IOP Conference Series: Materials Science and Engineering

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To cite this article: K Hudha *et al* 2019 *IOP Conf. Ser.: Mater. Sci. Eng.* **530** 012031

IOP Conf. Series: Materials Science and Engineering **530** (2019) 012031 doi:10.1088/1757-899X/530/1/012031

Multi-objective Optimization of Vehicle Speed Control using Gravitational Search Algorithm for Electro-Mechanical Continuously Variable Transmission

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Abstract. This paper describes the application of the Gravitational Search Algorithm (GSA) for the optimization of the vehicle speed control in the longitudinal vehicle model equipped with Continuously Variable Transmission (CVT) system. In this study, multi-objective procedure which are to minimize the vehicle speed error and fuel usage has been proposed. For comparing and analysing the optimization results, CVT and Automatic Transmission (AT) are placed in the vehicle transmission model separately, and the Proportional, Integral and Derivative (PID) controller is implemented to control the vehicle speed. The simulation work is modelled and performed using MATLAB/Simulink software. Overall, the results show that the vehicle speed transient responses and fuel usage optimized by GSA are better than the values gained from PID controller.

1. Introduction

Latest technologies in the vehicle must satisfy the environmental issues such as excessive pollution emissions and usage of the hydrocarbon fuels. To attain better fuel economy and lower emissions, researchers have taken a lot of effort in improving the transmission system efficiency. Therefore, general type of transmission which is manual transmission (MT), automatic transmission (AT) or continuously variable transmission (CVT) system has been upgraded to fulfill the demand for better vehicle performance. Among these transmission systems, CVT has more capability to achieve good fuel economy and vehicle acceleration. The CVT offers a continuous gear ratio operation within its range without shift shock, thus allow the engine to run in optimum stage for various vehicle load conditions [1].

Most of the vehicle equipped with CVT system use electro-hydraulic-mechanical (EHM) actuation type to change the CVT ratio for smooth and comfortable drive. However, the EHM actuation system requires weighty components, leakage possibility within the hydraulic line system and difficult to control due to the non-linearity of the compressible hydraulic oil [2]. Another option is used electro-mechanical (EM) actuation system, where the system only consists of electronic and mechanical components in getting the desired transmission ratio. In this study, AT and single actuator double acting electro-mechanical (SADAEM) CVT system are used to represent the vehicle transmission model [3]. Meanwhile, for the vehicle performance simulation, lateral forces acting on the vehicle model are neglected, thus the longitudinal vehicle dynamic model is selected to be used in the simulation. For controlling the vehicle performance, several vehicle parameters need to be accessed, such as the vehicle speed and fuel usage.

Previous studies were implemented conventional Proportional, Integral and Derivative (PID) or advanced controllers such as Fuzzy and Fuzzy-PID to control the vehicle speed [4-7]. Additionally, several optimization techniques were introduced such as Particle Swarm Optimization (PSO), Genetic Algorithm (GA), Ant Colony Algorithm (ACA) and Gravitational Search Algorithm (GSA) [8]. By merging one or more of this optimization algorithm, better and optimal of desired results can be obtained [9-10]. All these techniques are used mathematical programming such as MATLAB software for optimizing the desired objectives. Thus, in this study, the GSA that has been proposed by Rashedi et al. [11] is chosen because can performs better results compared to PSO [12]. The objectives of this study are to minimize the vehicle speed error and fuel usage of the developed vehicle longitudinal model.

This paper is structured as follows. Section 1 gives the introduction and several previously and current related works. Next, Section 2 focuses on the development of vehicle longitudinal model which involve several important mathematical equations and systems. Then, Section 3 shows the proposed of the vehicle control strategies using PID and optimization using the GSA. Section 4 discusses the simulation results based on the proposed control strategies and is followed by the conclusion in Section 5.

2. Longitudinal Vehicle Dynamics Model

Consider a hatchback type of vehicle is moving on an inclined road as shown in Figure 1. The longitudinal vehicle dynamic model is presented by a lumped mass, (m) with a forward velocity of (V) along the X-direction. The vehicle wheels rotate at angular velocity presented by (ω_i) with wheel rolling radius of (R_i) and a polar moment of inertia, (J_i). The subscript of i=f,r describes either the front wheel or rear wheel of the vehicle. Then, the coefficient of friction between the road surface and front wheel is donated by (μ_f) and rear wheel is presented as (μ_r), respectively. The dimensions (l_f) and (l_r) represent the distance between the center of vehicle mass and the front axle and rear axle, respectively, with the vehicle wheelbase is denoted by the dimension of (L). The gap between the vehicle center of mass and the ground is presented as (H) and (ϕ) indicates the angle of road inclination.

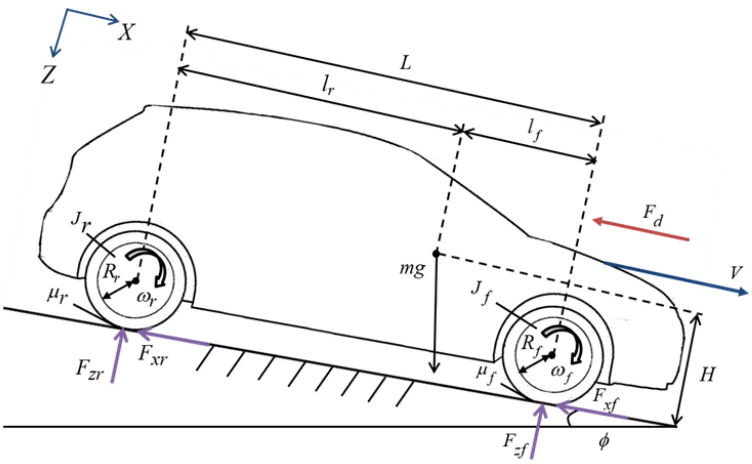


Figure 1. Longitudinal forces acting on a vehicle moving on an inclined road

By applying Newton's second law which is based on the forces balance acting on the vehicle, the vehicle acceleration can be obtained as follow:

$$\ddot{x} = \left[-F_{X_T} - F_d + mg\sin(\phi) \right] / m \tag{1}$$

where total forces that acting at the vehicle body is denoted as (F_{XT}), (F_d) is the aerodynamic resistance force, gravitational acceleration is denoted as (g) and (m) is the mass of the vehicle.

In the longitudinal vehicle model, the main forces that influence the vehicle dynamics are act in the X-direction as well as in the Z-direction. The total forces that act on the vehicle through the vehicle wheels or road surface can be defined as below:

$$F_{X_T} = 2F_{xf} + 2F_{xr} \tag{2}$$

$$F_{xf} = \mu_f F_{zf} ; F_{xr} = \mu_r F_{zr} \tag{3}$$

where (F_{zf}) and (F_{zr}) is the normal force that act at the front wheel and rear wheel, respectively. Meanwhile, (F_{xf}) and (F_{xr}) is the reaction force at the front and rear wheel, respectively.

Another important parameters, namely the load distributions, drag forces, wheel dynamics and wheel tractive properties were referred to Aparow et al. [13] and Satar et al. [14]. Then, in term of engine dynamics, 75 kW engine was selected which delivers maximum torque of approximately 160 Nm at engine speed about 3000 rpm.

In this study, two types of transmission were used, namely AT and CVT. A 5-speed AT was selected with the gear shift map data was taken from the CarSim software. In the gear ship map data, there are two corner points at each of the gear shift profile, which at 20% and 80% of throttle opening setting. This allows the engine to operate at the desired torque region. The engine speed will increase when the throttle opening is risen, results the gear numbers to upshift in sequence. In contrast, the gear shift map allows the AT to drop into lower gear if the engine speed is reduced by applying the brake or minimizing the throttle opening. The MATLAB/Simulink block namely *If=else* is used to represents the gear shift map in either for upshifting or downshifting the gears.

On the other hand, another type of transmission is considered, namely SADAEM CVT system which has a pulley gear ratio range from 0.3 to 2.9, that was placed in the transmission model. In simulation, virtually the input shaft of SADAEM CVT system is connected to the ICE via torque converter and the output shaft is coupled with the differential which has a final gear ratio of 4.1 [3].

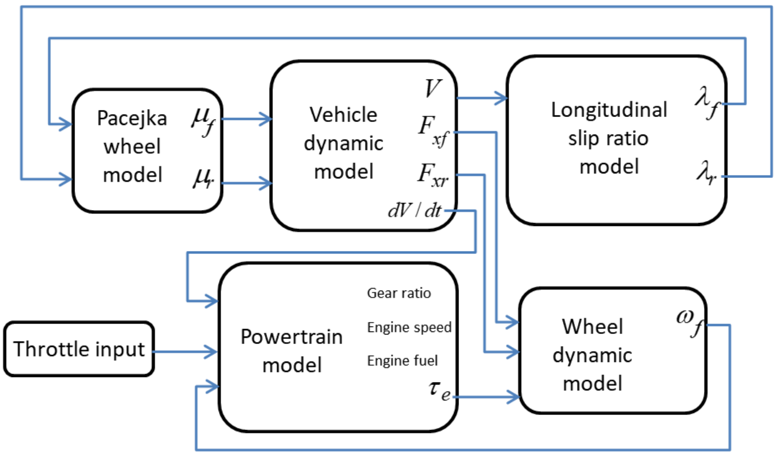


Figure 2. Longitudinal vehicle model in the MATLAB/Simulink software

The developed vehicle dynamics model in longitudinal direction was simulated in the MATLAB/Simulink software where the vehicle was assumed to run on a flat and dry tarmac road surface as shown in Figure 2. A sudden acceleration at 100 % throttle opening was set up to be input to the vehicle model. The simulation was performed using *Bogacki-Shampine* solver for a period of 100 seconds with a fixed step size of 0.01 second. Meanwhile, all the parameters values can be assessed in the CarSim software. All parameters for the vehicle model in the MATLAB/Simulink and CarSim software were kept at the same values and are listed in Table 1. For verification process, both models were used hatchback vehicle type and AT system.

Table 1. Parameters of hatchback type vehicle model

Description	Symbol	Value
Wheelbase	L	2.347 m
Front length from COG	l_f	1.103 m
Rear length from COG	l_r	1.244 m
Height of vehicle from COG	H	0.540 m
Vehicle sprung mass	m	747 kg
Initial vehicle velocity	Vo	1 m/s
Aerodynamic drag coefficient	Cd	0.48
Vehicle frontal area	A	1.6 m^2
Air mass density	ρ	1.206 kg/m^3
Gravitational acceleration	g	9.81 m/s^2
Wheel radius	Rw	0.2786 m
Gear final	η_f	4.10
Gear 1	η_1	3.78
Gear 2	η_2	2.12
Gear 3	η_3	1.36
Gear 4	η_4	1.03
Gear 5	η_5	0.84

3. Vehicle Control Strategies

3.1. PID controller

For the vehicle equipped with AT, outer control loop was considered in evaluating the vehicle speed control of the developed model. Meanwhile, regarding vehicle with CVT, two control loops were designed. An outer loop to control the vehicle speed, while another loop in the transmission model to control the CVT ratio. Both AT and CVT were used same outer loop controller, where a linear, simple and robust controller, namely PID controller was applied as shown in Figure 3. On the other hand, in the transmission model, two types of controller were used which are the PD and PI controllers as shown in Figure 4. To obtain the proportional, integral and derivative gain values, trial and error technique was carried out. For the outer loop, the value of proportional, integral and derivative gain was set to value of 150, 1 and 5, respectively. In contrast, inside the transmission model, the proportional and derivative gain was fixed at 250 and 3, respectively. Meanwhile, proportional and integral gain was set at 53 and 25, respectively.

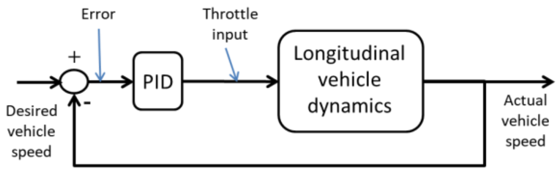


Figure 3. Vehicle speed control structure using PID controller

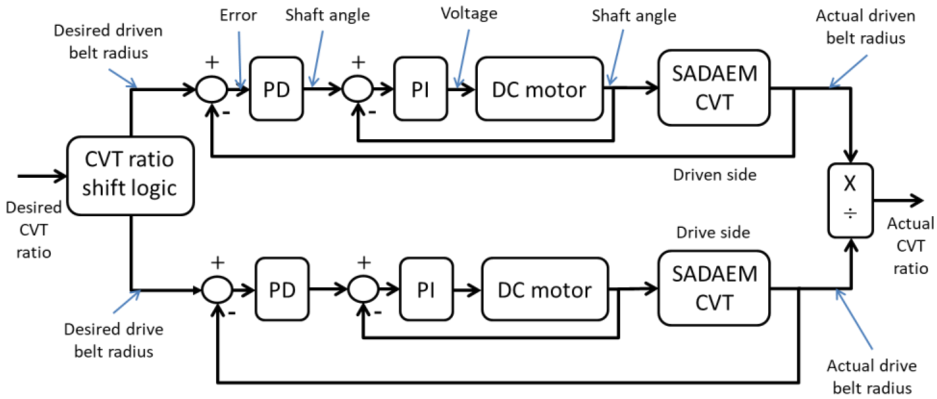


Figure 4. CVT ratio control structure in simulation

3.2. Optimization using the Gravitational Search Algorithm

The GSA utilizes Newton's law of gravity and law of motion to find the optimum values of PID controller parameters. In GSA, there are four parameters, namely position, inertial mass, active and passive gravitational mass for each agent that need to be considered. By using the Newton's law of motion, the acceleration is calculated before the velocity and position of each agent is updated. The updated position will give an update solution for the optimization problem through a fitness function. The updating process is repeated until the criterion is achieved. The GSA flow procedure can be summarized as shown in Figure 5. All the parameters and equations related to the GSA have been studied as stated by Sabri et al. [8], Rashedi et al. [11] and Amer et al. [15].

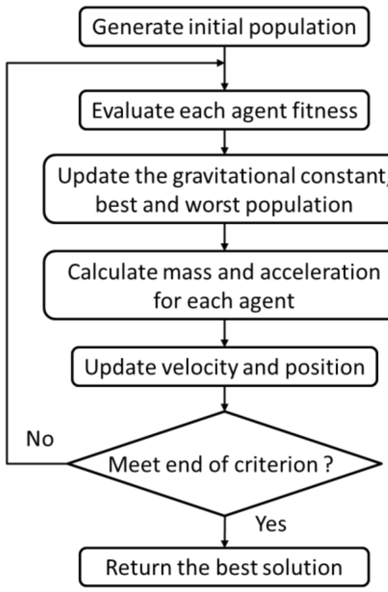


Figure 5. An algorithm of the GSA [8]

In this study, since the vehicle speed error and fuel consumption were intended to be minimized, the fitness function (j) related to these aims can be expressed by:

$$j = \sqrt{\left(\frac{e_v}{e_{v\max}}\right)^2 + \left(\frac{f_c}{f_{c\max}}\right)^2} \tag{4}$$

where the vehicle speed error is expressed as (e_v), the maximum error of vehicle speed is denoted as (e_{vmax}), the fuel consumption is stated as (f_c) and the maximum fuel consumption is represented as (f_{cmax}). The optimization process only concentrates on outer loop only. Moreover, initial value of the parameters using the GSA are listed in Table 2. On the other hand, both PID and optimization using GSA control strategies are using *Bogacki-Shampine* solver with a step size of 0.01 second for a period of 100 seconds.

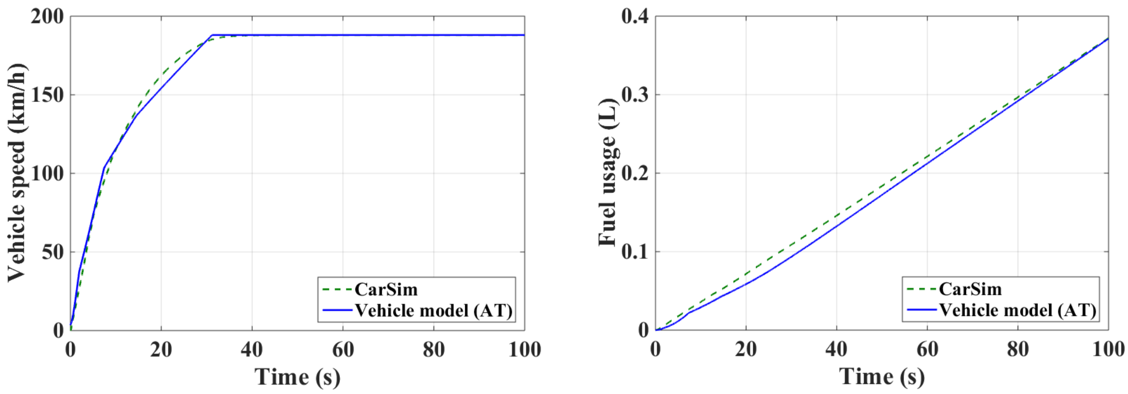
Table 2. Initial parameters for GSA simulation

Parameter	Value
Agents number, N	50
Iteration number, T	50
Bheta, β	0.1
Epsilon, ε	0.01
Dimension number, D	3
Upper bound limit [Kp Ki Kd]	[200 10 20]
Lower bound limit [Kp Ki Kd]	[0 0 0]

4. Results and Discussion

4.1. Verification of vehicle longitudinal with CarSim model

Verification results are presented through comparison between vehicle model in MATLAB/Simulink and CarSim vehicle model in term of vehicle speed and fuel usage as shown in Figure 6. These vehicle responses are analysed by evaluating the percentage difference of the root mean square (RMS) for each model. As seen in the graphs, response trends between CarSim and simulation models are relatively in good agreement with minor deviation. For CarSim model, the vehicle speed and fuel consumption have the RMS value of 187.964 km/h and 0.372 L, respectively. Meanwhile for vehicle model, vehicle speed records RMS value of 187.999 km/h and fuel consumption shows RMS value of 0.371 L. Therefore, RMS percentage difference is up to 0.035 % for vehicle speed and 0.269 % for fuel consumption. The small RMS percentage difference and minor deviation of responses pattern are caused by various simplified model assumptions in designing the vehicle longitudinal model. However, the developed vehicle model is adequate to be used for determining another desired vehicle responses.



(a) Vehicle speed against time (b) Fuel usage against time

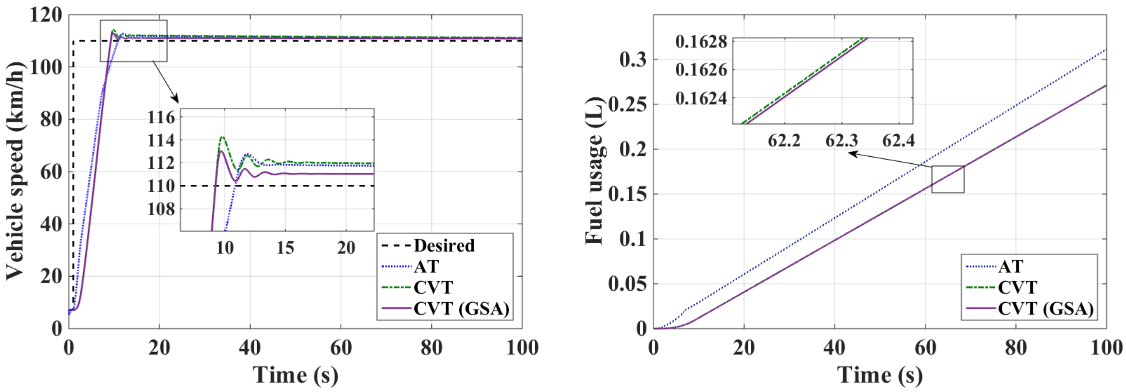
Figure 6. Verification results between developed vehicle and CarSim model

4.2. Optimization results

Figure 7 (a) and (b) show the vehicle speed and fuel usage responses using PID controller in AT and CVT system, while optimization using GSA has been performed for CVT system. From the graph, with same controller which is PID, the vehicle model consists of CVT is perform better than AT in term of rise time, settling time and fuel usage. Despite higher value of overshoot (OS) percentage and steady state error compared with AT system, the CVT system inside the vehicle model still can performed with acceptable level. In addition, optimization using GSA will improve the fuel usage up to 0.04 % when compared with CVT (PID) model. Moreover, the results proven that the vehicle model with GSA managed to obtain better transient responses and fuel usage results. The analysis of vehicle speed transient responses for each transmission and controller are tabulated in Table 3.

Table 3. Transient analysis for vehicle speed response

Transmission / Controller	Rise time (s)	Percentage of OS (%)	Settling time (s)	Steady state error (%)
AT / PID	10.90	2.54	16.00	1.36
CVT / PID	9.27	3.91	15.80	1.55
CVT / GSA	9.25	2.73	15.70	0.91



(a) Vehicle speed against time (b) Fuel usage against time

Figure 7. Vehicle responses with different transmissions and controllers

5. Conclusion

The developed vehicle longitudinal model was verified against the vehicle performance obtained from CarSim software in term of vehicle speed and fuel usage. The verification results show the pattern of the responses were almost similar but varies in term of RMS percentage difference values, approximately below 0.3 %. Moreover, the vehicle with CVT system shows better performance in term of acceleration and transient responses compared with AT system. Furthermore, in optimizing the vehicle speed and fuel usage in the CVT, the vehicle speed control using GSA has better characteristics and good dynamic behaviour than PID, which less response time, smaller overshoot, shorter settling time and less steady state error. Therefore, the proposed GSA method can be used in solving complex and highly non-linear of vehicle system.

Acknowledgement

This project is supported by Ministry of Education Malaysia through the research grant LRGS (LRGS/B-U/2013/UPNM/DEFENCE & SECURITY-PI).

IOP Conf. Series: Materials Science and Engineering **530** (2019) 012031 doi:10.1088/1757-899X/530/1/012031

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