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Original scientific paper

Boris Stojić[1]
Nebojša Nikolić[2]
Luka Sekicki[3]

A FLEXIBLE COMPUTATIONAL MODEL FOR FUEL CONSUMPTION ESTIMATION IN ARBITRARY VEHICLE DRIVING CYCLES

ABSTRACT: Vehicle fuel consumption and related topics have long attracted considerable research interest because of their economic, environmental and social implications. In this paper, the FuelSim_v1.0 computer program for vehicle fuel-consumption estimation was developed in GNU Octave programming environment. Consumption estimation model is based on the vehicle longitudinal dynamics and uses quasi-static method of the fuel consumption estimation. This approach divides the driving cycle into a large number of short time intervals, applies a steady-state calculation to each interval, and subsequently sums the corresponding partial fuel-consumption quantities. Input data for vehicle and driving cycle are provided through externally configurable files so that different kinds of vehicles can be examined in wide spectrum of driving conditions. Program, that is primarily conceived as educational and research tool, can be used for analysis of various types of motor vehicles in various usage patterns. It enables examination of influence of multiple kinds of design and/or operation parameters on the result, parameter sensitivity analyses etc. Preliminary examination of the simulation outputs indicated correct program functionality and consistency of the results obtained with the expected qualitative behaviour of the simulated vehicle. Formal validation, however, is left for the future work.

KEYWORDS: motor vehicle, fuel consumption, energy consumption, driving cycle, simulation

FLEKSIBILNI RAČUNARSKI MODEL ZA PROCENU POTROŠNJE GORIVA U PROIZVOLJNIM VOZNIM CIKLUSIMA VOZILA

REZIME: Potrošnja goriva vozila i srodne teme već dugo privlače značajnu pažnju istraživača zbog svojih ekonomskih, ekoloških i društvenih implikacija. U ovom radu razvijen je računarski program FuelSim_v1.0 za procenu potrošnje goriva motornih vozila, u programskom okruženju GNU Octave. Model za procenu potrošnje zasnovan je

[1] *Boris Stojić, University of Novi Sad, Faculty of Technical Sciences, Trg Dositeja Obradovića 6, 21000 Novi Sad, Serbia, bstojic@uns.ac.rs, https://orcid.org/0000-0001-8344-1466*
[2] *Nebojša Nikolić, University of Novi Sad, Faculty of Technical Sciences, Trg Dositeja Obradovića 6, 21000 Novi Sad, Serbia, nebnik@uns.ac.rs, https://orcid.org/0000-0003-4657-3640*
[3] *Luka Sekicki, University of Novi Sad, Faculty of Technical Sciences, Trg Dositeja Obradovića 6, 21000 Novi Sad, Serbia, luka.sekicki@uns.ac.rs, https://orcid.org/0009-0007-9170-8352*

na uzdužnoj dinamici vozila i koristi kvazistatičku metodu procene potrošnje goriva. Ovaj pristup podrazumeva podelu voznog ciklusa na veliki broj kratkih vremenskih intervala, primenu proračuna za slučaj ustaljenog kretanja u svakom od njih, a zatim sabiranje odgovarajućih parcijalnih količina potrošenog goriva. Ulazni podaci za vozilo i vozni ciklus zadati su putem eksterno konfigurabilnih datoteka, tako da se mogu analizirati različite vrste vozila u širokom spektru uslova vožnje. Program, koji je prvenstveno zamišljen kao obrazovni i istraživački alat, može se koristiti za analizu različitih vrsta motornih vozila pri različitim načinima upotrebe. Omogućava ispitivanje uticaja različitih konstrukcionih i/ili eksploatacionih parametara na rezultat, analize osetljivosti na promenu parametara i slično. Preliminarno ispitivanje rezultata simulacije ukazalo je na ispravno funkcionisanje programa i usklađenost dobijenih rezultata sa očekivanim kvalitativnim ponašanjem simuliranog vozila. Formalna validacija, međutim, ostavljena je za buduća istraživanja.

KLJUČNE REČI: motorno vozilo, potrošnja goriva, potrošnja energije, vozni ciklus, simulacija

INTRODUCTION

Vehicle energy consumption, and the associated consumption of fuel or other energy resources, has long been an important economic, environmental and social concern. The importance of this topic stems largely from its direct consequences, including resource depletion and pollutant emissions. The study of energy and fuel consumption in motor vehicles along with connected topics has therefore long attracted considerable research attention and has been addressed in numerous publications. Besides directly addressing fuel saving potentials and pollutant minimization, other topics that are frequently dealt with by researchers comprise e.g. control optimisation, route planning, smart resource exploitation, alternative and renewable energy sources research etc. Examples include e.g. the works of researchers such as Ben-Chaim, M. et al. (2013), Dekraker, P. et al (2018)., Genikomsakis, K. N. & Mitrentsis, G. (2017), Pfriem, M. & Gauterin, F. (2016), Yuan, X. et al. (2017), and many others.

Therefrom, motivation for this paper arises in order to provide further contributions to this research field. Goal of the paper is to develop simulation tool, i.e., computer program for implementation of the model aimed to estimate the fuel consumption of arbitrary motor vehicle in arbitrary driving cycle. Program should exhibit flexibility with regard to characteristics of the applied driving cycle, vehicle parameters and powertrain operating conditions, since these parameters substantially impact consumption. Flexibility is provided in a way that both driving cycles and vehicle-related parameters are provided as externally configurable datasets, allowing analysis of the broad spectrum of motor vehicles and driving situations without modification of the computational structure itself.

The presented approach is primarily intended as a flexible research and educational tool suitable for comparative analysis of vehicle design and operating conditions, driving cycle and transmission strategies influence on estimated fuel consumption. In addition to calculation of consumption itself, such tool should be capable of being implemented for more general studies, such as:

- Comparison of various standardized and non-standardized driving cycles.
- Influence study of the vehicle design and operational parameters.
- Sensitivity analysis.
- Assessing the effectiveness of technologies regarding fuel consumption.
- Assessing driving style impact and saving potentials through its improvement.
- Different areas of optimisation (design parameters, control strategies, gear shifting schedule, route planning...).

Method should be computationally efficient, relying on simple but accurate physical model of vehicle longitudinal dynamics, not requiring highly-detailed vehicle and its subsystems description and complex datasets. Model and software structure are supposed to be convenient for further upgrades, enhancements and extensions with new features.

Generally, procedures for consumption estimation share many common elements and approaches regardless of the powertrain type in regard. Additionally, in some cases powertrain-specific aspects still have to be accounted for. This paper and work carried out herein focuses on powertrain that can still be considered as conventional one, i.e., one consisting of the internal combustion (IC) engine as a prime mover, main clutch and a gearbox with the finite number of gear ratios. This approach should be viewed as an introductory development phase, as future developments of the program and its capabilities are intended.

THEORETICAL BASICS OF VEHICLE ENERGY AND FUEL CONSUMPTION

Motion resistances and driving cycle energy requirements

During arbitrary driving conditions, vehicle encounters a range of sources of energy consumption. Impact of these individual sources, i.e., their contribution to overall energy consumption in relation to each other, depends on vehicle type and design parameters, as well as current driving pattern and conditions.

External driving resistances

External motion resistances are summarized in Figure 1 and briefly described subsequently.

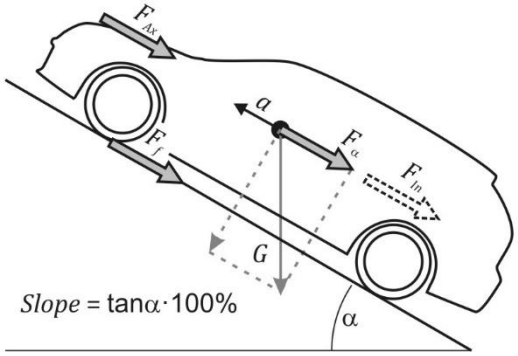


Figure 1 Motion resisting forces: $\boldsymbol{F_f}$ – rolling resistance (summary for all wheels); $\boldsymbol{F_{Ax}}$ – aerodynamic resistance; $\boldsymbol{F_\propto}$ – road gradient resistance; $\boldsymbol{F_{In}}$ – D'Alembert inertial force ($\boldsymbol{a}$ – acceleration); $\boldsymbol{G}$ – vehicle weight; $\boldsymbol{\alpha}$ – road gradient angle

First dissipative force, which always accompanies vehicle motion, is rolling resistance F_f originating from internal friction or hysteresis in rolling tyre. It is usually described by expression (1):

$$F_f = m \cdot g \cdot f \cdot \cos\alpha \tag{1}$$

Here, m is vehicle mass, g is gravitational acceleration, α is road gradient or slope angle, and f is rolling resistance coefficient, for which dependency on multiple factors such as vehicle speed, ground force and/or tire pressure can be introduced by modelling if required, which is omitted herein. For values of road gradients typically encountered by road vehicles, cosine from expression (1) remains very close to unity and can therefore usually be approximated as 1. Should, however, vehicle encounter higher road gradients, such as in off road driving or other extreme conditions, it's more exact value should be used.

Next dissipative force is aerodynamic resistance F_{Ax} (expression (2)), originating from complex interactions of the moving vehicle with the surrounding air.

$$F_{Ax} = C_{Ax} \cdot A \cdot \frac{\rho \cdot v^2}{2} \tag{2}$$

Here, C_{Ax} is aerodynamic drag coefficient, A is vehicle frontal area, ρ is air density, and v is vehicle speed, providing no wind is present (otherwise, relative speed between vehicle and wind has to be used, possibly including dependency on the yaw angle of the wind direction with regard to the vehicle). If vehicle moves at very low speed, contribution of the aerodynamic resistance to vehicle energy consumption can be practically neglected compared to other effects. On the other hand, at highway speeds it typically becomes one of the dominant contributors to the vehicle's motion resistance.

Road gradient resistance $F_\propto$ (expression (3)) is present when vehicle negotiates upward road slope. It represents component of gravitational force acting along the direction of travel. Contrary to rolling and air resistances, this force, originating from gravitational potential energy due to altitude changes, is conservative. For downward slope, numerical value of this force is negative, meaning that it aids vehicle motion instead of opposing it.

$$F_\propto = m \cdot g \cdot \sin\alpha \approx m \cdot g \cdot \frac{Slope\ [\%]}{100} \tag{3}$$

As shown in expression (3), for determination of the road gradient resistance F_α, road $Slope$ in percent can also be used instead of the angle α, taking into account that $\tan\alpha = Slope/100$ (see also Figure 1), and that approximation $\sin\alpha \approx \tan\alpha$ for small angles is fairly useful for road vehicles (e.g. Mitschke, M. & Wallentowitz, H. 2014).

Next conservative effect is storing of kinetic energy in the vehicle, which requires vehicle to accelerate until certain amount of it is reached. Effect of the kinetic energy accumulation through the acceleration is described by expression (4) through fictitious inertial force F_{In} according to the D'Alembert principle:

$$F_{In} = \left(m + \frac{J_{Equ}}{r_W^2}\right) \cdot a \approx \delta \cdot m \cdot a \tag{4}$$

Here, a is vehicle longitudinal acceleration, and J_{Equ} represents equivalent moment of inertia of all rotational masses as felt at the wheels. For calculation simplification, this quantity can be replaced by the coefficient δ, describing the same effect in the empirical way. Finally, r_W is effective wheel radius.

In this paper, empirical relationship according to Gillespie, T. (2021) will be used in order to evaluate coefficient δ, as given by the expression (5). This value depends on overall transmission ratio, since it is influenced by the share of rotational masses in overall kinetic energy of the vehicle. Therefore, it has to be calculated for every transmission gear individually.

$$\delta = 1.04 + 0.0025 \cdot i_{Tr}^2 \tag{5}$$

Here, i_{Tr} represents the current value of the overall transmission gear ratio, i.e., in the gear currently selected.

Internal driving resistances and summary of internal dissipative effects

Transmission losses (also dissipative by nature) and drive of auxiliary units can be regarded as internal driving resistance. Most frequently, they are taken into account in the simplified way through the overall transmission efficiency η_{Tr} that is assumed to have constant value, as given e.g. by Gillespie, T. (2011), Mitschke, M. & Wallentowitz, H. (2014), Guzzella, L. & Sciarretta, A. (2022). As shown subsequently, dissipative effects of the friction braking (and possibly energy losses of the recuperative braking if applied) also have to be taken into account if entire driving cycle energy balance is considered.

Wheel force and driving cycle energy balance

Driving wheel force F_W required to overcome all driving resistances can be calculated according to the expression (6):

$$F_W = F_f + F_{Ax} \pm F_\alpha + F_{In} \tag{6}$$

On the other hand, value of this force that is available from the powertrain is given by the expression (7):

$$F_W = \frac{T_W}{r_W} = \frac{T_e \cdot i_{Tr} \cdot \eta_{Tr}}{r_W} \tag{7}$$

Here, T_W is the summary driving torque applied to the driving wheels, T_e is the engine torque, i_{Tr} is current value of overall transmission gear ratio, and η_{Tr} is overall transmission efficiency.

If negative value of required wheel force is obtained in certain phases of driving cycle by using expression (6), that means that the vehicle is under braking instead of driving (braking torque is being applied to the wheels). In that case, the power flow changes direction, going from the vehicle to the wheels, and therefrom to the transmission, and eventually to the engine (which then provides braking torque) if the main clutch is not disengaged. This backward flowing power / energy can then, partially or entirely, be used for overcoming transmission losses, auxiliary units drive and engine friction and pumping losses, at a cost of vehicle kinetic or gravity energy. If all these losses and energy requirements are completely covered by available kinetic energy of the decelerating vehicle, and/or gravitational energy of the downwards driving one, the engine fuel supply can be cut off for consumption savings and pollutant emission reduction. Should available backward power be larger than this limit value for a finite amount of time, remaining energy can be partially recuperated if appropriate system is installed in the vehicle. Without such system, all braking energy is lost by dissipation through the action of friction braking system or engine braking. At this place it should be highlighted that, even when using recuperating device, its own efficiency has to be taken into account

because some energy losses will inevitably occur during energy conversion steps. These losses also add to dissipative effects.

As long as the vehicle is in motion, i.e., as long as any amount of kinetic energy remains available, it will be dissipated through rolling and air resistances, even during braking. This prevents full amount of kinetic energy stored in the vehicle to be supplied to recuperation system during recuperative braking.

If zero-wheel force is obtained by expression (6), that means that wheels are in the state of the free rolling, which occurs when external motion resistances are in balance with mechanical energy stored in the vehicle. Some amount of fuel will, even in such situation, be consumed for covering internal losses mentioned above.

Considering energy balance of the vehicle motion, vehicle itself can be seen as a storage system for conservative energy forms (kinetic and altitude gravity potential), supplied to the vehicle by the powertrain, for which itself certain amount of energy supply through the fuel consumption is required. On the other hand, different aforementioned dissipative mechanisms and auxiliary units drive can be seen as effects that drain this energy storage (Guzzella, L. & Sciarretta, A. 2022). Dissipative effects require constant energy supply through powertrain for the vehicle to keep moving, as opposed to conservative effects, which require a corresponding amount of energy to be supplied only once. On the other hand, as shown, compensation of energy dissipation partially occurs also through conservative mechanical energy that has already been stored in the vehicle.

Finally, to establish connection between vehicle and engine operating point (defined by pairs (F_W, v) and (T_e, n_e), respectively), required for fuel consumption calculation, relationship between vehicle speed and engine speed is required. This is given by expression (8):

$$v = \frac{\pi \cdot r_W \cdot n_e}{30 \cdot i_{Tr}} \tag{8}$$

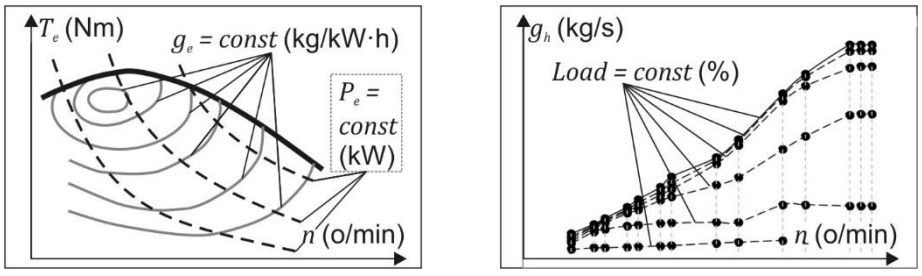
Here, v is vehicle speed in [m/s] and n_e is engine speed in revolutions per minute. If unit of [km/h] is used for calculations, appropriate unit conversion has to be carried out.

Prime mover – from energy to fuel consumption

Prime movers convert fuel energy to mechanical energy required for vehicle to negotiate certain driving cycle. This way, vehicle energy consumption turns to fuel consumption (which is also directly related to CO_2 and other pollutants emission, although this important topic is not covered by this paper). IC engines are generally characterized by relatively low efficiency. Peak efficiency values are typically not much higher than around 30% for spark-ignition engines and 40% for compression-ignition engines (Heywood, 2018). At partial loads, which occur frequently under typical road-vehicle driving conditions, efficiency is substantially lower. These relatively low efficiencies represent a major limitation in the overall conversion of fuel energy into mechanical energy.

Energy efficiency and/or fuel consumption data for IC engines can be provided by several mutually dependent indicators: thermal efficiency ratio (η_e, [-] or [%]), brake-specific fuel consumption (BSFC - g_e, [g/kWh] or equivalent unit) or fuel rate (g_h, [g/h] or equivalent unit).

Efficiency indicator values vary significantly with engine load and speed, usually exhibiting optimum at relatively high engine loads and moderate speeds. Two kinds of the diagrams for engine efficiency and fuel consumption are shown in Figure 2, indicating their general shape.



a) b)

Figure 2 a) Engine efficiency or BSFC map (shell diagram): thick line – engine torque at full load; thin lines: constant values of engine efficiency or BSFC; dashed lines: constant-power curves.
b) Fuel rate vs. engine rpm, for different values of engine load

Figure 2 a) shows efficiency/consumption map also known as shell diagram. This type of chart uses contour lines connecting points with equal efficiency values, indicated through thermal efficiency ratio or BSFC. While this type of

map is better suited for theoretical considerations, the second one, shown in Figure 2b), is more appropriate for practical calculations. This chart shows curves of fuel rate as a function of engine rotational speed (usually given by revolutions per minute – rpm), for constant values of engine load[1]. This is graphical representation of the tabular dataset, which is convenient to use when carrying out calculations. Alternatively, dataset can also provide fuel rate as a function of engine rpm and torque, which is also convenient for practical applications.

Transmission influence on fuel consumption

Besides transmission impact on vehicle energy and fuel consumption through unwanted energy losses, second and more important influence of the transmission is the possibility of changing engine operating point by controlling gear selection for the same vehicle operating conditions, thereby potentially improving engine efficiency. For given vehicle operating point, engine operating point can be located theoretically anywhere on the appropriate constant-power curve in the scope of available engine torque and rpm limits (Figure 2a). Particular engine operating point is determined by current transmission gear ratio. An upshift results in an increase in the required engine torque accompanied by a decrease in engine speed for the same vehicle operating point. This may move the engine operating point towards a region of higher efficiency. Shifting strategy that is optimized for a certain vehicle driving pattern can therefore contribute to fuel savings (Guzzella, L. & Sciarretta, A. 2022).

IMPLEMENTATION OF FuelSim_v1.0 COMPUTER PROGRAM IN GNU OCTAVE PROGRAMMING ENVIRONMENT

Basic principles

Calculating of the fuel consumption for the steady-state vehicle motion (constant speed, constant wheel force and constant gear selected) is straightforward and generally does not require any aid of sophisticated computational tools. Knowing vehicle operating point and the transmission gear applied, engine operating point is determined by using relationships discussed in previous part of the paper. Therefrom, fuel rate value is determined using engine map data. This is then easily converted into some other appropriate unit, e.g., typically litres per 100 km.

Regarding driving cycles with operating points changing in time, approach becomes more demanding. As described by Guzzella, L., & Sciarretta, A. (2022), one possible solution is to use method described for steady-state drive, assuming averaged out value of engine efficiency. This is easy to implement, but gives very rough and unreliable consumption assessment. Next solution is forward physical modelling of longitudinal vehicle dynamics, integrating differential equations of motion starting from engine torque-rpm curves and using integrated driver closed loop control model in order to obey given speed profile in time. In other words, starting from the engine torque map and given load input, vehicle motion is determined. Then energy and fuel consumption can be obtained along with other results of such simulation. This is most correct approach from physical point of view, but most demanding to implement.

A compromise between these approaches is to use quasi-static method, that uses backward modelling approach. That means that, starting from the current vehicle operating point prescribed, current required engine operating point is determined. Method is applied by dividing driving cycle into many discrete short time intervals, so that partial quantities of the fuel consumed are determined for every of them by using steady-state approach described above. Total consumption is then obtained by summing these partial results. Implementation of this approach, if appropriate tailor-made software tool is not available, requires at least basic level of computer programming skills due to loops and branches required in general case to simulate vehicle negotiating given driving profile. Despite some characteristic limits and constraints that will be addressed in a subsequent section, this approach has been proven to provide very accurate and reliable computational results as stated by Dekraker, P. et al. (2018).

In the scope of this work, software tool named FuelSim_v1.0 was developed in GNU Octave version 9.4.0 programming environment, Eaton, J. W. et al. (2024).

GNU Octave programming environment

GNU Octave uses a programming language with syntax that is very similar to and highly compatible with MATLAB. It is free software distributed under the terms of the GNU General Public License (GPL), as published by the Free Software Foundation (Free Software Foundation, 2007). GNU Octave provides extensive support for numerical computations, matrix operations, interpolation, and data processing, making it well suited for the implementation of

[1] In order not to confuse "engine load" and "engine torque", what is meant by "load" here is the current position of the engine load control device, which can roughly be seen as an accelerator pedal position.

the proposed model. Its high compatibility with MATLAB also facilitates potential future migration of the developed code to the MATLAB environment.

Description of the model and algorithm

FuelSim is a computational model developed for estimating the fuel consumption of a road vehicle over an arbitrary driving cycle, defined by a vehicle speed–time profile and, optionally, a road-gradient profile. Although the general principles of the model are largely independent of the propulsion concept, the present version, FuelSim_v1.0, is specifically configured for a vehicle powered by an IC engine and equipped with a gearbox with finite number of gear ratios and a friction clutch. The model also incorporates a predefined gear-shifting strategy based on vehicle speed and engine load.

The required input data include vehicle parameters, transmission ratios and efficiency, engine torque characteristics, fuel-flow maps, upshift and downshift schedules, and the prescribed driving cycle. Engine torque and fuel-flow data are provided as functions of engine speed and load. The driving cycle is represented by discrete time–speed points, with road gradient optionally assigned to each point.

The calculation is performed sequentially over the driving cycle using a quasi-static approach. For each time step, vehicle acceleration and the corresponding inertia, rolling, aerodynamic and road gradient-resistance forces are determined. Their combination gives the required wheel force. Based on the currently engaged gear, the corresponding engine speed and required engine torque are then calculated, while interpolation of the engine map provides the associated engine load.

The calculated operating point is checked against the engine operating limits. If the required torque exceeds the maximum available engine torque, a forced downshift is performed where possible. Conversely, if the prescribed engine-speed threshold is exceeded, a forced upshift is applied. After each such gear change, the engine operating point and the gear-dependent inertia effects are recalculated.

For the final engine operating point, the instantaneous fuel mass flow is obtained by two-dimensional interpolation of the fuel-consumption map as a function of engine speed and load. Fuel consumed during each time step is calculated from the corresponding fuel mass flow and time-step duration and accumulated over the complete driving cycle. The resulting total fuel mass can subsequently be converted into conventional fuel-consumption measures, such as l/100 km.

Finally, the regular gear-shifting strategy is applied using predefined upshift and downshift curves. For the current engine load and gear, the corresponding shift speed is obtained by interpolation and compared with the current vehicle speed. If the respective criterion is satisfied, the selected gear is changed for the following calculation step.

The program was tested using several purpose-built driving cycles designed to verify various aspects of its functionality.

Model limitations

The present version of FuelSim is intended as a relatively simple and flexible tool for fuel-consumption assessment and therefore incorporates several simplifying assumptions. The model follows a quasi-static (quasi-steady-state) approach, which is based on determining required engine operating point based on the currently given operation point of the vehicle. Unlike a forward dynamic simulation, in which vehicle motion is obtained by integrating the equations of motion from the powertrain output and applied control inputs, the quasi-static approach determines the engine operating point required to satisfy a prescribed instantaneous vehicle state. In quasi-static model, the vehicle and engine operating conditions are evaluated at discrete time steps without modelling the transient behaviour of the powertrain. Consequently, effects such as engine torque response dynamics, turbocharger dynamics, and the duration and transient behaviour of gear shifts are not considered.

The transmission model is also simplified. Transmission efficiency is assumed to be constant, while clutch slipping losses during vehicle launch and gear shifts, as well as interruptions in power flow during shifting, are neglected. Hydrodynamic torque converters and continuously variable transmissions are not included in the present model.

The implemented regular shifting strategy assumes that upshifts occur during acceleration and downshifts during deceleration only. Furthermore, once the corresponding shift criterion is satisfied, the newly selected gear is applied from the following calculation step. These simplifications may not fully reproduce actual transmission or driver behaviour under all operating conditions. The discretisation of the driving cycle may therefore also affect the calculated engine operating points, gear-shift timing and, ultimately, the estimated fuel consumption.

Results example

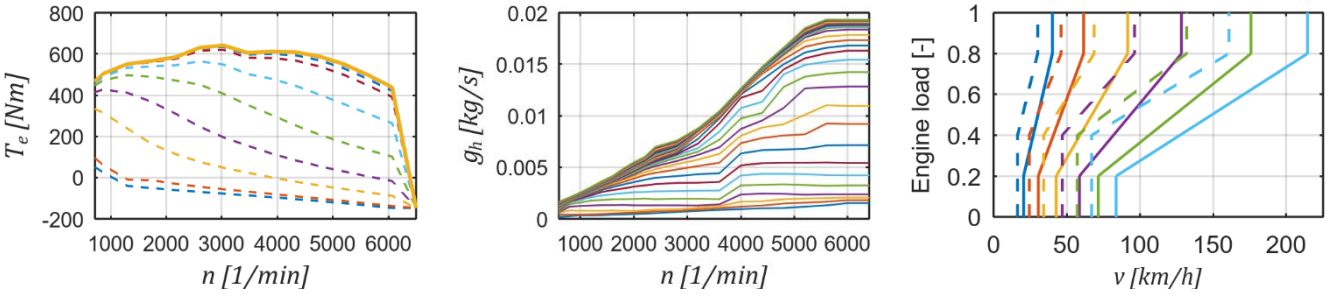
Vehicle data

Some of the program capabilities are briefly demonstrated by example shown herein. The input data used in this example do not represent any specific vehicle; instead, they constitute an illustrative dataset representative of a generic passenger car. Data are shown in the Table 1, as well as additionally in Figure 3.

Table 1 Vehicle data

Quantity	Label	Unit	Value
Vehicle mass	m	kg	1350
Frontal area	A	m^2	2.65
Drag coefficient	C_{Ax}	-	0.35
Wheel radius	r_W	m	0.33
Number of gears	-	-	7
Gearbox ratios	$i_{G,j}, j = 1 \div 7$	-	4.38, 2.86, 1.92, 1.37, 1, 0.82, 0.7
Final drive ratio	i_{FD}	-	3.98
Transmission efficiency	η_{Tr}	-	0.9
Gear shifting schedule	-	-	Figure 3 c
Engine torque map	$T_e(n, Load)$	Nm	Figure 3 a
Engine fuel consumption map	$g_h(n, Load)$	kg/s	Figure 3 b

Engine torque map (Figure 3 a) and fuel consumption map (Figure 3 b) are given as a function of engine rpm for a number of engine load values, as numerical tabular dataset. Since this data form is inconvenient to be represented, they are illustrated herein as charts based on these data. The same principle applies for gear-shift schedule (Figure 3 c), which is given as a series of curves of shifting speed vs. engine load, for all gears and for both shifting directions.



a) b) c)

Figure 3 Graphical representation of tabular datasets used for calculations: a) Engine torque map; b) Engine consumption map; c) Gear-shifting schedule (solid lines upshifts, dashed lines downshifts)

Driving cycle and simulation output examples

For the results demonstration, Worldwide harmonized Light-duty vehicles Test Cycle (WLTC) Class 3 (United Nations Economic Commission for Europe [UNECE], 2025) has been chosen. It is implemented in the scope of Worldwide Harmonised Light vehicles Test Procedure (WLTP), which represents widely adopted standard for determination of fuel consumption and the levels of pollutants emission for vehicles with various types of the powertrain (UNECE, 2025).

Firstly, distance travelled is calculated by the program. Result obtained is:

S = 23.266 km – total distance travelled

This corresponds to the distance declared by (UNECE, 2025).

Main fuel consumption results are:

m_F = 1.65 kg – total mass of fuel consumed

$V_{l/100km}$ = 8.86 l/100 km – average fuel consumption

Figure 4a shows speed profile of the WLTC Class 3 which is used as simulation input for the vehicle driving profile. Other parts of the Figure 4 show changes of several selected parameters during cycle duration:

- Figure 4b: total mass of fuel consumed.
- Figure 4c: gear selected.
- Figure 4d: engine load (roughly corresponding to accelerator pedal position).
- Figure 4e: engine rpm

Detailed discussion and interpretation of results represented herein will be omitted. Their purpose is just to provide preliminary insight in overall simulation features. Based on numerical values and visual appearance of time curves, it can be empirically assessed that simulation results exhibit physical plausibility and consistency with the expected qualitative behaviour of the simulated vehicle. Detailed verification and validation are, however, left to the next phase of the research.

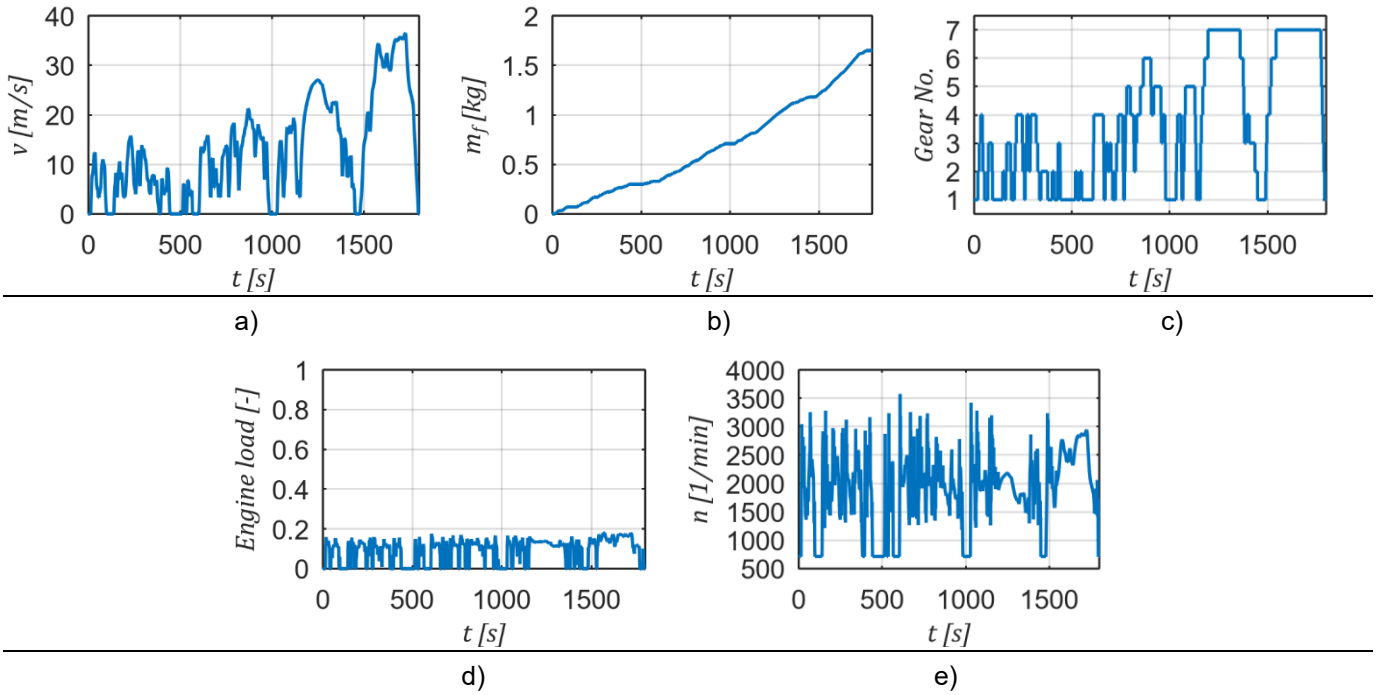


Figure 4 Simulation results: a) WLTC speed profile, b) mass of fuel consumed, c) gear selected, d) engine load (roughly corresponding to accelerator pedal position), e) engine rpm.

CONCLUSIONS

FuelSim_v1.0 provides a functional and flexible educational and research framework for quasi-static estimation of vehicle fuel consumption over arbitrary prescribed driving cycles. Its externally configurable vehicle, engine, transmission and driving-cycle data allow the same computational framework to be applied to a broad range of comparative studies.

The implemented approach exhibits certain limitations that should be taken into account by users, and that result from both intrinsic features of the applied modelling technique itself and some simplifications adopted for easier model implementation. Main limitations of the quasi-static modelling technique are among others inability to capture transient phenomena and possible susceptibility to the temporal resolution of the driving cycle. Further model limitations are mostly connected with some simplification in transmission behaviour representation, e.g., omitting of the clutch slip impact and somewhat simplified implementation of the gear shifting schedule.

In this work WLTC driving cycle was used to demonstrate some of the simulation outputs. Results obtained generally show proper function of the program. The numerical values and time histories obtained appear physically plausible and consistent with the expected qualitative behaviour of the simulated vehicle. However, formal verification and validation of the model is most important next step in forthcoming research. Other future development prospects can include e.g.:

- Formal verification/validation and benchmarking.
- Improved transmission, braking/fuel-cut-off and energy-loss modelling.
- Extension to additional powertrain architectures.
- Possible transition towards higher-fidelity forward dynamic simulation.

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