case, a vehicle on a hill (whether started from the hill or propelled up the hill from fuel expended earlier in the trip) will convert potential energy to power that is consumed by the sinks regardless of the vehicle's fuel injection state.

Grade force contribution to power balance equation (21) is:

$$P_{Source, GR} = \dot{x} F_{GR} \delta_+ (F_{GR}) \quad (25)$$

$$P_{Sink, GR} = \dot{x} F_{GR} \delta_- (F_{GR}) \quad (26)$$

Rolling resistance and aerodynamic resistance are non-conservative forces and they will always appear as sinks.

$$P_{Source, AR} = 0 \quad (27)$$