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Model for developing an eco-driving strategy of a passenger vehicle based on the least fuel consumption

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ABSTRACT

Additional consumption of fuel in an intense traffic condition is inevitable. Excess fuel consumption may be avoided, if an optimal driving strategy is implemented subject to the surrounding condition of a vehicle and existing constraints. Development of an optimal driving strategy has been the subject of eco-driving. A model of optimal driving strategy has been developed and it has been applied for assessment of eco-driving rules. The model may be categorized as an optimal control and the objective function is minimization of fuel consumption in a given route. Vehicle speed and gear ratio are identified as control variables. The effect of working load has been considered according three engine running processes of Idle, part-load and wide open throttle. The model has then been applied to identify the optimal driving strategy of a vehicle in different traffic congestion based on eco-driving rules.

1. Introduction

Fuel consumption in transport sector contributes to the emission of greenhouse gases in general and CO_2 in particular. Emission of CO_2 is a function of state of the vehicle technology and the traffic flow. Improvement in energy efficiency and the design of vehicles has reduced fuel efficiency of vehicles considerably over last three decades. But number of vehicles has increased in different parts of the world in the period 1980–2007. Rapid economic growth in many developing countries and increased welfare of population in industrialized nations has led to expansion of car industry world wide. Number of vehicles on the road has increased considerably and traffic congestion is observed in major cities in the world, including both developing and industrialized countries.

Traffic intensity has increased in urban transport systems and driving pattern over a specified route varies to great extent. High traffic congestion and variation of driving pattern has resulted in excess fuel consumption. Excess fuel consumption has reduced the effectiveness of energy efficiency of vehicles. Contribution of improved energy efficiency of vehicles to reduction in fuel consumption and emission of greenhouse gases has therefore been limited. Hence, optimization of driving behavior and developing an optimal driving strategy subject to the surrounding condition of a vehicle and the traffic flow is being considered as an important element in the process of improving the energy intensity of transport sector.

Driving strategy has been introduced as a key rule in eco-driving instructions [1–3]. The objective of eco-driving has been to ease the condition of local environment through improvement of driving behavior and local traffic flow. Implementation of eco-driving leads to the improvement of quality of local environment and reduces fuel costs, costs for maintenance through reducing aggressive driving and maintaining a steady speed.

The idea of eco-driving is not new and its integration into car driving courses was introduced as far back as 1993 [4]. Many companies had long before recognized the value of training their drivers to save fuel. The first curriculum entitled 'training for environmentally conscious and driving behavior during driving instruction' was published by the technical University of Berlin (in conjunction with the German DOE and Volkswagen); this was later integrated into the curriculum 'practical car training' developed by the German Federation of Driving Instructor Associations [4].

Eco-driving in high density traffic flow is extremely affected by accelerating smoothly, decelerating gradually and also changing gear as early as possible at a modest engine speed [5]. Accordingly, aggressive driving based on sudden acceleration and deceleration results in fuel wastage of approximately 33% at high speeds in the highway and about 5% around towns [6]. Maintaining an efficient speed is an effective means of keeping mileage up. In addition to that, optimal energy efficiency can be expected while cruising with no stops, at minimal throttle and with the transmission in the highest gear. Therefore, acceleration should be quite gentle.

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Many analytical tools have been developed for studying driving strategy and its contribution to the implementation of the objectives of eco-driving [7–9]. Some kind of mechanistic emission and fuel consummation simulators consider speed as the only parameter for explaining driving strategy [10–11]. It is assumed in all cases that speed of the vehicle is major function of the vehicle performance, route characteristics and traffic flows. Observation whit the case study of driving cycle indicates that the selection of a convenient gearing together with the speed of the vehicle determines the pattern of fuel consumption. This finding has been considered as a foundation of the eco-driving plan which leads to high energy efficiency and low emission of pollutants.

Identification of optimal driving strategy has been the subject of the present work. A model of optimal driving strategy has been developed and it has been applied for identifying the appropriate means of achieving the objectives of eco-driving. Controlling speed and selecting an appropriate gear ratio with respect to different engine loads have been considered as decision variables. Theoretical concept of the model shall be presented in the next section and it shall be followed by demonstration of the application of model for different traffic flows.

2. Theoretical concept

The model has been founded on the micro economic theory of production of services and it has been tailored to identify the short term operation of specified vehicle or a transport technology [12]. The objective of the model has been to minimize the operation costs or fuel consumption of a certain technology in a traffic flow. Implementation of this concept has help to identify optimal driving strategy while the technology is considered as fixed.

Fuel consumption had been considered as a function of speed in the previous version of the model. Such an assumption has had drawbacks and the effect idle working of a vehicle on the fuel consumption has hardly been studied in heavily congested traffic flow. It has been difficult to apply the model for analysis of eco-driving rules. The model has therefore been improved and fuel consumption has been considered as a function of the speed, gear ratio and engine load. Therefore, the basics of the model shall be summarized in this section and constraints representing the additional aspects relating to the features of eco-driving shall be presented.

2.1. Representation of driver behavior based on speed and gear ratio

Driver is usually forced to accelerate, decelerate, brake and change gear more often in a heavy traffic during rush hour. Such a process has significant impact on the fuel consumption. Fuel consumption is in part determined by degree of acceleration resistance. This occurs when the speed of a vehicle is raised. To put a vehicle into motion or to raise the speed is involved with the use of energy while the acceleration resistance is zero at a constant road speed. The acceleration resistance depends on the increase in speed, changing gear ratio and the total weight of the vehicle. Driver's behavior is reflected in changes in vehicle speed and gear ratio. Speed and gear ratio at each time point depend on acceleration or deceleration. Any action that prevents unnecessary acceleration and deceleration could be undertaken with help speeding up and gear ratio. Therefore, control of speed and gear ratio is important for developing optimal driving strategy.

Speed of vehicle is subject to the total forces acting on the vehicle and it is indicated by

$$kM\frac{dv}{dt} = F_T - \sum F_R \tag{1}$$

where, M, F_T and F_R are mass of vehicle, traction force and motion resistance forces respectively. k is the weight factor which equals to $\frac{M+M_e}{M}$ and M_e is the mass of rotating components. The resistance force is the summation of total resistance such as aerodynamic, rolling and gradient. Traction force is the outcome of energy conversion in engine and it depends on the technical features of the vehicle which can be defined as a function of torque (τ), gear ratio (i), dynamic radius (r) and the transmission efficiency (η_t). Torque can be estimated through the third order equation of angular speed (ω) according to the relationship

$$\tau = a + b\omega + c\omega^2 \tag{2}$$

a, b, c and d are the constant coefficients that may be defined through analysis of engine performance curve (torque–rpm).

The speed of the vehicle can be determined through solving first order differential equation (2) by the following format

$$v = \frac{\left[\tan(0.5t\sqrt{4\alpha_3\alpha_1 - \alpha_2} + Arc\tan\frac{\alpha_2}{\sqrt{4\alpha_3\alpha_1-\alpha_2}})\right]\left[\sqrt{4\alpha_3\alpha_1 - \alpha_2} - \alpha_2\right]}{2\alpha_1} \tag{3}$$

where

$$\alpha_1 = \frac{1}{kM}\left(\frac{c\eta_t i^3}{r^3} - 0.5\rho_a A_v C_D v_a^2\right) \tag{4}$$

$$\alpha_2 = \frac{b\eta_t i^2}{kMr^2} \tag{5}$$

$$\alpha_3 = \frac{a\eta_t i}{kMr} - \mu g - g\sin\beta \tag{6}$$

$$\text{And also, } \alpha_2 \leqslant \sqrt{4\alpha_3\alpha_1} \quad \alpha_1, \alpha_3 \geqslant 0 \tag{7}$$

where μ, β, v_a, ρ_a, C_D and A_v are the rolling resistance coefficient, the road slip angle, the air speed, the air density, the drag coefficient and the frontal area of vehicle, respectively. Eq. (3) reveals that route conditions influence the speed of vehicle through road surface and inclination. The road surface has an effect on rolling resistance. Rolling resistance is influenced by the weather condition, which is therefore determined by combination of different factors. But it is also affected by physical factors such as the weight of the vehicle, tire profile and pressure and wheels positions. The influence of weather condition on fuel consumption is mainly determined by three factors: air resistance, temperature and precipitation where air resistance is more important than others. Air resistance is dependent on the shape of the vehicle which is represented through the frontal flat area (A_v) and drag coefficient (C_D) in the speed profile equation.

It can be concluded that Eq. (3) manifests the relationship between the quality of movement of vehicle and required force to move it. It is also evident that gear should be set at the right level if the fuel consumption is to be minimized. Hence, identification of the relationship between the speed and gear ratio is an important factor of the eco-driving.

It will be hard to drive consistently in an urban traffic where the vehicle should join and leave the traffic flow more often. It may be necessary to accelerate stronger to leave the traffic flow when turning into a major road or merging with a faster lane. In contrast, it may be required to decelerate when the vehicle is to join the intense traffic flow. This phenomenon is influenced by driving style which requires more flexibility and quick response by the driver. Impact of joining and leaving the traffic flow on driving strategy can be considered on the basis of the fundamental theory of a standard traffic flow. According to this theory, a traffic flow can generally be divided into three homogeneous states (1) that of free driving, (2) that of bunched driving, and (3) that of standing. The contribution of the traffic states within any congested traffic situations can then be easily obtained from the functional relationship between three parameters: the mean speed (v), the traffic flow rate (φ), and the traffic density (ρ) which is called fundamental diagram [6–7]. An equilibrium relationship $\varphi = \rho v$ is identified for each traffic flow. When the probabilities for the individual states are given, the shape of the ρ–v relationship can be constructed. The fundamental diagram can be described completely by five essential parameters:

- The desired speed v_0.
- The net gap τ_{ko} within a fluid convoy.
- The mean speed v_{ko} of the fluid convoy.
- The net mean gap τ_{go} of the jam convoy.
- The maximum jam density $\rho_{\max}$.

Equilibrium relationship based on the above parameters yields the maximum allowable velocity of the traffic flow, (V_T) (see Ref. [13–14]). Permitted speed can be considered as upper bound on the vehicle speed while the vehicle is moved through specific traffic flow.

2.2. Interrelationship between fuel consumption and speed and gear ratio

Traction force is initiated through conversion of chemical energy and transmission of power via differential and gearbox. Gear ratio indicates the transmission of power from engine to the wheels. Generation of engine power depends on the engine performance, kinetics of combustion reactions and heat loss from cylinder and exhaust pipe. The process of power generation and transmission may be represented with the help of energy flow diagram (EFD) of power train of a passenger vehicle [15–17]. EFD can be depicted as a series and parallel combination of system blocks (control volumes). The performance of engine is a function of the mechanical operation of the vehicle during four strokes: intake, compression, combustion and exhaust. The stages of the energy conversion processes are presented as virtual system blocks which indicate the four strokes and interaction of energy and fluid flow through various parts of the vehicle system. Fig. 1 shows the energy flow diagram in the power train system. This approach to the vehicle power train system provides means of segregating it into sub-layers and each sub-layer may be represented by mass and energy equations. The equations reflect mass and energy conservation for each control volume and they may be conceptualized as follows:

$$\frac{dE}{dt} = \left.\frac{\partial E}{\partial t}\right|_{cv} + \int_{cs} e\rho u \, dA \tag{8}$$

where E, e, ρ, u and A are defined as flow of energy through a control volume, energy intensity through a control volume, fluid density, fluid speed and surface area of each control volume.

Mass and energy conversion constraints are then integrated into a simultaneous set of equations. Interrelationship between fuel and energy flow shall be considered according to the following relationship:

$$m_{ft} = \frac{E_{Ht}}{NCV} \tag{9}$$

where m_{ft} and NCV are defined as fuel flow rate at time period t and net calorific heat value of fuel. E_{Ht} represents the chemical energy of fuel as a result of combustion reactions. Application of the concept of control volume and interaction between system blocks of EFD (Fig. 1) provides means of estimating E_{Ht} through coupling following energy balance constraints

$$E_{Tt} = F_T \cdot x_t = (F_R + F_a)x_t = (F_R \cdot x_t) + \frac{M v_t^2}{2} \tag{10}$$

$$E_{Tt} - E_{Pt}\eta_t = 0 \tag{11}$$

$$E_{Ht} = \sum_j E_{Pjt} + \left(\sum_j E_{\alpha jt} + \sum_j E_{\gamma jt} \right) \eta_e + E_{Ft} + E_{ext} + E_{Wt} \tag{12}$$

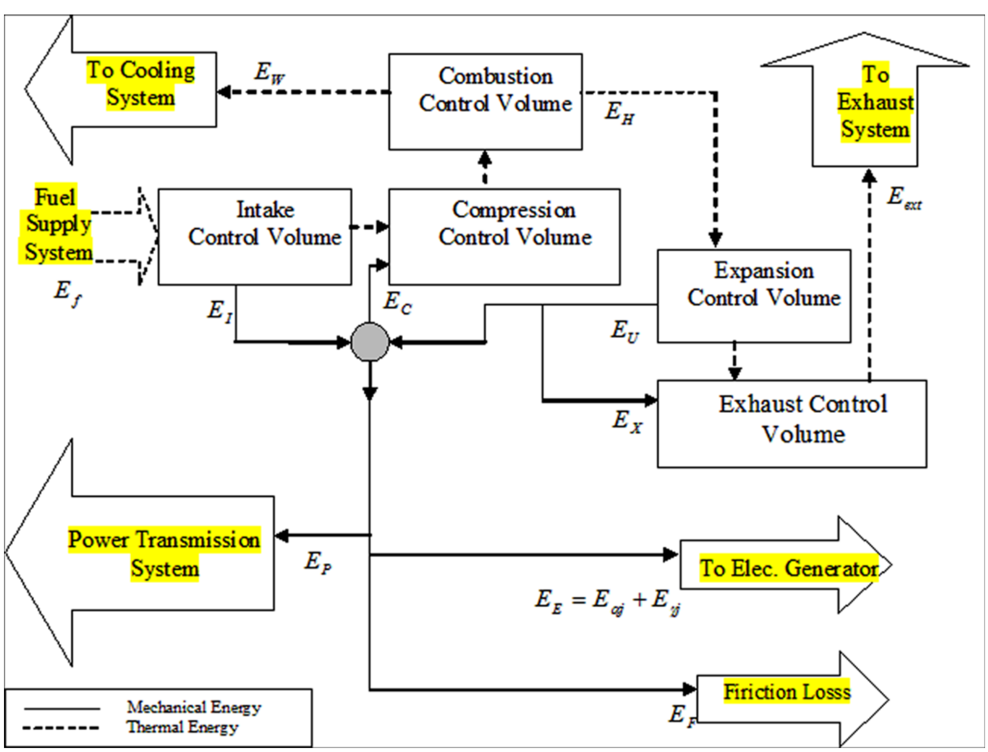

Fig. 1. Energy flow diagram in the power train system.

where E_{ext} is the exhaust energy loss based on exhaust gas temperature at time period t. E_{Wt} is the thermal energy loss from cylinder wall which could be calculated by Annad equation based on physical properties of working fluid and wall temperature [18]. $E_{\alpha jt}$ is the operative electricity consumption in the adiabatic process (i.e. turbo machineries). $E_{\gamma jt}$ is the given non operative electricity consumption in the adiabatic process (i.e. cabin components). E_{Ft} is the friction energy loss of cylinder during movement of the piston in the adiabatic process. E_{Tt} is the required motion energy or traction at time period t.

2.3. Impact of engine load on fuel consumption

Actual performance of a vehicle on the road indicates that major portion of a journey is associated with part-load engine operation. Part-load covers about 20–80 percent of engine load when the engine speed is increased gradually. Part-load operation can be improved when fuel–air mixture is diluted with excess air. Dilution of fuel–air mixture improves the eco-driving through fuel conversion efficiency. Excess fuel consumption is also observed in an intense traffic flow where the engine runs idle for longer time than anticipated. The effect of idle engine running on excess fuel consumption shall be considerable when the driver spends much of his/her time in standing traffic flow. It is also observed that many drivers tend to use high traction power in rapidly varying traffic flows when they join a free traffic flow while the engine is run in *WOT*[1] condition. The effect of engine load on fuel consumption during idle and *WOT* conditions is shown in Fig. 2. It is observed in the figure that the equivalence ratio (φ) or air–fuel ratio (AFR) is influenced by idle running (0–20% of engine load) and *WOT (80–100%* of engine load).

The effect of *WOT* condition on fuel consumption shall be represented in the model by following constraint:

$$m_{f,WTO} = \frac{V_{tot}}{\rho_a(1+\alpha_m)} \tag{13}$$

where $m_{f,WTO}$, V_{tot} and α_m are defined as maximum fuel consumption in *WOT* condition, total design volume of the engine and minimum air–fuel ratio (generally equals to 12). It is assumed that the total volume of engine is filled by air–fuel mixture in a *WOT* condition. Assumption on fuel consumption when the engine is running idle shall be based on empirical data. The experimental data indicate that fuel consumption per hour in an idle case would be 1.87 l for engine rotation of 500–700 rpm and 2.8 l for 900–1200 rpm [19]. Therefore, effect of engine load on fuel consumption shall be considered according to the following constraints:

$$\begin{cases} m_f = m_{f,\mathrm{Idle}} & v_t = 0 \\ m_{ft} = f(v_t, i_t) & v_t < v_m \\ m_{ft} = m_{f,WTO} & v_t \geqslant v_m \end{cases} \tag{14}$$

where v_m represents the possible maximum value of speed when the gear is selected. It can be observed from Fig. 2 that, the AFR is near to a constant value 13.4 (φ = 1.1) while engine is run in part-load condition.

Analysis of the direct impact of engine load on emission of greenhouse gasses shall be based on the k-value of combustion reactions. Six major combustion products are considered which are: CO_2, CO, H_2, O_2, H_2O and N_2. The totality of reaction chains shall be represented through equilibrium chemical reaction as follows

[1] Wide open throttle.

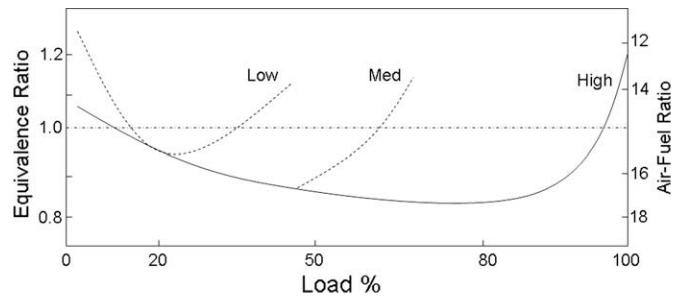

Fig. 2. Effect of engine running in idle and *WOT* condition on fuel consumption.

$$x_{C_8H_{18}}C_8H_{18} + x_{C7_8H_{16}}C_7H_{16} + x_{C_7H_8}C_7H_8 + \frac{1}{\phi}m_s(O_2 + 3.76N_2) \rightarrow aCO_2 + bH_2O + cN_2 + dO_2 + eCO + fH_2 \tag{15}$$

where $x_{C_8H_{18}}, x_{C7_8H_{16}}$ and $x_{C_7H_8}$ are given as mole fraction of the fuel components. m_s is given stochastic ratio of number of moles of oxygen to number of moles of fuel. And a–f represent the mole fraction of the combustion products which must be calculated on the basis of the amount of consumed fuel (Eq. (10)). Mole fractions of the combustion products shall be determined with following set of equations:

1. Carbon balance:

$$8x_{C_8H_{18}} + 7x_{C7_8H_{16}} + 7x_{C_7H_8} = a + e \tag{16}$$

2. Oxygen balance:

$$9x_{C_8H_{18}} + 8x_{C7_8H_{16}} + 4x_{C_7H_8} = b + f \tag{17}$$

3. Hydrogen balance:

$$\frac{m_s}{\phi} = a + \frac{b}{2} + d + \frac{e}{2} \tag{18}$$

4. Nitrogen balance:

$$\frac{3.76m_s}{\phi} = c \tag{19}$$

5. Equilibrium coefficient of water dissociation at combustion temperature:

$$K_{H_2O} = \frac{n_{H_2O}}{n_{H_2}\left[\frac{n_{O2}}{n_T}\right]^{1/2}} \tag{20}$$

6. Equilibrium coefficient of carbon dioxide dissociation at combustion temperature:

$$K_{CO_2} = \frac{n_{CO_2}}{n_{CO}\left[\frac{n_{O_2}}{n_T}\right]^{1/2}} \tag{21}$$

7. Total mole balance:

$$n_T = a + b + c + d + e + f \tag{22}$$

K_{H_2O} and K_{CO_2} are given at constant temperature of combustion of gasoline (2400 °C). For a given φ, based on the engine load, amount of the combustion products shall be calculated through Eqs. (16)–(22).

3. Optimal control approach for eco-driving

The criterion for optimality would be the least fuel consumption during a specified movement of the vehicle. Fuel consumption over drive-cycle is therefore minimized while the demand for an urban transport service is met. This service is expressed as distance traveled by a passenger (p-km) and shall be represented by following set of equations:

$$\begin{cases} Q = \sum_t px_t \\ \frac{dx}{dt} = v \end{cases} \tag{23}$$

where Q is transport services, p is number of passengers.

Therefore, the complete structure of the model may be described as follows:

$$\sum_{t=0}^{t=\Delta t} m_{ft}(v_t, i_t)\Delta t \rightarrow \min \tag{24}$$

Subject to satisfying following constraints:

(1) Speed profile: (3)–(6).
(2) Energy flows: (8)–(12).
(3) Engine load: (13) and (14).
(4) Combustion products: (16)–(22).
(5) Demand for transport service: (23).
(6) Gear ratio selection: $i_t \in [i_f, i_{1st}, i_{2nd}, i_{3rd}, i_{4th}]$ i_f means freewheel

With following initial and boundary conditions:

$$x(0) = x_0 \quad v(0) = v_0 \tag{25}$$
$$x(T) = x_T \quad v(T) = V_T \tag{26}$$

Constraint on gear ration shall be discussed further. Selection of gear ratio is usually based on vehicle speed variation. The interrelationship between speed and gear ration is included according the following modes of movement in different traffic conditions:

- *First mode.* Putting vehicle into motion:

$$0 < v_t < v_m^1 \Rightarrow i_t = i_{1st} \tag{27}$$

- *Second mode.* Part-load running of engine in the traffic flow:

$$\begin{cases} v_t < v_m^1 & \text{and} \quad v_m^1 \leqslant V_T \Rightarrow i_t \in [i_{1st}, i_{2nd}] \\ v_m^1 < v_t < v_m^2 & \text{and} \quad v_m^2 \leqslant V_T \Rightarrow i_t \in [i_{2nd}, i_{3rd}] \\ v_m^2 < v_t < v_m^3 & \text{and} \quad v_m^3 \leqslant V_T \Rightarrow i_t \in [i_{3rd}, i_{4th}] \\ v_m^3 < v_t < v_m^4 & \Rightarrow i_t = i_{4th} \end{cases} \tag{28}$$

- *Third mode.* Idle running of engine in a standing traffic flow:

$$V_T = 0 \Rightarrow v_t = 0 \Rightarrow i_t = i_f \tag{29}$$

- *Forth mode. WTO (full-load)* running of engine when the vehicle is joining a free flow traffic:

$$\begin{cases} v_t \geqslant v_m^1 & \text{and} \quad v_m^1 \leqslant V_T \Rightarrow i_t = i_{1st} \\ v_t \geqslant v_m^2 & \text{and} \quad v_m^2 \leqslant V_T \Rightarrow i_t = i_{2nd} \\ v_t \geqslant v_m^3 & \text{and} \quad v_m^3 \leqslant V_T \Rightarrow i_t = i_{3rd} \\ v_t \geqslant v_m^4 & \text{and} \quad v_m^4 \leqslant V_T \Rightarrow i_t = i_{4th} \end{cases} \tag{30}$$

The optimal control problem is then formulated as a constrained optimization programming with mixed integer nonlinear constraints (MINLP[2]) and DICOPT[3] method is used to solve the MINLP problem though following format [20]:

$$Min\, f(x, \alpha) \tag{31}$$

Subject to

$$G_i(x_j, \alpha_j) + H_i y_i \sim b_i \tag{32}$$
$$l \leqslant x_j \leqslant u \tag{33}$$
$$y_i \in [0, 1] \tag{34}$$
$$x_j, \alpha_j \geqslant 0 \tag{35}$$

$f(x, \alpha)$ represents the objective function of the model. x is the state variable of the model which is defined as mass and energy flows. Set of control variables (i.e. v_t and i_t) are represented by α where i_t is a discrete control variable. $G(x, \alpha)$ represents set of constraints of the model. H is defined as a constant and its value is considered as a predefined gear ratios in manual gear box. y is the binary integer variable of the model and can be selected through $y \in [0,1]$. Therefore, the term of Hy shall be defined on the basis of combination of the value of gear ration and y according to the aforementioned four modes. Hy can then be used in Eqs. 3, 4, 5, 6 and (10) in order to determine the value of the traction energy and speed profile. Following constraint has also been included to prevent selection of the same gear ratios at each time period

$$\sum_{n=1}^{N} y_n = 1 \tag{36}$$

where, N is changed from 1 to 4.

4. Results and discussion

The model that has been developed in the present work has been applied for analysis of the impact of eco-driving on fuel consumption and emission of greenhouse gasses in different intense traffic flows. Five scenarios have been design on the basis of the movement of a vehicle in a specified traffic conditions in capital city of Iran (Tehran). The scenarios are as follows:

Scenario (1) Jam convey traffic flow in a crowded urban highway.
Scenario (2) Fluid traffic flow in a free urban highway.
Scenario (3) Bunched convey traffic flow in a high crowded urban highway.
Scenario (4) Bunched convey traffic flow in an inclined urban street (β = 15%).
Scenario (5) Jam traffic flow in an urban street.

Parameters of the abovementioned traffic flows are described in Table 1. The information on the traffic flow has been obtained from Tehran Traffic Control Center (TTCC) and the maximum possible speed of traffic is then estimated on the basis of the fundamental diagram. A specific vehicle has been selected for the scenario analysis. The technical specification of the selected car is shown in Table 2.

Rolling resistance is considered to be 0.013 on the basis of asphalt properties and type of the vehicle tire. The wind speed effect is assumed negligible.

4.1. Impact of shifting gears

Speed profile has been obtained and it may be observed in Fig. 3 as the main result of the model. The figure shows the speed profile in each scenario. The model results indicate that the changing up heavy gears from 1st gear to 3rd gear should be followed immediately. This has led to lower engine speed and also lower friction losses. These losses increase with engine speed. The losses are limited when engine speed remains low. Maintenance of such a situation leads to efficient power generation. It shall therefore be an efficient operation during acceleration, if gear is shifted up as soon as possible (at low engine speed). The results show that the chang-

[2] Mixed integer nonlinear programming.
[3] Discrete and continues optimizing.

Table 1
Parameters of the different traffic flows.

Scenario	1	2	3	4	5
Mileage (m)	960	1225	2338	505	1185
Measured traffic density (pc/km)	40	10	60	60	82
Measured net gap within a fluid convoy (s)	1.6	1	1.4	1.6	1.8
Measured net mean gap of the jam convoy (s)	1.2	1.1	1.2	1.2	1.2
Estimated maximum traffic speed (km h^{-1})	46.75	110	76	55	30

Table 2
Design specification of the case study.

Vehicle type	Peugeot 306	Drag coefficient	0.3
Engine type	XU7JP/L3	Tire dimensions	185/65 R15 88H
Net weight	1200 kg	Fuel properties	Petrol with 95 RON
Wheel base	2670 mm	Axle ratio	4.1: 1
Front width	1720 mm	Gear ratios	1st 3.75:1
Font height	1980 mm		2nd 1.85:1
			3rd 1.25:1
			4th 1:1

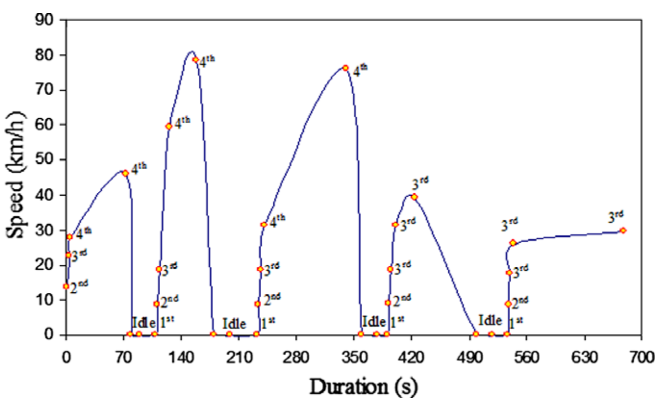

Fig. 3. The trajectories of vehicle speed estimated with the help of the model.

ing up should be less than 7 s for all five scenarios. This will be independent of traffic conditions. The engine speed varies from 1174 rpm in a heavy traffic flow (Scenario 5) to 2689 rpm in a free traffic flow (Scenario 2). Therefore, average engine speed during the changing up is estimated 1930. The results of the model indicate that the process of acceleration should be associated with changing up from 3rd gear to 4th gear. Driving should then continue at 4th gear.

4.2. Impact of engine loads

The impact of variation in engine loads on the fuel consumption is then studied on the basis of AFR which is embedded in the model. It may be explained by the fact that the average car only requires 5 kW of power to drive at a steady speed of 50 km h^{-1}. The remaining 80% (or more) of the engine's power is only needed to overcome the resistances due to acceleration. The accelerator pedal operates the throttle/butterfly valve and consequently, the AFR. When the engine load is increased by pressing the accelerator pedal, the accumulated fuel consumption is increased while the traction power is maintained at the high speed. This situation is more appropriate when one drives the vehicle in a free way traffic flow (Scenario 2). Application of high traction power in an intense traffic flow would lead to excessive fuel consumption because it would not be possible to achieve high speed in the 3rd or 4th gear. Therefore, excess fuel shall be consumed due to sudden decelerations and many stops. The results of the model indicate that the maximum speed is reached at 78.26 km h^{-1} in a fluid traffic flow in a free urban highway (Scenario 2). Such a high speed enables useful conversion of 82.9% of total engine load (the maximum engine power equals to 75 kW). Engine load is lower than this value in other scenarios which represented high intense traffic flows. Therefore, operation of the engine in a condition of *WOT* is not recommended. The model estimate of maximum engine power is shown with black circle indicators in Fig. 4. It can be observed in Fig. 4 that the value of maximum power required for a selected gear is lower than its maximum value at *WOT* condition (i.e. 75 kW). It has been found that the optimal range for engine loading is achieved from 36% in the jam traffic flow in an urban street (Scenario 5) to 83% in a fluid traffic flow in a free urban highway (Scenario 2). This means that a gentle acceleration with less aggressive driving is preferred which is compatible with the objective of the eco-driving.

4.3. Impact of Idle running

The case of idle running of the engine is studied with a constant engine speed of 700 rpm which is equivalent to an engine torque of approximately 107 Nm. The duration of idle running is shown in Fig. 3. The effect of idle engine running is then attributed as inevitable additional fuel consumption of about 0.33 l during 276 s.

4.4. Optimal SFC[4] (g/kW)

The optimum Specific Fuel Consumption (SFC) in each scenario is estimated. The results of the model are observed in Table 3. It can be seen in Table 3 that the optimal SFC may be achieved through driving in a free traffic flow (Scenario 2) with a constant speed of 78.26 km h^{-1}. The emission of CO_2 and its intensity will also be the least at optimal SFC and the thermal efficiency could be the highest at the optimal speed of 78.26 km h^{-1}.

4.5. Optimal predicted driving strategy

The specification of an optimal driving strategy in an intense combined traffic flow is described in Table 4. The specification of the driving strategy is based on three major parameters: maximum engine load, maximum speed in light selected gear ratio and minimum speed in heavy selected gear ratio. It is expected that the maximum vehicle speed reaches to the maximum allowable speed gradually by using light gearings. However, the vehicle speed may be limited by aerodynamic resistance and also degree of inclination. The optimal driving strategy indicates that low speed is maintained with heavy gearings. The maximum engine load is achieved while traveling in a highway with free and bunched convey flows. Optimal driving strategy also indicates that the maximum engine load should not be understood as generating excess torque by using heavy gears when the vehicle moves towards uphill. An improvement of vehicle speed in uphill bunched convey street can be observed in Table 4.

Comparison of the model results, real cycle and standard NEDC in similar conditions for driving cycle has been depicted in Table 5.

As the table shows, maximum deceleration in real cycle and model results is very close together. It indicates that drivers use more traction power when the join to free or bunched traffic flow and therefore, more energy will be lost through several decelerations.

Also, average speed in real cycle and based on the model results is smaller than NEDC because several stop-go driving.

[4] Specific fuel consumption.

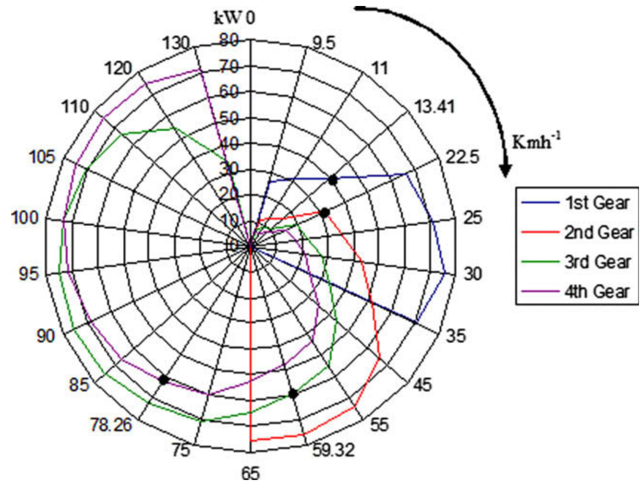

Fig. 4. Maximum engine power in each selected gear based on the model results.

Table 3
Comparison of main eco-driving parameters based on optimal results.

Scenario number	1	2	3	4	5
SFC (g/kW h)[a]	261.64	229.5	251	260.2	293
Fuel efficiency (l/100 km)	10.1	8	9.6	9.8	13.7
CO_2 intensity (g/km)	160	139	144.15	156.5	161.3
Thermal efficiency (%)	30.6	34.8	31.9	30.7	27.3

[a] Including excess consumed fuel during idle engine running.

Table 4
Optimal driving strategy in an intense combined traffic flow.

	Jam convey highway	Free flow highway	Bunched convey highway	Bunched convey street[b]	Jam crowed street
Max engine load (%)	52.1	82.9	80.8	50.2	35.8
Max speed (km h^{-1})	45.9 (4th)[a]	78.26 (4th)	76 (4th)	39 (3rd)	29.5 (3rd)
Min speed (km h^{-1})	13.4 (1st)	9 (1st)	8.7 (1st)	9 (1st)	8.6 (1st)
Max engine power (kW)	39.1 (1st)	62.2 (4th)	60.6 (4th)	37.7 (3rd)	26.9 (3rd)
Min engine power (kW)	24.8 (3rd)	22.73 (1st)	22.8 (1st)	24.2 (1st)	22.9 (1st)

[a] 4th gear is selected.
[b] Uphill running (15% inclinations).

Table 5
Comparison of model results with real cycle and standard NEDC.

Cycle	Ave acc. (m/s^2)	Max dec. (m/s^2)	Ave speed (km h^{-1})	Max speed (km h^{-1})	Dist. (km)	Duration (s)
Real[a]	0.45	−2.21	29.3	84	15.9	1955
NEDC	0.54	−1.39	32.23	120	10.9	1220
Model	0.39	−2.09	30	83	6.2	744

[a] Information about real cycle is collected from Tehran Traffic Control Center (2006).

5. Conclusion

The objective of this research work has been to introduce an optimal eco-driving of passenger vehicle based on the minimum fuel consumption. The results indicate that implementation of optimal driving strategy based on coordination of speed and gear ratio through engine load would lead to minimization of fuel consumption in an intense traffic flow. The results show that optimal driving strategy can be obtained with engine loading in the range of 36% in the jam traffic flow in an urban street to 83% in a fluid traffic flow in a free urban highway. The optimal speed has been identified at rpm of 1930.

The speed profile is defined on the basis of optimal driving strategy. So, the model would be introduced as a new challenger to predefined eco-driving tests such as NEDC[5] and ARTEMIS[6] .These standard tests have been developed as a harmonized emission model for transport sector which are used to estimated vehicle fuel consumption in laboratory conditions. Application of the model results would enable to identify the optimal pattern of fuel consumption in a real cycle. The model provides an alternative approach to predefined standard test cycles.

The results of the model indicate that the potential of saving fuel is about 1.5 L/100 km in a mixed intense traffic flow represented by five scenarios of the case study. The saving in fuel is hypothetical and subject to the assumption that driver will behave rationally. Therefore, a policy may be put in place to encourage the drivers to behave in a way that is compatible with the results of the model. Implementation of such a policy could be possible when a logical controller PLC board is developed and applied according to the fundamentals of the model. Such a controller can be connected to the traffic navigation system to control the speed and provide the driver with appropriate alarms when changing a gear is necessary. This idea is recommended as a future work for further development of the model.

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