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Computer Simulation of Light Duty Truck's Performance and Fuel Consumption Under Steady Driving Conditions

คณิต วัฒนวิเชียร[1] Phan Minh Duc[2]
ฉัตรชัย หงษ์อุเทน เทอดศักดิ์ ชัยสุริยะพันธ์
ภาควิชาวิศวกรรมเครื่องกล คณะวิศวกรรมศาสตร์ จุฬาลงกรณ์มหาวิทยาลัย
ปทุมวัน กรุงเทพมหานคร 10330
โทร 0-2218-6607 โทรสาร 0-2252-2889 Email: wkanit@chula.ac.th[1], duc.aut@dng.vnn.vn[2]

Kanit Wattanavichien[1], Phan Minh Duc[2]
Chatchai Hongsa-Utain, Therdsak Chaisuriyaphun
Internal Combustion Engine Research Laboratory, Faculty of Engineering,
Chulalongkorn University, Phaya-Thai Road, Patumwan, Bangkok 10330, Thailand
Tel.: +66-22186607 Fax: +66-22522889 Email: wkanit@chula.ac.th[1], duc.aut@dng.vnn.vn[2]

Abstract

A computer model simulating the performance and fuel consumption of vehicles operating under steady driving conditions was developed and verified. The model calculations were based on the total vehicle's aerodynamic drag, grade and rolling resistance, and other vehicle design parameters. These include engine performance characteristic (load and fuel consumption), fuels, drive-train details, and total vehicle weight. The steady state driving conditions were simulated which accounts for highway and open roads driving conditions. The developed model was of a modular structure and was designed to run on a personal computer. The road test protocol to investigate constant vehicle speed fuel consumption was developed. Verification of the model was achieved by comparing the predicted fuel consumption and results from the constant vehicle speed road test with diesel and crude palm oil diesel (CPO), under steady driving conditions, with the manufacturer data of a common light duty truck. An excellent quantitative agreement was obtained. The overall results demonstrated the versatility and benefits of using the developed model in the early stages of alternative fuels' development and vehicle design process.

1. Introduction

The remarkable increase in road transport over the past few decades led to a significant and numerous environmental and human health problems on either regional or global scales. Moreover, this accelerated depletion of the fossil fuels reserves. Therefore, the demand to reduce emission of "classic" pollutants (CO, HC and NOx) and vehicle fuel consumption is indispensable. It is apparent that reducing the overall harmful impact of transport can only be achieved by maximizing vehicles' performance and fuel economy. This leads to the concept of developing the "3-litre" car, consuming 3 liter of fuel for every 100 km it travels. The development of such cars required a drastic modification to engines, transmission systems, body designs and fuel system. The cost and effort involved in such modification are very high and, often; do not necessarily deliver the ultimate designers' aims. Hence, there is an ever-increasing need for more flexible and low cost approach/tools to avoid unnecessary research and development cost particularly in the early design stage.

Earlier work [10] successfully resulted in the development and verification of a computer model accurately predicting the fuel consumption of small passenger vehicles under steady driving conditions. However, some factors were not considered in the program; these included the flexible change of engine, fuel, load, road conditions, and vehicle parameters when running the model, as well as the output of results. In addition, it was less convenient due to the programming language used to construct the model.

This study compliments the previous works and details features of a model that predicts a vehicle's performance and fuel consumption under steady driving conditions. Verification of this model is based on the available data of real common vehicles and the real road test.

2. Theoretical analysis

A full theoretical analysis of the total energy consumed by vehicles and their performance, fuel consumption may be seen in many common textbooks and studies [2], [3], [5], [10], [11]. A summary of this analysis, which was based on the total forces opposing a vehicle movement under steady driving conditions, as well as aspects of calculating the vehicle's performance and fuel consumption under dynamic driving conditions, is presented hereafter.

2.1. Vehicle aerodynamics – The vehicle aerodynamic drag is determined by Equation 1.

$$R_D = 0.5 * \rho_{air} * A * C_D * V^2 \quad (1)$$

The total velocity, V, is the algebraic summation of wind, Vw, and vehicle, Vv velocities. The sign of Vw is taken negative as wind blows in the direction of the vehicle movement and visa versa.

2.2. Rolling resistance – Vehicle rolling resistance is generated by the interaction between a vehicle's tires and road, and is calculated by Equation 2.

$$R_r = R_{rf} + R_{rr} = C_r * (M_V + M_L) * g \quad (2)$$

Where, R_{rf} and R_{rr} are the rolling resistance of the front and rear tires, respectively. The rolling coefficient, C_r is a dimensionless factor expressing the effects of the physical properties of tires and road surface. The rolling coefficient is directly proportional to the level of tire deformation and inversely proportional to the tire radius. Therefore, the coefficient value tends to increase with greater loads, higher speed and lower tire pressure. The rolling coefficient values may vary from 0.013 to 0.3, corresponding with concrete/asphalt and sand road surfaces, respectively.

2.3. Grade resistance – Grade resistance, a function of the road slope, is defined by Equation 3.

$$R_g = M * g * \sin\alpha = (M_V + M_L) * g * \sin\alpha \quad (3)$$

2.4. Total resistance force

Under steady driving conditions, the vehicle's acceleration is zero, hence the total force opposing the movement of the vehicle consist of the aerodynamic drag, rolling resistance and grade resistance.

$$R_{total}^{S} = R_D + R_g + R_r \quad (4)$$

In general, acceleration mode travel, the total resistance force can be defined as Equation 5.

$$R_{total} = \delta_i * (R_D + R_g + R_r) = \delta_i * R_{total}^{S} \quad (5)$$

Where, δ_i is the torsion mass factor at the i^{th} gear, relative to the unstable rotation of all spin elements in the vehicle. It depends strongly on the gear at which the vehicle travels.

The force resulted from torsion torque transmitted from engine to the driven wheel and the interaction between it and road surface, and acting on the driven wheel by road is termed the tractive force, F_T. This force must overcome the total resistance force to maintain the vehicle to travel at a constant speed. Thus, the required engine power, P_e, is calculated from Equation 6.

$$P_e = \frac{R_{total} * V}{\eta_t} + P_q = \frac{R_{total} * V}{\eta_t} * K \quad (6)$$

Where, P_q is the required power for driving cooling water pump, compressor, dynamo, etc. Normally, this amount of power is often rated by the proportion of road-load power via a coefficient K (K>1).

2.5. Drive-train and gear transmission – One of the principle requirements of a transmission system is to properly match the engine as well as the vehicle characteristics (speed and acceleration) to achieve the desired fuel economy and vehicle acceleration. In manual gear transmission, the ratio of the highest gear has to be chosen so that the maximum power occurs at the maximum vehicle speed. In practice this ratio may slightly be greater, leading to a small effect on the maximum vehicle speed, but enables the maximum speed to be maintained against gradient or opposing wind. The ratio of the highest gear of a gearbox, i_n, can be or less than 1. The latter leads to the decrease of engine speed corresponding with vehicle's maximum speed. Hence, lower friction power, higher fuel economy and durability of the engine can be achieved.

The ratio i_{da} of the drive axle can be determined from the maximum vehicle speed at highest gear as Equation 7.

$$i_{da} = \frac{\pi * n_{emax} * R_W}{30 * i_n * V_{max}} \quad (7)$$

The ratio of the first gear is determined to satisfy three conditions. First, total resistance force corresponding with the required maximum road gradient can be overcome by maximum tractive force. Second, no slip condition of the driven wheel must be satisfied. Third, the vehicle can travel stably at full load condition with the minimum speed not exceed 5 km/h. Therefore, the ratio of the first gear is determined from Inequality 8.

$$\begin{cases} \max(i_1^{\psi}, i_1^{v}) \le \ i_1 \ \le i_1^{\varphi} \\ i_1^{v} = \dfrac{12 * \pi * n_{e\,min} * R_W}{5 * 10^2 * i_{da}} \\ i_1^{\psi} = \dfrac{(Mv + M_L) * g * (\sin\alpha_{max} + Cr) * R_W}{T_{emax} * i_{da} * \eta_t} \\ i_1^{\varphi} = \dfrac{M_{\varphi} * g * \varphi * R_W}{T_{emax} * i_{da} * \eta_t} \end{cases} \quad (8)$$

The gear's quantity and the ratio distribution of intermediate gears in gear box depend on i_1 and i_n.

2.6. Performance - Vehicle performance can be analyzed by considering the relation between the tractive force at driven wheels and the total resistance force. Maximum tractive force, changing corresponding with engine speed at its full load condition and the gear at which the vehicle travels, can be determined by Equation 9.

$$F_T^{max} = \frac{T_{e\,max} * i_{da} * i_j * \eta_t}{R_W} \quad (9)$$

2.7. Vehicle's fuel consumption – Specific fuel consumption of a vehicle at a particular speed is determined from the engine's brake specific fuel consumption (bsfc) map given by manufacturer or measured directly. This map usually contains engine performance information including bsfc at a particular speed. Engine torque, T_e, speed, n_e, and power, P_e, corresponding to vehicle speed V, at j[th] gear, are calculated from Equation 10, 11 and 12, respectively.

$$T_e = \frac{T_w}{i_{overall} * \eta_t} = \frac{R_{total} * R_W}{i_j * i_{da} * \eta_t} \tag{10}$$

$$n_e = \frac{30 * V * i_j * i_{da} * \eta_t}{\pi * R_W} \tag{11}$$

$$P_e = \frac{\pi}{30} * T_e * n_e \tag{12}$$

Vehicle fuel consumption, SFC, at constant vehicle speed V, is then calculated from Equation 13.

$$SFC = \frac{bsfc * P_e}{\rho_{fuel} * V} * 10^5 \tag{13}$$

3. Case study vehicle

A case study vehicle, a used light duty truck FORD model RANGER Super Cab 4x2, was selected to illustrate the effects in its performance and fuel consumption as some designed parameters: road condition, as well as fuel changed. Validation of the model was achieved by comparing the road test data with the simulated results. Vehicle's specifications are summarized in Table 2. The rolling coefficient value, Cr, was initially chosen as 0.015, corresponding to road surface made from concrete or asphalt. The C_D of 0.52 was selected to match the road test data. The aerodynamic drag, characterized by Equation 1, was calculated from constant air density of 1.225 kg/m^3 and the vehicle's frontal area of 2.203m^2. The engine performance and brake specific fuel consumption maps, corresponding to different fuels: diesel and crude palm oil diesel (CPO) was measured and illustrated in Figure 1 and 2. The vehicle has a five speed manual gear box with the ratios of 4.2, 2.215, 1.433, 1.0, and 0.825 corresponding to the gears from 1[st] to 5[th], respectively. The vehicle's drive axle ratio is 4.444. Transmission efficiency are 0.94, 0.902 corresponding to 4[th] gear (directly drive) and the rest, respectively. The tire size is 195/75R16 with static radius of 323 mm.

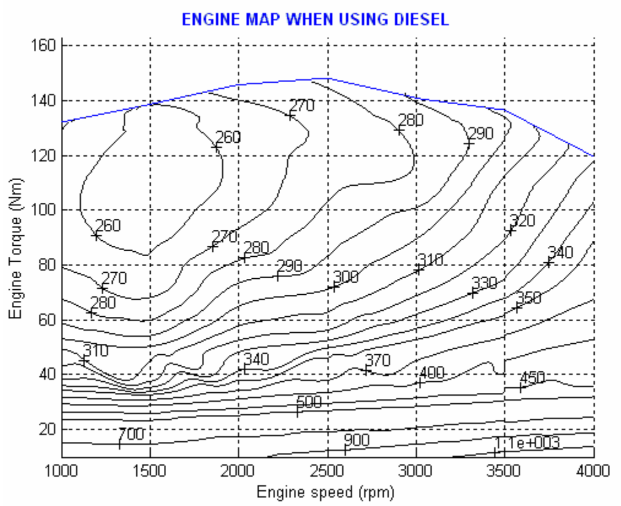

Figure 1 – Brake specific fuel consumption of the considered engine fuelled with diesel

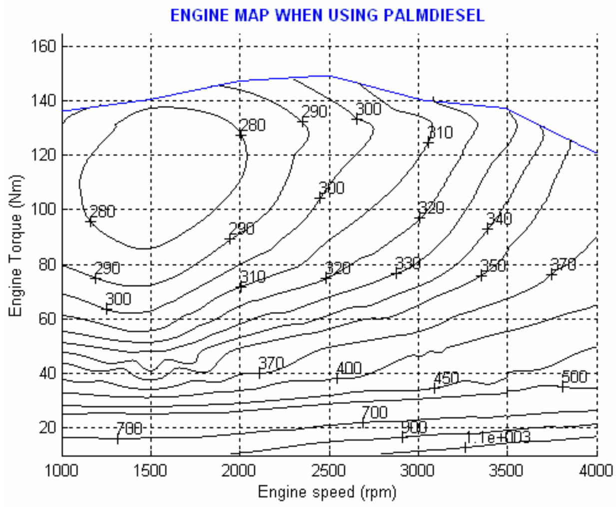

Figure 2 – Brake specific fuel consumption of the considered engine fuelled with crude palm oil diesel

Some factors were considered when performing road test to determine real vehicle fuel consumption. Firstly, the chosen roads on which the tested vehicle traveled were horizontal and adequate to easily maintain the constant vehicle speeds. Secondly, experiments were performed at night time so that the ambient temperature was not varying too much during the tests. In addition, with this test time the constant vehicle speeds was easier to maintain. Third, the collected data must be corresponding to steady states. To ensure this, a set of other parameters, the engine's operating temperature of inlet cooling water, outlet cooling water, engine oil, and exhausted air were chosen to measure in order to determine the driving period at which the vehicle reached steady state condition. Forth, vehicle fuel consumption was measured directly by volumetric method. Lastly, measured data at ambient conditions was calculated and adjusted to standard condition (ISO 3046). The road test specific vehicle fuel consumption obtained from the road test is shown in Table 1.

Table 1 – Measured specific fuel consumption of considered vehicle fuelled with diesel and palm diesel

Gear/ Speed (km/h)	Vehicle fuel consumption, SFC (liter/100km)	
	Commercial diesel	Palm diesel
4/ 60	6.46	6.69
4/ 70	7.00	7.51
4/ 80	7.77	8.31
5/ 60	5.76	5.75
5/ 70	6.40	6.56
5/ 80	7.25	7.16
5/ 90	7.87	8.01

4. Developed model

This model is a modular structure written in MATLAB, allowing the calculations of vehicle performance and fuel consumption at constant vehicle speed from basic vehicle design parameters. Its flow chart is briefly shown in Figure 3.

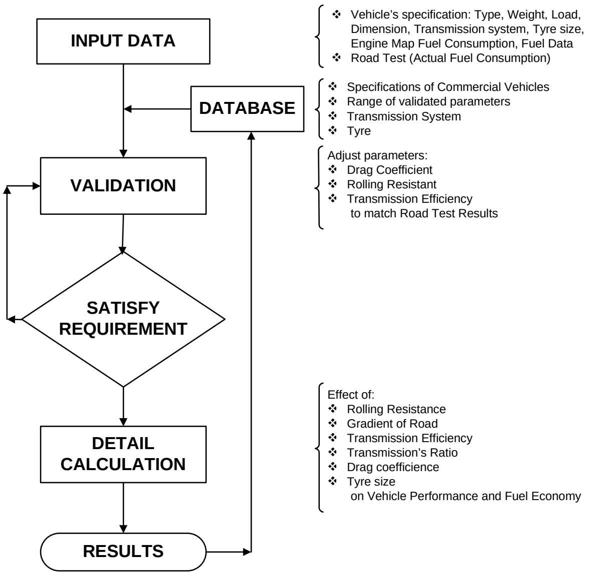

Figure 3 – Constructed model diagram

Basic parameters including vehicle's mass, aerodynamic drag coefficient, gearbox ratios, transmission efficiency, engine torque/speed and brake specific fuel consumption map, air density, tire size, and rolling coefficient were used in the simulations. Predictions, for the steady driving conditions, can be performed with two modes. With the first mode, the calculation is performed immediately from a set of refined input data get from another calculating program or manufacturers. With the second, the calculations start with the initial input, and then adjust data to determine the engine power characteristic. The vehicle fuel consumption, SFC, under the same parameters as road test was iterated with the drag coefficient and transmission efficiency adjustment to match the road test result. The vehicle fuel consumption then was calculated with the refined parameters, at various gears and speeds, then stored in result files and shown on graphs. The drag coefficient and transmission efficiency are parameters considered to be adjusted in this model via other parameters and road test data since it is quite more difficult to measure them in complicated vehicle assembly compared with the others.

5. Results and discussion

The adjusted values of drag coefficient and transmission efficiency of real considered vehicle were 0.52 and 0.902, respectively. These values are also agreed with the statistical data of these ranges for vehicles produced recent years.

Different fuels under the same conditions of load, road and vehicle resulted in different performance and SFC. The considered vehicle's performance tended to decrease with considered alternative fuel. As shown in Figure 4, maximum tractive forces of the considered vehicle fuelled with CPO were slightly lower than diesel especially at low gears. Similarly, SFC of the vehicle fuelled with the alternative fuel was higher than that with diesel. As shown in Figure 5, SFC of the considered vehicle fuelled with CPO was higher than diesel about 10.6%, 8.7%, 7.7%, 8.2% at 3rd gear @ 60km/h, 4th gear @ 80km/h, 5th gear @ 100km/h, 5th gear @ 120km/h, respectively. The decreases in performance and the increase in fuel consumption of the vehicle fuelled with CPO may be due to the lower heating value as well as the change in the period of combustion phenomena, moving toward TDC of CPO compared with that of diesel [6], [12].

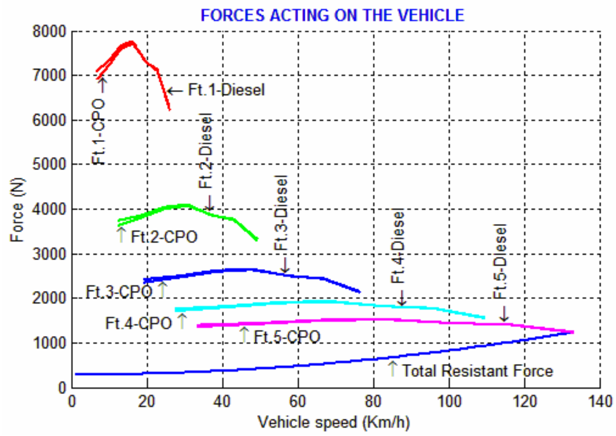

Figure 4 – Comparison in performance of considered vehicle fuelled with diesel, CPO, and diesohol

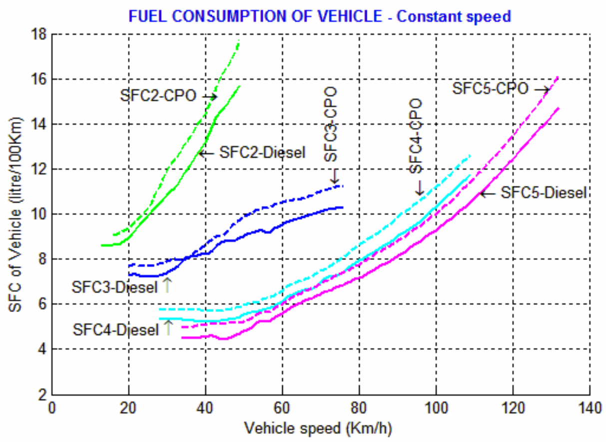

Figure 5 - Comparison in SFC at 3rd, 4th, and 5th gear of considered vehicle fuelled with diesel and CPO

The gear ratios, not only that of the 1st and the highest but also that of the intermediate gears, impact strongly on vehicle performance and fuel consumption. For instance, a slight decrease in ratio of the highest gear might result in a slight decrease in tractive force, also a decrease in SFC of a vehicle at partial loads or full load and near zero grade of road. This testified the fundamental to determine the highest gear's ratio as mentioned above. Change of this ratio from manufacture's value, 0.825, to 0.7 resulted in a decrease of 7.4% in SFC of the considered vehicle from 9.29 to 8.61 lit/100km at speed of 100 km/h, at zero-slope road as shown in Figure 6. However, higher decrease in ratio of the highest gear may cause negative effect on vehicle SFC and performance, as shown in Figure 7.

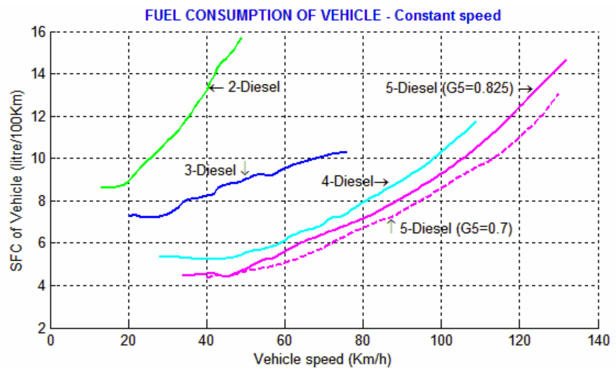

Figure 6 – Comparison in SFC of considered vehicle fuelled with diesel as the highest gear's ratio changed

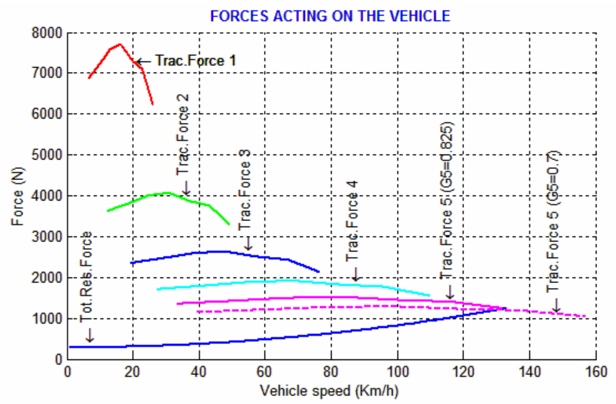

Figure 7 – Comparison in performance of the vehicle fuelled with diesel as the highest gear's ratio changed

Similarly, the tire size (of driven wheel) impacts vehicle performance and SFC. A slight increase in driven wheel's radius might result in the same trend as a slight decrease in ratio of the highest gear. Also, higher increase in driven wheel radius may cause negative effect on vehicle SFC and performance. For instance, when tire size changed from the standard size 195/75R16 to 225/75R16, corresponding to the 7.1% increase of wheel's radius from 323 mm, considered diesel fuelled vehicle shows an decrease in SFC of about 10%, 4.8%, 3.3%, and 5% from 9.53, 7.93, 9.29, and 12.44 lit/100km at 3rd gear @ 60km/h, 4th gear @ 80km/h, 5th gear @ 100km/h, and 5th gear @ 100km/h, respectively, as shown in Figure 8.

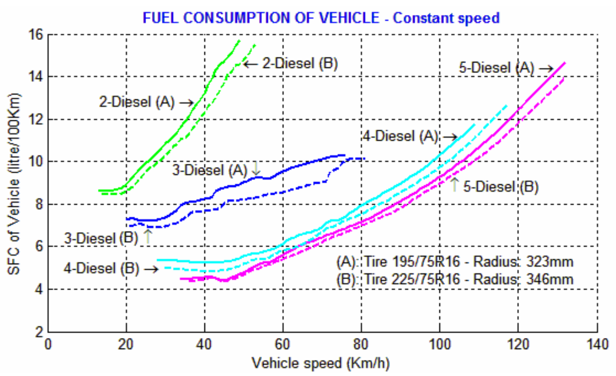

Figure 8 – Comparison in SFC of considered vehicle fuelled with diesel as the tire size changed

The considered vehicle's performance and fuel consumption were also strongly influenced by the number of gears. This is apparent as considering performance and SFC of the diesel fuelled vehicle as Figure 9 and 10. Assume that the slope of road is not less than 7% and the 4th gear is not used. In this case, the vehicle can travel with the 1st, 2nd, and 3rd but can not with the highest gear. Hence, the achieved maximum speed decreases and its fuel consumption increases. By this simulation results, it also can be seen that driving skills and habit influence vehicle fuel consumption. Travel at inappropriate gear results in higher vehicle SFC. Regarding to this, automatic transmission that the gear ratio can be automatically changed to match the tractive force curve corresponding to the change of total resistance force and to maintain the engine at full load condition, would be the best solution.

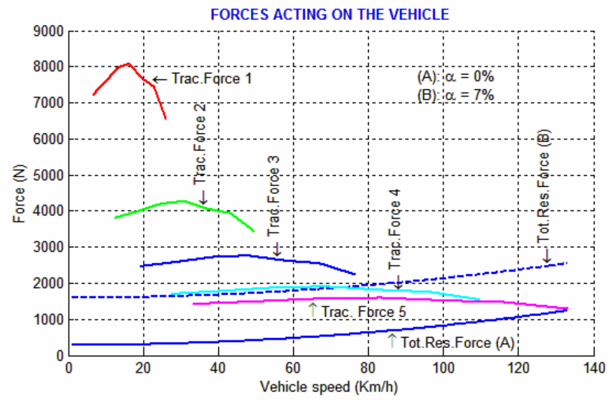

Figure 9 – Considered vehicle performance fuelled with diesel as the road slope is at 7% gradient

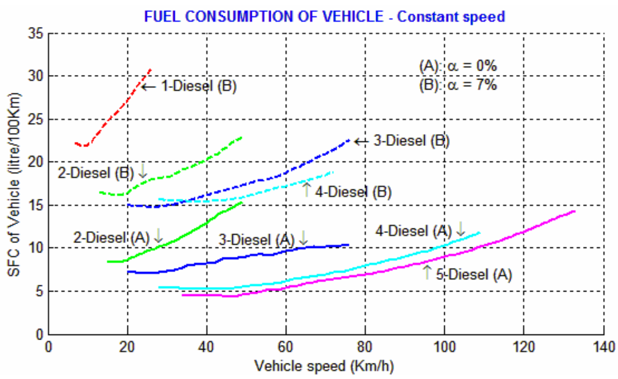

Figure 10 – SFC of the considered vehicle fuelled with diesel, road slope changes from 0% to 7%

Generally, vehicle SFC tends to decrease with one or more of these factors: the decrease in the total vehicle mass (vehicle or/and load), the increase in transmission efficiency, the decrease in drag coefficient. This appearance is due to the fact that higher energy needs to be generated to overcome the higher total resistance force of the vehicle. It can be seen that with the same SFC, vehicle load can be increased directly from the reduction of vehicle mass. The important point which should be noted is that the amount of fuel saved will be much higher at high vehicle speed. The following calculated results with the considered vehicle fuelled with diesel shows more detail in Figure 11. In this figure, for the case of about 5% increase in transmission efficiency from 90.2% to 94.7%, SFC can be saved 3.4% and 4% at 5th gear @ 100 km/h and 5th gear @ 120 km/h, respectively. Similarly, for the case of decrease in drag coefficient from 0.52 to 0.45, SFC can be saved 5.7%, 6.6%, and 8.5% at 4th gear @ 80 km/h, 5th gear @ 100 km/h, and 5th gear @ 120 km/h, respectively, as shown in Figure 12.

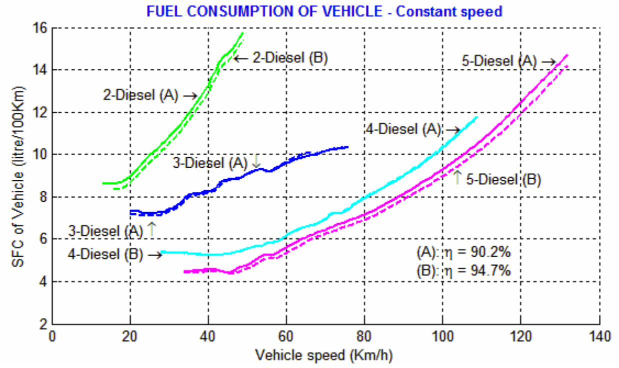

Figure 11 – Change in transmission efficiency and SFC of the considered vehicle fuelled with diesel.

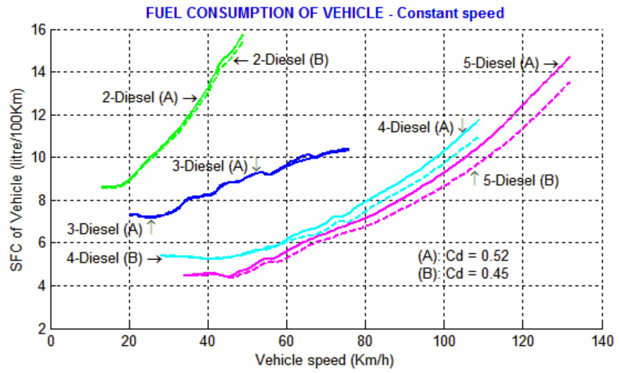

Figure 12 – Change in drag coefficient and SFC of the considered vehicle fuelled with diesel.

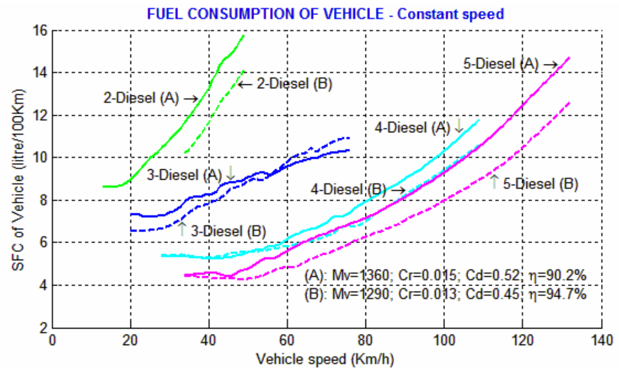

Figure 13 – Decrease in SFC of the considered vehicle fuelled with diesel as some parameters changes.

In summary, for the case of the decreases in tire rolling coefficient from 0.015 to 0.013, in drag coefficient from 0.52 to 0.45, 70 kg in vehicle mass from 1360 kg, and the increase in transmission from 90.2% to 94.7%, SFC can be saved 11.4%, 13.9%, and 15.7% at 4th gear @ 80 km/h, 5th gear @ 100 km/h, and 5th gear @ 120 km/h, respectively, as shown in Figure 13. The higher the vehicle speed, the more effective to achieve from these changes. In addition, not only fuel but also engine's fuel economy is the very important factor playing role for vehicle fuel consumption reduction. From these analyses, it is cleared that a reduction of over 10% in the SFC, at a vehicle speed of 120 km/hr, can be achieved by such improvements. Hence, it may be concluded that the development of the "3-litre car" may be possible if a vehicle is modified drastically to accommodate these improvements.

6. Conclusion

- An adequate model used to calculate vehicle fuel consumption at steady driving conditions was successfully developed.
- The model is verified by comparing the calculated values with the road test values where good quantitative agreement were obtained.
- Consequently, it is revealed that the model provides a powerful, flexible and cheap tool to optimize the vehicle design/performance at the early stages of the vehicle design, with respect to improving vehicle's performance, fuel economy and reducing emission.

Table 2 – Designed specifications of the case study vehicle FORD RANGER Super Cab 4x2

Dimension		
Overall dimension Length x Width x Height	mm	4760 x 1760 x 1730
Wheel base	mm	2835
Front / Rear track	mm	1405 / 1410
Tire size Static wheel radius	- mm	195/75 R16 323
Weight		
Vehicle mass	kg	1360
Standard payload	kg	550
Total standard veh. mass	kg	1910
Engine		
Model	-	WL
Layout	-	4 cylinders inline
Capacity	cm^3	2499
Maximum speed	rpm	4000
Max. torque, at 2500rpm	Nm	170
Maximum power	KW	60
Bore / Stroke	mm	93 / 92
Compress ratio	-	21.6 : 1
Basic fuel	-	Diesel
Type of chamber	-	Swirl chamber
Transmission system		
Type	-	Manual
Gear box ratios	-	4.2, 2.215, 1.433, 1.0, and 0.825
Drive axle gear ratio	-	4.444
Driven wheel	-	Rear axle (4 x 2)

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Nomenclature

A: Vehicle frontal area (m^2)

C_D, Cr: Aerodynamic drag and tire rolling coefficient

g: Gravitational acceleration (m/s^2)

i_1, i_n: Gear ratio of the 1st gear, the highest gear in the gearbox with n speed

i_{da}: Gear ratio in the drive axle

$i_{overall} = i_n * i_{da}$: Overall gear ratio of transmission system

i_1^V, i_1^φ, i_1^ψ: Limit value of 1st gear ratio corresponding to the minimum vehicle speed of 5km/h at 1st gear, the no slip condition of driven wheel at 1st gear, and the maximum road slop required to overcome, respectively.

φ: Dynamic friction efficiency between wheel and road

M_φ: Mass acting on driven axles (kg)

M= $M_V + M_L$: Total mass of vehicle (kg)

M_V, M_L: Mass and load of vehicle (kg)

n_e, T_e, P_e: Speed (rpm), output torque (Nm), and power (W) of engine

bsfc: Brake specific fuel consumption of engine (g/kW/h)

R_W: Wheel radius (m)

R_g, R_r: Grade and rolling resistance (N)

R_{rf}, R_{rr}: Rolling resistance of the front and rear tires (N)

R_{total}: Total resistance force (N)

V: Total velocity (km/h)

V_v, V_w: Vehicle, wind velocity (km/h)

α: Road slop angle (degree)

η_t: Transmission efficiency

δ_j: Torsion mass factor at jth gear

ρ_{air}: Air density (kg/m^3)

ρ_{fuel}: Fuel density (kg/litre)

IDI: Indirect injection