# Gear Shift Schedule Optimization and Drive Line Modeling for Automatic Transmission

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***Abstract*—Gear shifting strategy is the core of intelligent control of any automatic transmission used in modern vehicles. It directly influences the vehicle performance, drivers feel and fuel economy. This paper describes two different design methods of gear shift schedules for optimum dynamic performance and fuel economy separately. Mathematical model of vehicle driveline containing automatic transmission and other parts of vehicle is developed to evaluate the effectiveness of these methods. The developed model is simulated on European drive cycle to establish effectiveness of the derived gear shift schedule methods. Simulation results show that the designed methods are quite effective and can be used to generate gear shift schedules for automatic transmissions.**

**Keywords:** Automatic transmission, Gear Shift Schedule, Vehicle Driveline.

## I. Introduction

Automatic transmission (AT), automated manual transmission (AMT) and intelligent gear shift schedules are key technologies to improve performance of a vehicle. With the advent of new technologies in mechatronics, control engineering and embedded systems, development of modern vehicle technology is stepping in to a completely new era. One area needs more focus is to derive different gear shift schedules to get optimum vehicle performance. These methods can be developed offline and hard coded in to transmission electronic control unit or a real time adaptive intelligent strategy depending on the vehicle states. This paper addresses and compares two offline methods to derive gear shift schedules

### A. Motivation

At present manual transmissions are most widely used in vehicles. In this, gear shifting is controlled by driver's wish. Improper gear shifts or gear shifting which is not required are most common experiences which result in reduced fuel economy and reduced performance of the vehicle. So, in this scenario, automatic transmission with intelligent control of gear shifting is a remedy to improve fuel efficiency of the vehicle [13]. Hence, gear-shifting control strategies for vehicle automatic transmissions are important issues for design engineers and researchers to attain high efficient driveline systems.

Conventional gear shift operations for automatic transmission based vehicles are implemented in the form of gear shift maps. Gear shifting points are generated based on current vehicle speed and accelerator pedal position i.e. throttle opening. The shift maps are traditionally built on intuitive knowledge of engineers.

Many fuel-efficient gear shift algorithms are reported in literature. Authors in [9] have developed a gear shifting algorithm based on chaotic neural network. It is self learning algorithm which adapts to changing driving conditions. Researchers in [2] have developed shifting strategy by considering matching of engine and torque converter. Whereas Ngo.et.al [3] have derived gear shifting schedule by optimizing cost function of fuel consumption with the help of dynamic programming algorithm.

This paper discusses the method of designing the dynamic 3-parameter gear-shift schedule by using vehicle accelaration along with vehicle speed and throttle opening [2]. This paper also discusses the method of designing optimal three parameter fuel economy gear shift schedule with the help of vehicle's fuel consumption expression [1]. These gear shift schedules are compared on a simulated model of a car of gross vehicle weight of $1100$ Kg having $70$ hp engine and with a torque converter.

### B. Paper Structure

This paper is organized as follows. In Section II mathematical modeling of automobile driveline system is presented. Section III describes the method of designing optimal 3-parameter gear-shift schedule based on dynamic performance as well as optimal fuel economy. Section IV describes the design of transmission controller based designed on gear-shift schedules. Various simulation results of developed driveline model along with transmission controller on standard drive cycle and fuel economy calculations are presented in Section V.

## II. MODELING OF VEHICLE DRIVELINE

Vehicle driveline is a group of devices that produce, modulate and transfer the mechanical power within vehicle [5]. General schematic of vehicle driveline is depicted in Fig.1.

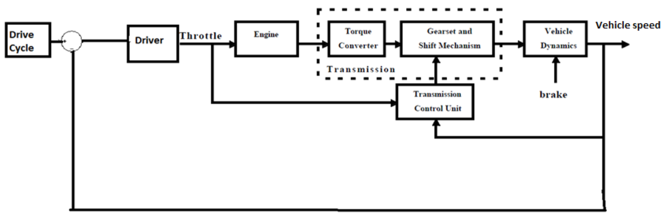


Fig. 1: Vehicle Driveline Block Diagram

### A. Engine

Mean value approach [6] is used to capture dynamics of an engine. In this approach, mean torque produced by the engine over the entire power cycle is considered. Engine rotational dynamics can be written as,

$$(J_e + J_p)\frac{d\omega_e}{dt} = T_b - T_{imp} \tag{1}$$

where, $\omega_e$ is angular speed of the engine crankshaft (rad/sec), $J_e$is moment of inertia of engine ($kg.m^2$), $J_p$ is moment of inertia of impeller of the torque converter ($kg.m^2$), $T_b$ is the engine brake torque ($N.m$) and $T_{imp}$ is the impeller torque of the torque converter ($N.m$). The engine brake torque is obtained from a steady state engine torque map. This torque map gives the engine brake torque as a function of throttle opening and engine rotational speed. Steady state engine torque-speed map is obtained from dynamometer test.

### B. Torque Converter

Torque converter transfers the engine power to the transmission. It does it so with the aid of hydraulic coupling. For this study it is represented by static nonlinear input-output model as [7],

$$T_t = R_{TQ} T_{imp} \tag{2}$$

where, $T_t$ : turbine torque ($N.m$), $T_{imp}$ : impeller torque ($N.m$) and $R_{TQ}$ is torque ratio of the torque converter. Impeller torque is related to $k-$factor of the torque converter as follows,

$$T_{imp} = \left(\frac{\omega_e}{k}\right)^2 \tag{3}$$

where $k$ factor and torque ratio $R_{TQ}$ are the functions of the speed ratio of the torque converter as given below,

$$R_{TQ} = f_1\left(\frac{\omega_t}{\omega_e}\right) \tag{4}$$

and,

$$k = f_2\left(\frac{\omega_t}{\omega_e}\right) \tag{5}$$

where, $\left(\frac{\omega_t}{\omega_e}\right)$ is the speed ratio of the torque converter where $\omega_t$ is the torque converter output speed (turbine speed). Functions $f_1$ and $f_2$ are implemented from standard torque converter characteristics.

### C. Transmission

Automatic transmission employs the planetary gear box having different gear ratios. As the main interest of this study is to optimize gear shift point with respect to throttle opening and vehicle speed, complex shift mechanisms relating to gear sets, hydraulic systems and in between shift phenomena are neglected. The transmission gear box is modeled as a simple algebraic input-output relation with gear ratio $R_{TR}$ and efficiency $\eta_{TR}$ as,

$$T_{out} = R_{TR} T_{in} \eta_{TR} \tag{6}$$

where, $T_{out}$ : transmission output torque ($N.m$) and $T_{in}$ : transmission input torque ($N.m$).

$$N_{in} = R_{TR} N_{out} \eta_{TR} \tag{7}$$

where, $N_{in}$ : transmission input speed (rad/sec) and $N_{out}$ : transmission output speed (rad/sec).

### D. Differential

Differential allows the driving wheels to rotate at different speeds. This is necessary when the vehicle turns, making the wheel that is traveling around the outside of the turning curve roll farther and faster than the other wheel. Differential is modeled as fixed axle ratio or final drive ratio as,

$$T_{wheel} = R_{FDR} T_{out} \eta_{FDR} \tag{8}$$

where, $R_{FDR}$ : final drive ratio, $T_{out}$ :transmission output torque ($N.m$) and $\eta_{FDR}$ is the efficiency of differential.

### E. Vehicle Dynamics

To model vehicle dynamics, simple one dimensional translational dynamics are considered [7]. Driving force is derived from wheel torque coming from driveline. Various opposing forces for vehicle dynamics can be listed as,

*Aerodynamic Drag*

$$F_a = \frac{1}{2}\rho_a A_f c_d v^2 \tag{9}$$

where, $F_a$ : Aerodynamic drag force (N), $\rho_a$ : density of ambient air ($kg/m^3$), $A_f$ : vehicle frontal area ($m^2$), $c_d$ :aerodynamic drag coefficient and $v$ : vehicle translational speed (m/sec).

*Rolling Friction*:

$$F_r = c_r m_v g \cos(\alpha) \tag{10}$$

where, $F_r$ :rolling frictional force (N), $c_r$ : rolling frictional coefficient, $m_v$ : mass of the vehicle (kg) and $\alpha$ : angle of inclination of surface with horizontal (rad).

*Brake Force*:
Brake force ($F_b$) is generated by driver if vehicle speed is greater than the reference speed.
Expression for the wheel force is as given below,

$$F_{wheel} = \frac{T_{wheel}}{R_w} \tag{11}$$

where, $F_{wheel}$ : wheel force (N) and $R_w$ : effective radius of the wheel (m). Vehicle speed can be obtained by solving following differential equation,

$$m_v \frac{dv}{dt} = F_{wheel} - (F_a + F_r + F_b) \tag{12}$$

### F. Driver

Driver is modeled as PID controller. It works on error produced between vehicle speed from simulation and reference speed from standard drive cycle. It produces two output signals viz. throttle and brake. Throttle or accelerator pedal position output will act as an input to engine to produce desired engine brake torque as explained in above section. Brake is applied when vehicle speed exceeds desired speed from drive cycle. So, dynamic and algebraic equations described above for various subsystems viz. engine, torque converter, transmission, differential and vehicle dynamics represent the dynamical model of the automobile driveline.

## III. DESIGN OF OPTIMAL GEAR SHIFT SCHEDULE

Gear shift schedule represents the vehicle speed at which each gear shift (upshift or downshift) shall occur [13]. They are represented as vehicle speed as a function of accelerator pedal position for a given gear ratio change (for instance 1-2 shift). Gear shift schedule is designed via two methods which are described as below.

### A. Gear Shift Schedule for Optimal Dynamic Performance

*1) Accelaration and Vehicle Speed Equations:* According to the vehicle dynamics, vehicle acceleration as a function of gear ratio can be given as,

$$a_n = \frac{\left(\frac{T_b R_{TR} R_{FDR} \eta_T}{R_w}\right) - F_a - F_r - F_b}{\delta_n m_v} \tag{13}$$

where, $F_a$ is aerodynamic drag force, $F_r$ is rolling frictional force and $F_b$ is braking force as described in above section. $\delta_n$ is equivalent mass of rotary parts of vehicle which mainly depends on engine flywheel moment of inertia ($J_e$), vehicle wheel moment of inertia ($J_w$) and transmission ratios. It is calculated as,

$$\delta_n = 1 + \frac{\sum J_w}{m_v R^2{}_w} + \frac{J_e {R_{TR}}^2 R^2{}_{FDR} \eta_T}{m_v R^2{}_w} \tag{14}$$

vehicle speed as a function of engine speed and gear ratio can be given as,

$$v_n = \frac{7.2\pi R_w N_e}{60 R_{TR} R_{FDR}} \tag{15}$$

here, $v_n$ is vehicle speed (expressed in Km/hr) and $N_e$ is engine speed (rpm)
Hence, keeping accelerator pedal position constant, for each gear ratio, series of vehicle acceleration and vehicle speed is obtained for series of engine brake torque and engine speed values which are given in engine steady state torque speed map. Hence, vehicle acceleration obtained for each gear position can be plotted against range of vehicle speed. Fig.2 shows the vehicle acceleration versus vehicle speed for each gear.

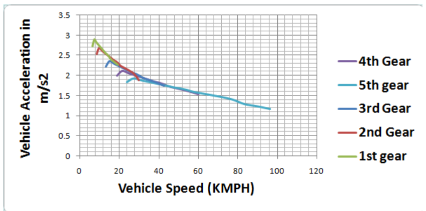


Fig. 2: Acceleration Curves

*2) Design Law for Upshifting:* The shift schedule design for dynamic performance is to pursue maximum acceleration of the vehicle [10]. Hence, shift points should be chosen at intersections of the acceleration curves of two adjacent gears at the same accelerator pedal position, i.e.

$$a_n = a_{n+1} \tag{16}$$

where, $a_n$ and $a_{n+1}$ are the accelerations of gear (n) and gear $(n+1)$ at the same accelerator pedal position respectively.

If the accelerations of adjacent gears have no intersection point at the same accelerator pedal position, then maximum speed of lower gear is chosen as shift point. Above procedure is repeated for range of accelerator pedal positions, to get shift speeds for each upshift. Upshift gear shift schedule is generated by connecting these shifting points of vehicle speeds at various accelerator pedal positions for each upshift.

*3) Downshift Schedule Design:* Automatic transmission must avoid unnecessary and repeated gear-shifts called as gear hunting. If same upshift schedule is used for down-shifting, very small deviations in vehicle speed from shift speed will cause unwanted gear hunting. Hence, buffer zone between downshift and upshift is effective way to reduce hunting [10]. Buffer zone can be defined as,

$$A_n = \frac{v_n \uparrow - v_{n+1} \downarrow}{v_n \uparrow} \tag{17}$$

where, $v_n \uparrow$ is is the upshift speed from gear (n) to gear (n+1) at given accelerator pedal position and $V_{n+1} \downarrow$ is the downshift speed from gear (n+1) to gear (n) at give accelerator pedal position. $A_n$ is taken between 0.4 to 0.45. Therefore, the downshift schedule can be calculated based on obtained upshift schedule as,

$$v_{n+1} \downarrow = (1 - A_n) v_n \uparrow \tag{18}$$

Hence, combining upshift and downshift schedule, resultant gear-shift schedule for optimal dynamic performance is shown in Fig.3

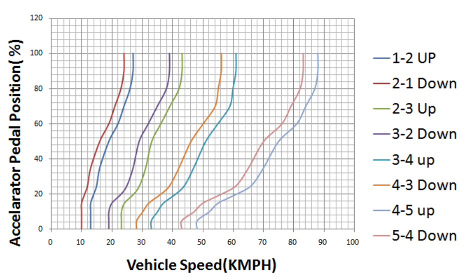


Fig. 3: Designed Gear-Shift Schedule for Optimal Dynamic Performance

## B. Gear Shift Schedule for Optimal Fuel Economy

The rule of the optimal fuel-economy gear shifting schedule is to make the automatic gear shift operate at economic shifting point, in order to achieve the goal of reducing the consumption of fuel. 3-parameter gear-shift schedule is designed based on vehicle's fuel consumption formula.

*1) Equations of Fuel Consumption:* According to the dynamics of vehicle and engine operation theory, the relationship between fuel consumption, vehicle structure and operating conditions can be expressed as follows [1],

$$Q_t = \frac{g_e v}{3600 \eta_t} \left( F_r + F_a + \delta_n m_v \frac{dv}{dt} \right) \tag{19}$$

where, $Q_t$ is vehicle fuel consumption per hour (kg/hr), $g_e$ is brake specific fuel consumption (Kg/kW.hr), $F_r$ is rolling frictional force (N) and $F_a$ is aerodynamic drag force (N). The relation between the fuel consumption per hour and total fuel consumption of vehicle $Q$ (kg) is given by,

$$Q_t = \frac{dQ}{dt} \tag{20}$$

above equation can be written as,

$$Q_t = \frac{dQ}{dv} \frac{dv}{dt} \tag{21}$$

hence,

$$Q = \frac{1}{a} \int Q_t dv \tag{22}$$

where, $a = \frac{du}{dt}$ is vehicle acceleration ($m/\sec^2$).

*2) Design Law for Upshifting:* For a minimum fuel consumption, a necessary condition is to minimize equation $(22)$. In order to guarantee the best fuel economy of a vehicle, following condition must be satisfied [1],

$$Q_{t,n} = Q_{t,n+1} \tag{23}$$

where, $Q_{t,n}$ and $Q_{t,n+1}$ are fuel consumptions per hour of gear (n) and gear (n+1) respectively. From expression (19) and (23), we can write,

$$\begin{aligned} & \frac{g_{e,n} v}{3600 \eta_t} \left( F_r + F_a + \delta_{n,n} m_v \frac{dv}{dt} \right) \\ = & \frac{g_{e,n+1} v}{3600 \eta_t} \left( F_r + F_a + \delta_{n,n+1} m_v \frac{dv}{dt} \right) \end{aligned} \tag{24}$$

Brake specific fuel consumption (BSFC) at particular gear position ($g_{e,n}$) can be expressed as a function of vehicle speed ($v$) as,

$$g_{e,n} = A_{e,n} v^2 + B_{e,n} v + C_{e,n} \tag{25}$$

where, $A_{e,n}$, $B_{e,n}$, $C_{e,n}$ are the curve fitting coefficients of the BSFC against vehicle speed data. These curves are fitted with the help of engine fuel consumption data obtained from dynamometer test.

Hence, substituting expression (25) in (24) and solving for vehicle speed, biquadratic equation in vehicle speed is obtained as follows,

$$Av^4 + Bv^3 + Cv^2 + Dv + E = 0 \tag{26}$$

where, coefficients A, B, C, D and E depend on vehicle's physical parameters like frontal area, transmission ratios etc. as well as they depend on curve fitting coefficients of BSFC-vehicle speed data. If the solution of above expression (26) is smaller than the great running speed of this gear, and greater than the least running speed of next gear, then this solution is the best shifting speed of optimum fuel economy. Same procedure is carried out to calculate shift speed for each upshift for particular accelerator pedal position and this procedure repeated for range of accelerator pedal positions. Shift speeds for various accelerator pedal positions for particular upshift are connected to generate upshift line. This way upshift schedule is generated.

*3) Downshift Schedule Design:* Similar procedure as that of optimal dynamic performance gear-shift schedule is carried out by making use of buffer zone to generate downshift schedule so as to avoid gear hunting.

Hence, gear-shift schedule designed for optimal fuel economy is as shown in Fig.4

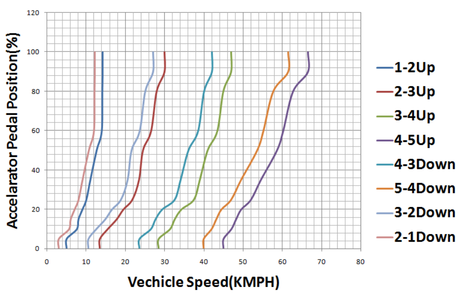


Fig. 4: Designed Gear-Shift Schedule for Optimal Fuel Economy

## IV. DESIGN OF TRANSMISSION CONTROLLER

In automatic transmission, transmission is fully controlled without driver's intervention with the help transmission controller. Transmission controller takes two inputs viz. vehicle speed and accelerator pedal position. Above described gear-shift schedules are implemented in the transmission controller. A simple algorithm designed to implement transmission controller is shown in Fig.5.

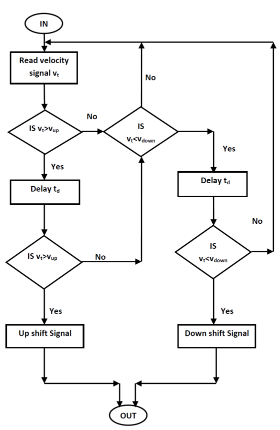


Fig. 5: Algorithm for Gear-Shifts

## V. SIMULATION RESULTS

Developed mathematical model of vehicle driveline is established in MATLAB-SIMULINK environment. Designed transmission controller alongwith designed gear shift schedules are embedded in vehicle driveline model. Transmission controller is designed with the help of MATLAB-STATEFLOW explorer. Vehicle driveline model is simulated on european city drive cycle (ECE). Total time span for this cycle is 200 sec and total distance is 0.9914 km. Reference ECE drive cycle used is as in Fig.6.

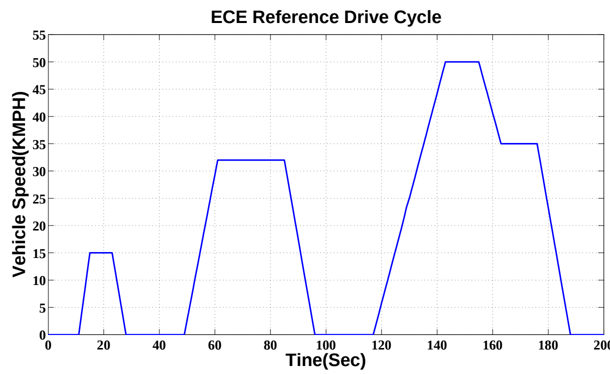


Fig. 6: Reference ECE Drive Cycle

### *A. Simulations For Gear-Shift Schedule For Optimal Dynamic Performance*

Vehicle speed profile obtained by simulating the model is as shown in Fig.7.

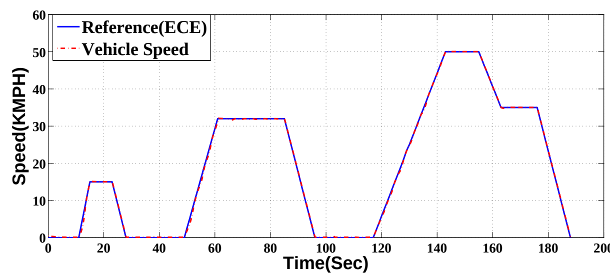


Fig. 7: Vehicle Speed

Fig.8 depicts Gear-shift profile obtained for above drive cycle.

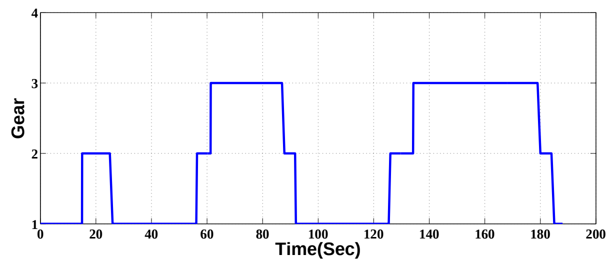


Fig. 8: Gear Shift Profile

From above results, it is evident that reference drive cycle is followed with reasonable accuracy as well as gear hunting is absent in gear shift profile.

### *B. Simulations For Gear-Shift Schedule For Optimal Fuel Economy*

Vehicle speed simulated is shown in Fig.9.

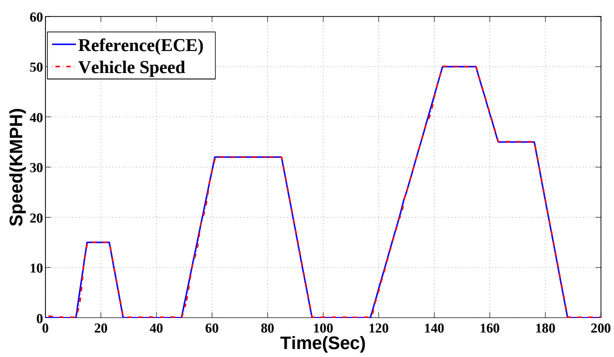


Fig. 9: Vehicle Speed

Fig.10 shows the gear-shift profile obtained for above drive cycle.

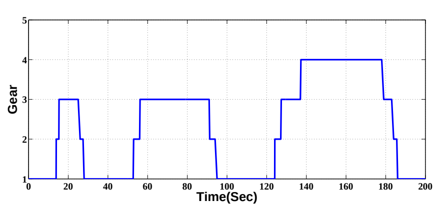


Fig. 10: Gear Shift Profile

From gear shift profile, it is clear that more number of gear shift operations are necessary when gear shift schedule for optimal fuel economy is implemented as compared with gear-shift schedule of dynamic performance.

### *C. Fuel Economy*

Fuel economy (F.E.) is expressed in terms of distance covered in kilometers per litre of the fuel consumed (Km/litres). Integrating instantaneous product of brake power and brake specific fuel consumption will give the net fuel consumption in grams and dividing this integral with density of the fuel will give the net fuel consumption in liters. Integrating the vehicle speed over the period of the drive cycle will give the total distance covered in kilometers. Hence,expression of the fuel economy can be written as,

$$F.E. = \frac{\rho \left( \int_0^{t_f} v dt \right)}{\left( \int_0^{t_f} g_e P_b dt \right)} \tag{27}$$

where, $\rho$ is density of diesel ($kg/l$) and $P_b$ is brake power generated by engine (kW) and $g_e$ is brake specific fuel consumption (Kg/kW.hr).

### *D. Wide Open Throttle Performance*

To study the effect of both designed gear shift schedules on vehicle acceleration performance, wide open throttle test is simulated as per the procedure described in SAE J 1491 standard.

### *E. Summary of Simulations*

Above simulation results are summarized in following table.

TABLE I: Summary of Simulations

| | Gear Shift Schedule | |
|---|---|---|
| | Optimal Dynamic Performance | Optimal Fuel Economy |
| Percentage Distance Error | 0.17 | 0.08 |
| RMS error in vehicle Speed | 0.4661 | 0.4513 |
| Fuel Economy(Km/Litre) | 13.4 | 15.32 |
| Time required to achieve speed of 97KMPH (sec) | 14.98 | 15.31 |

The percentage distance error and RMS error of vehicle speed are calculated between vehicle speed from simulation and reference speed from standard drive cycle. From above table, it is clear that better fuel economy performance is obtained from gear shift schedule corresponding to optimal fuel economy where as better acceleration performance is obtained from gear shift schedule corresponding to optimal dynamic performance.

## VI. CONCLUSION

In this paper two gear shift schedules have been designed for optimizing vehicle acceleration and fuel economy. To test the effectiveness of these gear shift schedules, mean value mathematical dynamic model of vehicle driveline has been developed. Developed mathematical model of vehicle driveline along with transmission controller has been established in MATLAB-SIMULINK platform. It is concluded from the simulation results that gear shift schedule designed for optimal dynamic performance maximizes vehicle acceleration as per the simulation of wide open throttle performance of the vehicle driveline model. This gear shift schedule is characterized by late upshifts of gear as seen in Fig.8. Gear shift schedule designed for the optimal fuel economy minimizes the fuel consumption of the vehicle driveline as seen from the fuel economy values in Table I. This gear shift schedule is characterized by early upshifts of gear as seen in Fig.10. The simulation results show the effectiveness of gear shift schedules designed. The designed gear-shift schedules provide basis for further research work on shift schedule optimization

## APPENDIX A
### VALUES OF PARAMETERS AND CONSTANTS

| S/No. | Parameters | Values |
|---|---|---|
| 1 | Gear Ratios ($R_{TR}$) | [3.65,2.79,1.8,1.28,1] |
| 2 | Final Drive Ratio ($R_{FDR}$) | 3.6 |
| 3 | Wheel Radius ($R_w$) | $0.3m$ |
| 4 | Vehicle Mass ($m_v$) | $1100Kg$ |
| 5 | Frontal Area ($A_f$) | $2m^2$ |
| 6 | Density of ambient Air ($\rho_a$) | $1.199kg/m^3$ |
| 7 | Aerodynamic drag coefficient ($c_d$) | 0.3 |
| 8 | Engine Flywheel Moment of Inertia ($J_e$) | $2.7kg.m^2$ |
| 9 | Density of Diesel ($\rho$) | $0.832kg/litre$ |
| 10 | Efficiency of gearbox and differential | 0.9 |

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