A MECHANISTIC APPROACH FOR ESTIMATING THE ROUGHNESS-INDUCED ROLLING RESISTANCE AND FUEL CONSUMPTION OF ARTICULATED TRUCKS



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Abstract

Several studies have been conducted on how the surface characteristics (roughness and texture) affect fuel consumption. Bumpy roads increase the amount of resistance a vehicle experiences as it travels down the road, similar to how driving into a strong headwind requires additional fuel to maintain a certain speed, though on a smaller scale. Increased resistance translates to an increase in fuel consumption. In this paper, a mechanistic approach to estimate the effect of roughness on fuel consumption is proposed. The approach uses numerical modeling of vehicle response to estimate the dissipated energy of the suspension system due to roughness. The results were compared with the National Cooperative Highway Research Program (NCHRP) 720 empirical model predictions. The analysis shows that there is general agreement between the predictions; however, there are some differences between the uncalibrated results from the mechanistic approach and the empirical results, notably the non-linear effect with speed from the mechanistic approach, as compared to the linear trend that was observed from field measurements.

Keywords: Fuel energy consumption; Non-deformable pavement; Pavement-vehicle interaction; Stochastic road roughness; Pavement and bridge loading, lifecycle management.

1. Introduction

According to the 2012 U.S. Greenhouse Gas Inventory Report (EPA 2018), road transport has produced 28% of all emissions of greenhouse gases produced in the US. There are several factors contributing to fuel consumption and therefore emissions: The thermodynamic efficiency of the engine, the vehicle aerodynamics and its weight, the technological characteristics of the tires, and the pavement characteristics. Rolling resistance plays an important role in vehicle fuel consumption. From a pavement engineering point of view, there are several factors that can significantly affect rolling resistance: the geometry, the surface characteristics of the pavement and its structure. Several studies have been conducted on how the surface characteristics, in particular roughness and macrotexture affect fuel consumption (Chatti & Zaabar 2012). Bumpy roads increase the amount of resistance a vehicle experiences as it travels down the road, similar to how driving into a strong headwind requires additional fuel to maintain a certain speed, though on a smaller scale. Increased resistance translates to an increase in fuel consumption. The level at which fuel efficiency is affected is heavily tied to the condition of the roads, or the vehicle-pavement interaction (VPI). As reported in the study conducted by Chatti & Zaabar (2012), roughness is the leading factor and, therefore, is a key indicator of fuel consumption. There are various models available for estimating fuel consumption of a vehicle under different operating, weather, and pavement conditions. One commonly used model, the World Bank's Highway Design and Maintenance Standards Model (HDM) versions HDM-3 and HDM-4, accounts for the impact of roughness as part of the rolling resistance forces (Bennett et al., 2003). In 2012, Chatti & Zaabar (2012) calibrated the mechanistic-empirical HDM-4 model for vehicles used in the United States. They investigated the impact of pavement condition on vehicle fuel consumption, including the impact of pavement roughness for five different vehicle classes (medium car, SUV, van, light truck, articulated truck) under different operating, weather, and pavement conditions. This model has gained importance in the context of pavement life-cycle assessment methods that relate pavement condition to rolling resistance and corresponding fuel consumption and environmental impacts (Wang et al. 2012).

This paper proposes a new mechanistic formulation and solution procedure for estimating the energy dissipation in the tire and the suspension of a heavy articulated truck traveling on a non-deformable pavement at a constant speed, and compares the mechanistic results to the empirical NCHRP 720 predictions (Chatti & Zaabar 2012).

2. Proposed Mechanistic Approach

The proposed method adopts the basic approach by (Louhghalam et al. 2015), while simplifying and correcting some of the inputs needed in the model. This can overcome the computational challenges associated with more complicated vehicle and/or pavement models considered. In this approach, first the road roughness is modeled as a filtered white noise and combined with a quarter-car model of the vehicle to calculate the relative velocities of the sprung and unsprung masses. The dissipative energy of the system is calculated using the relative velocities. Then, the dissipated energy is converted to fuel consumption using the engine efficiency. Subsequently, the obtained values were compared to the results from the empirical model from the NCHRP 1-45 study (NCHRP report 720).

2.1 Random Profiles Generation Using a Shaping Filter Approach

In order to simulate vehicle behavior for the purposes of calculating the velocities of each component, it is necessary to assume a road roughness input. Profiles taken along a lateral line show the superelevation and crown of the road design, plus rutting and other distress. Longitudinal profiles show roughness and texture. Time domain simulations require the road roughness to be specified as a function of distance. For random road profiles, this requires the generation of a random function with the correct spectral density. There are two main ways of achieving this: (i) by generating white noise digitally and then filtering it to achieve the desired spectral density, or (ii) by ‘reverse spectral analysis’, using a fast Fourier transform. Both methods work well and can be used for most applications. In this paper, the first method is used.

In this paper, we focus on the longitudinal profiles, which classifications are based on the International Organization for Standardization (ISO 8608). The ISO has proposed road roughness classification using the power spectral density (PSD) values as shown in Figure 1 and Table 1. Paved roads are generally considered to be among road classes A to D.

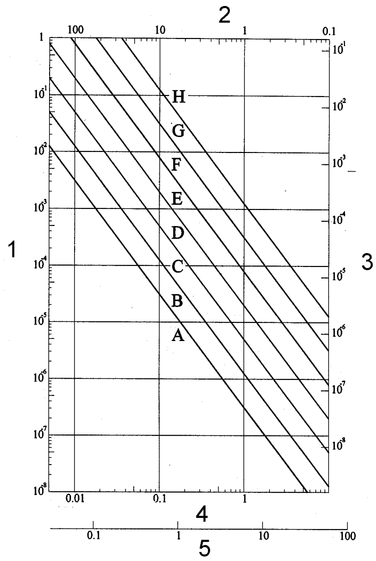

Figure 1 – Road Surface Classification (ISO 8608). The axes surrounding the frame are defined as 1: displacement PSD, Φ(n)[m3], 2: wavelength, λ[m], 3: displacement PSD, Φ(Ω)[m3], 4: spatial frequency, n[cycle/m], 5: angular spatial frequency, Ω[rad/m].

Table 1 - Degree of roughness expressed in terms of Ω (ISO 8608)

Road class	Degree of roughness $\Phi(\Omega_0)(10^{-6}m^3)$ where $\Omega_0 = 1$rad/m		
	Lower limit	geometric mean	upper limit
A (very good)	–	1	2
B (good)	2	4	8
C (average)	8	16	32
D (poor)	32	64	128
E (very poor)	128	256	512

The power spectral densities of roads show a characteristic drop in magnitude with the wave number. To determine the power spectral density function, or PSD, it is necessary to measure the surface profile with respect to a reference plane. Random road profiles can be approximated by a PSD in the form of:

$$\Phi(\Omega)=\Phi(\Omega_0)\left(\frac{\Omega}{\Omega_0}\right)^{-w} \text{ or } \Phi(\Omega)=\Phi(n_0)\left(\frac{n}{n_0}\right)^{-w} \tag{1}$$

Where $\Omega = \frac{2\pi}{L}$ in rad/m denotes the angular spatial frequency and L is the wavelength. Φ_0 or $\Phi(\Omega_0)$ in m^2/(rad/m) describes the values of the PSD at the reference wave number Ω_0 = 1rad/m, $n = \frac{\Omega}{2\pi}$ is the spatial frequency, n_0 = 0.1cycle/m, and w is the waviness (for most of the road surfaces, w = 2).

If the vehicle runs with constant velocity V, then the road profile signal, $z_R(t)$, whose PSD is given by equation 1, may be obtained as the output of a linear filter expressed by the differential equation (Hac, 1985):

$$z_R(t)=-\alpha V z_R(t)+w(t) \tag{2}$$

Where $w(t)$ is a white noise process with the spectral density $\psi_\omega = 2\alpha V\sigma^2$; σ^2 denotes the road roughness variance; and α depends on the type of road surface. The most used expression in the literature to define pavement roughness variance is defined as follow {(ISO 8608), (Sun and Kennedy, 2002)}:

$$\sigma^2=\frac{1}{2\pi}\int_0^\infty \Phi(\Omega)\,d\Omega \approx 4\Phi(\Omega_0) \tag{3}$$

To determine α, we simply use the relationship:

$$\Phi(\Omega_0)=\frac{2\alpha\sigma^2}{\Omega_0^2+\alpha^2}=\frac{4\alpha\Phi(\Omega_0)}{\Omega_0^2+\alpha^2} \tag{4}$$

which yields α = 0.127 (rad/m).

Figure 2 depicts the MATLAB Simulink® model of the first order filter given by Equation 2. Table 2 presents the values for the parameters needed to generate roughness profiles with different roughness levels.

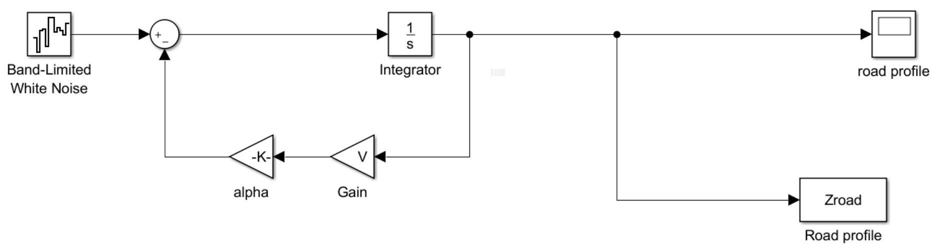

Figure 2 – First order linear system Simulink model for roughness generation

Table 2 - Road roughness model parameters

Road class	σ (10^{-3}m)	$\Phi(\Omega_0)$ ($10^{-6}m^3$), Ω_0 = 1	α (rad/m)
A (very good)	2	1	0.127
B (good)	4	4	0.127
C (average)	8	16	0.127
D (poor)	16	64	0.127
E (very poor)	32	256	0.127

2.2 Quarter-Truck Model

This section describes the analytical methods used to determine the dynamic response of an articulated truck for purposes of determining the instantaneous relative velocities of the sprung (body) and unsprung (axle) masses as they move along the road. There have been a number of such models with various specific parameter values proposed to emulate the response of a wide variety of vehicles. The 'Quarter car' parameters for an articulated truck with linear parameters from {(Cebon, 1993), (Gillespie, 1993), (de Pont, 1994), (Karagania, 1997), (Fu and Cebon, 2002)} are used in this analysis and given in Figure 3. Matlab Simulink/Simscape™ platform was used to implement the model due to its simplicity and its flexibility to include non-linear behavior for the suspensions in the future.

Model parameters		
Name	Unit	Value
m_s	Kg	3900
m_u	Kg	484
k_s	MN/m	0.878
k_t	MN/m	1.49
c_s	KNs/m	1.755
c_t	KNs/m	17.5

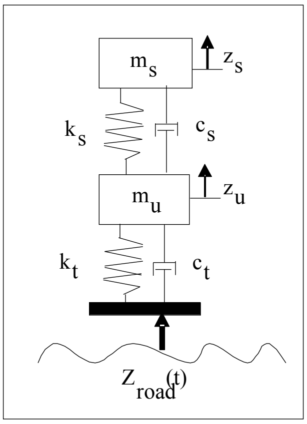

Figure 3 – Schematic of two degrees of freedom quarter-truck model

2.3 Energy Dissipation and Fuel Consumption Calculation

Measure of Energy Dissipation

In the context of a quarter vehicle model, the energy dissipation of the vehicle is due to only the relative velocities ($z = z_s - z_u$, $z_t = z_u - z_{road}$). Therefore, the Simulink model for the quarter vehicle was set up such that only the relative velocities will be output from the model. Then, the dissipated energy per travelled road length from the total system is calculated using Equation (5) at constant speed V:

$$D = \frac{C_s}{V} E[z^2] + \frac{C_t}{V} E[z_t^2] \qquad (5)$$

Calculation of Excess Fuel Consumption

The excess fuel consumption per travelled road length from the quarter system is calculated using Equation 6 at constant speed V:

$$\Delta Fuel_{IRI} = \frac{D_{IRI}}{\xi_b} = \frac{1}{\xi_b}\left(\frac{C_s}{V} E[z^2] + \frac{C_t}{V} E[z_t^2] \right) \qquad (6)$$

The factor ξ_b is the effective calorific value of the combustible, and is a function of the engine technology. According to Baglione (2007), the maximum efficiency is about 40% for Diesel engines. This percentage represents the energy released by the engines that will be available to move the vehicle. Since the calorific value is about 40 MJ/L for Diesel, the value used for ξ_b is 16 MJ/L, in accordance with the values reported in NCHRP 720 (Chatti & Zaabar 2012).

Calculation of Total Fuel Consumption

To calculate the total fuel consumption of a vehicle ($Fuel_0$) at the baseline conditions, first the calibrated HDM 4 model from the NCHRP 720 study is used assuming IRI=0 and with the corresponding vehicle mass M. For the articulated truck, tire stiffness and damping coefficients, k_t and c_s, shown in Figure 3, are typical for a dual tire set. The sprung mass, m_s, represents the portion of the total sprung mass supported by one-half of one-axle. The total axle load is therefore equal to twice the sum of the quarter-truck sprung and unsprung mass. The total mass of an equivalent articulated truck is calculated using equation 7:

$$M_{AT} = N_{4.4t} * (m_s + m_u) \tag{7}$$

For an 18 wheeler with 4x8.8t axles with dual wheels and one 4.4t axle with single wheel, $M_{AT} = 39.5t$. Figure 4 presents these predictions in mL/km at speeds ranging from 5 km/h to 150 km/h. The values were generated at a representative average temperature of 17℃, a grade of 0%, IRI of 0 m/km and texture depth, Tdsp, of 0mm. Then, the total change in fuel consumption of a vehicle is calculated by, first multiplying the dissipated fuel consumption due to roughness ($\Delta Fuel_{IRI}$) calculated using Equation 6 by the number of axles, then dividing the obtained value by the one from the NCHRP 720 study assuming IRI=0 m/km ($Fuel_0$).

$$Fuel_{IRI(i)} = \frac{\left(M_{AT} / 4.4 \right) \times \Delta Fuel_{IRI(i)}}{Fuel_0} \tag{8}$$

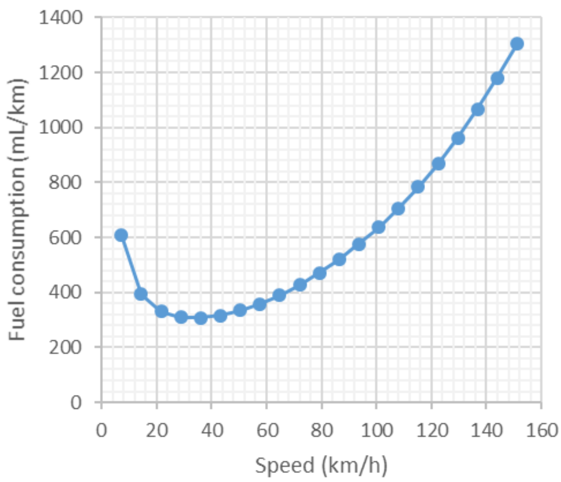

Figure 4 – Fuel consumption for the baseline condition as predicted by the NCHRP 720 model

3. Results

To evaluate the mechanistic approach, the results from the mechanistic approach were compared to the results reported in the NCHRP 720 report (Chatti and Zaabar, 2012). In the NCHRP study, the effect of pavement conditions was investigated using five instrumented vehicles to measure fuel consumption (FC) over different pavement sections with different pavement conditions. This data was used to calibrate the HDM 4 FC model. The calibrated models were verified and were found to adequately predict the fuel consumption of five different vehicle classes under different operating, weather, and pavement conditions. The details of the calibrated HDM 4 fuel consumption model are presented in (Chatti and Zaabar, 2012).

Pavement surface profiles with roughness levels ranging from 0.5 to 6 m/km at speeds ranging from 5 to 150 km/h were generated. For the mechanistic approach, the proposed analytical quarter-truck model was used to determine the dynamic response of the articulated truck to the generated surface roughness profiles in terms of relative velocities in the suspension and the tire. Then, the excess fuel consumption for different speed and roughness levels was calculated (Figure 5). The roughness and speed values were also used in the NCHRP 720 VOC module to calculate the fuel consumption at each speed and roughness level. The calculated values are presented in Figure 6. Both figures show that the effect of roughness increases then decreases as the vehicle speed increases, with the maximum being at about 40 km/h.

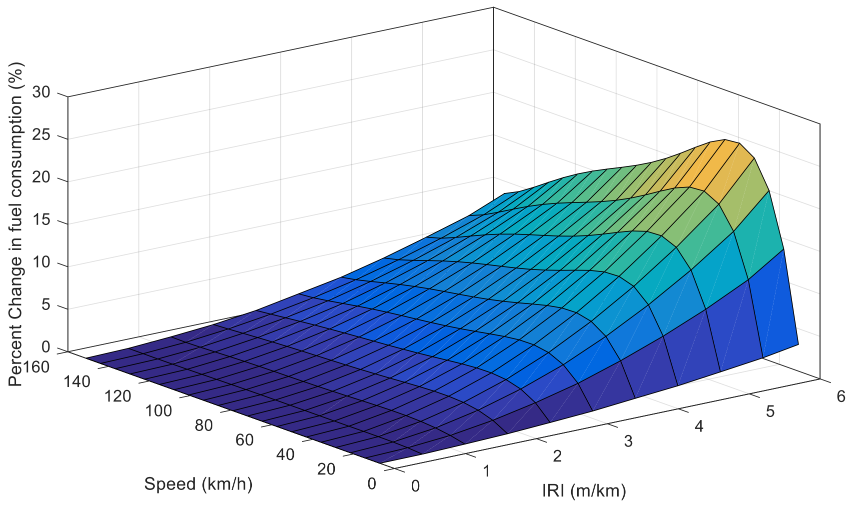

Figure 5 – Vehicle speed and roughness effect on fuel consumption of the total articulated truck as predicted by the mechanistic approach

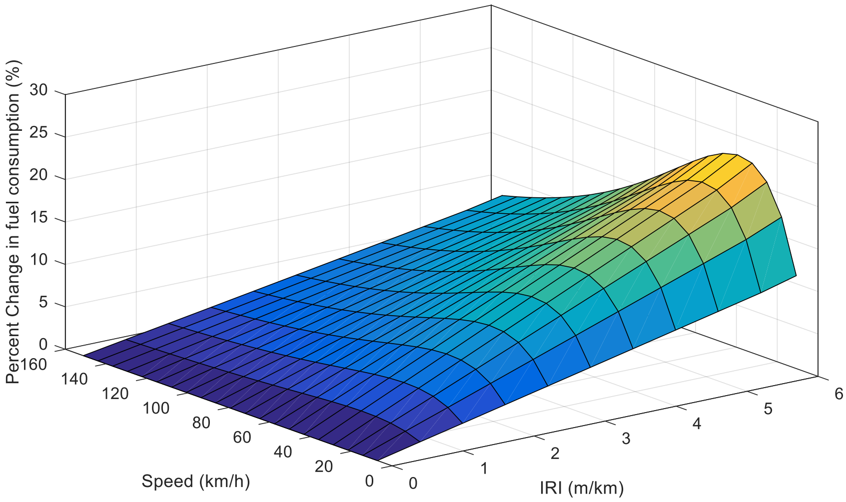

Figure 6 – Vehicle speed and roughness effect on fuel consumption of the total articulated truck as predicted by the NCHRP 720 model

Figure 7 shows the comparison of the predictions of fuel consumption by the heavy articulated truck using the new mechanistic approach and the empirical approach (NCHRP 720 model). The results are presented in terms of percent change fuel consumption relative to a baseline of IRI=0 m/km. The mechanistic approach shows that the relationship between roughness and fuel consumption is non-linear. Specifically, the dissipated energy scales with the square of IRI, as shown by Louhghalam et al. (2015). The analysis also shows that there are some differences between the uncalibrated results from the mechanistic approach and the empirical results. Specifically, the results indicate that the mechanistic model underestimates the fuel consumption at lower IRI values, compared to the empirical NCHRP 720 predictions, and overestimates it at higher IRI values. For comparison, the predictions from the mechanistic quarter car model for a passenger car is shown in Figure 8. The figure shows that the results from the mechanistic approach agree well with the empirical results, without the need for any calibration, up to about 3.5 m/km. The fact that the mechanistic and empirical models agree at the lower IRI values for the lighter passenger car suggests that the difference between the models for the heavy articulated truck case at lower IRI values is potentially due to the effect of structural rolling resistance from the dissipated energy due to the deformation of pavement structure. This would corroborate the empirical results reported by Chatti and Zaabar (2014) and Zaabar and Chatti (2014) that showed that the effect of pavement structure at low roughness levels is observed only for heavy trucks at low speed and summer temperature conditions. On the other hand, the difference between the mechanistic and empirical models at higher roughness levels could be caused by the quarter-car/truck model hypothesis, since the model assumes full contact at all times between the tires and the surface, which could be violated at higher levels of roughness. Also, although inertial properties (such as sprung and unsprung masses) of the vehicle can be estimated with reasonable accuracy, differences in the design of drive and trailer axle suspensions and active versus passive suspensions could explain some of the differences. The quarter-car model is widely used to study vehicle vibration behavior, particularly of cars. However, it ignores some effects which are important in heavy vehicle vibration (Fu and Cebon, 2002): (i) nonlinearity, particularly due to friction; (ii) roll and pitch motions of axles and sprung masses; and (iii) the influence of coupled suspension systems (for example, the load equalizing suspensions often fitted to tandem axles). Also, there are other sources of energy dissipation in the suspension such as the friction between various complex components of a vehicle suspension, etc.

4. Conclusions

In this paper, a mechanistic approach to estimate the effect of roughness on fuel consumption is proposed. The approach uses numerical modeling of vehicle response to estimate the dissipated energy of the suspension system due to roughness. The results were compared with the NCHRP 720 empirical model predictions. The analysis shows that there is general agreement between the predictions; however, there are some differences between the uncalibrated results from the mechanistic approach and the empirical results, notably the non-linear effect with speed from the mechanistic approach, as compared to the linear trend that was observed from field measurements.

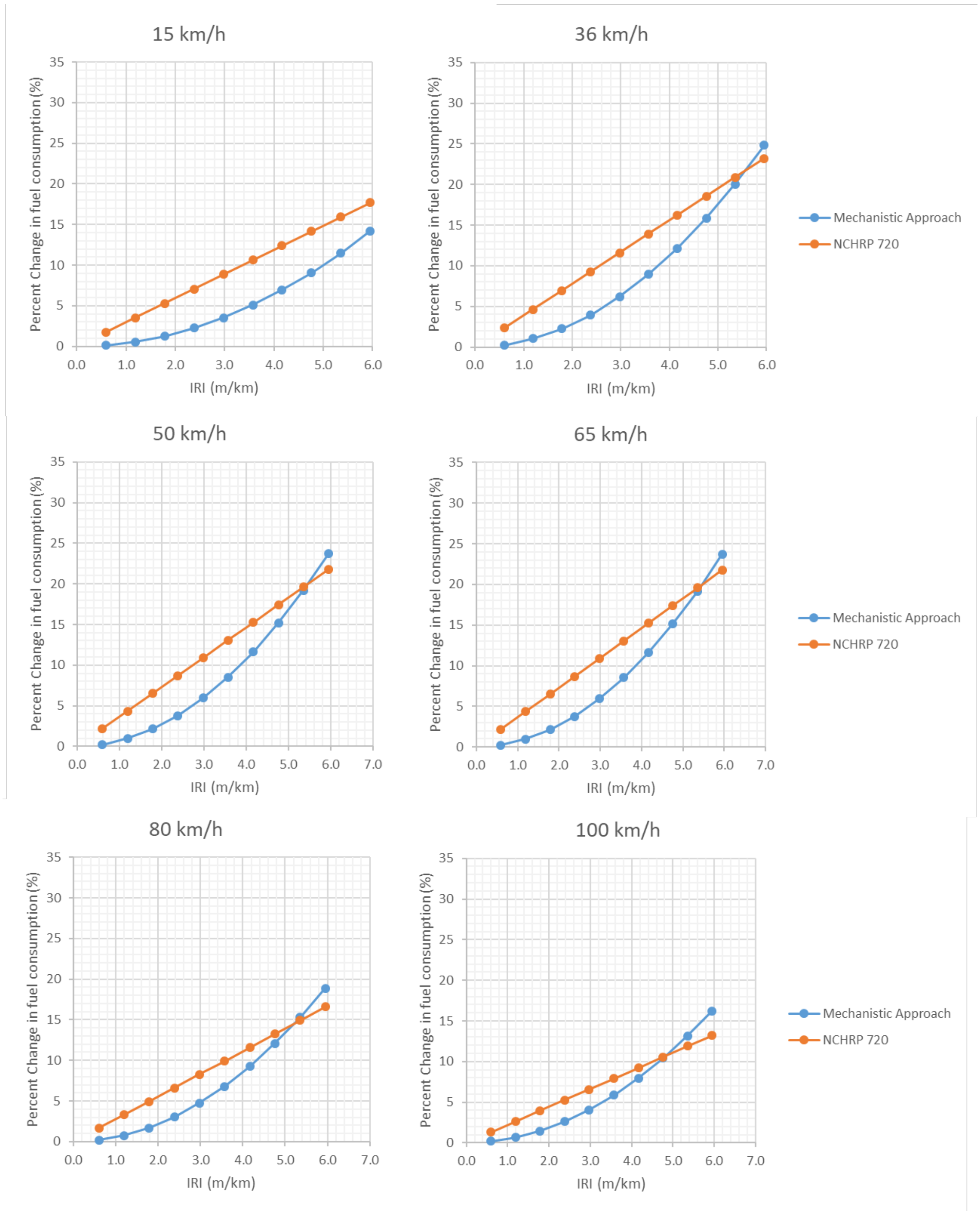

Figure 7 – Excess fuel consumption of an articulated truck as a function of IRI at different speeds

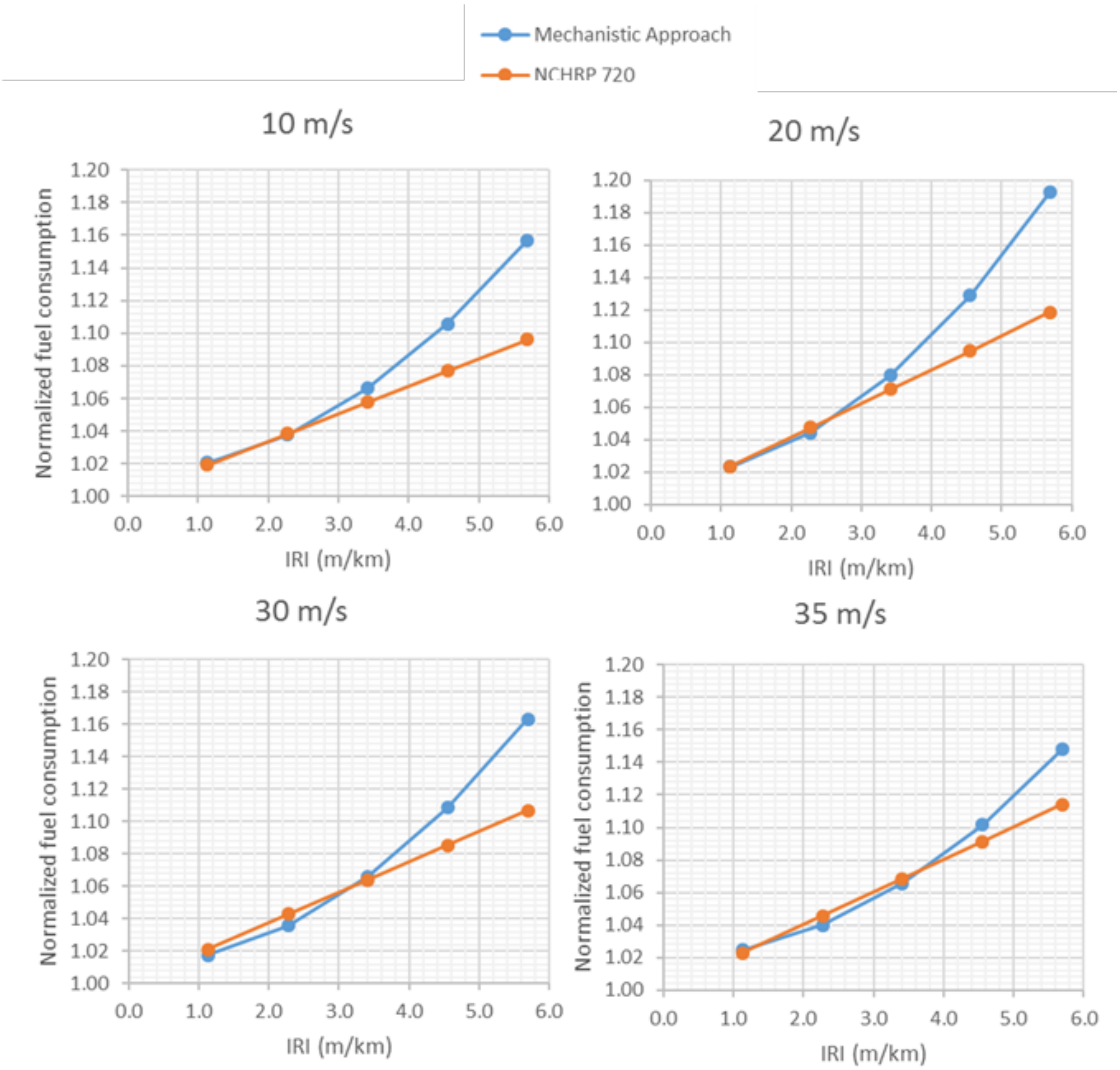

Figure 8 – Excess fuel consumption of a passenger car as a function of IRI at different speeds

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