# Full Vehicle Dynamic Model for PID Speed Controller Based on Practical Swarm Optimization Method

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**ABSTRACT**

The cruise control system CCS is considered to be an important system, because it is a fundamental component in autonomous vehicles. The CCS has high significant because CCS adds more autonomy to the vehicle. The CCS has many advantages such as making the driving less stressful and safer to the driver. In addition, it is important to the environment because it minimizes the fuel consumption and engine emissions. A full longitudinal vehicle dynamic model has been proposed in this work for forward motion. The model consist from drivetrain subsystem, resistance forces, and tire model. The significant in using full vehicle model gave a reality to the control performance specially tire model and slip ratio, and explained the vehicle behavior with/without the speed controller under different velocities and different conditions. The PID controller is being used for speed control system. The PID controller is widely used as its parameters are simple to adjust. Thus, many control systems especially controllers with feedback use PID controllers. Despite of its simple adjustment, it can be considered a very efficient controller, besides the system output response with PID controller become stable and reliable due its ability of linearization. The controller parameters which they are proportional KP, integral KI, and derivative KD are being tuned with different speeds for two times. The first by manual tuning and the second with using practical swarm optimization PSO algorithm. The overall performance of the proposed model has been evaluated with the two tuning methods at each desired speed. The experimental results show that PID based PSO gave better performance. With no overshot, zero steady state error, earliest raise time that 20 seconds earlier. This controller performance is desirable since it is difficult to adjust all the controller parameters at the same time. The PID based PSO Reaching the desired speed in a competitive time, furthermore it Reduce the instabilities that can be detected when the optimization is not used. The full model is being implemented in MATLAB/Simulink software environment

**Keywords:** dynamic model, cruise control, PID controller, PSO algorithm

## 1- Introduction

The enormous increase in the vehicle numbers worldwide made the transportation systems very difficult to control. Besides, the difficulty to prevent vehicle traffic accidents resulted from drivers' carelessness or other driving mistakes. The vehicle dynamics and control take huge attraction from vehicle manufacturers and researchers on different ways, specially the vehicle longitudinal motion because its represent to be the basic vehicle motion. The vehicle dynamics represent the understanding that the vehicle during moving which is the basic reason behind the vehicle founded, the vehicle trip must be done safely, comfortably, more environmentally friendly, and economically. The new supporting systems to the which provided the vehicle like cruise control, adaptive cruise control, parking assistant, line keeping and other Advanced Driver Assistant System ADAS systems are a group of systems achieved and still in development to enhance the vehicle rule. In addition they provide the safety conditions to the drivers which is the main concern, especially the vehicle speed control which depend on the vehicle longitudinal dynamics [1] there is much concern from the researchers and manufacturers due to the high range of accident because of the drivers excessive speed [2] The vehicle design modelling that emphasis on the longitudinal dynamics [3] [4] [5]and control is considered as one of the most important techniques in order to visualize and investigate all possibilities of the vehicle design. This helps to show how the vehicle workability and its interaction between its inner parts and the environment. The modeling using specialized soft wares can show interactions between the actuators and sensors where all modern vehicles are equated with them. Thus, the utilization of such kinds of software's shortens the time and the actual design costs of such vehicles. For this reason, using these soft wares can be an efficient way to guarantee an appropriate design that experiments all the expected scenarios. The vehicle speed control is represented by the popular PID controller which is proportional, integral, derivative controller that has widely been used in industry sector [6] almost 97% from the regulator controllers is classified as a PID controller [7]. Furthermore, PID controller is being used in the vehicles widely because it's simple application and suitable performance. Besides, it is easy to adjust and can be adjusted in real time[8].

In this work, a reliable, flexible and intelligent basic full dynamic vehicle model for longitudinal vehicle motion during acceleration with tire model for speed control is proposed. It accepts all possible updates easily. Also, it fits on different kinds of vehicles due to its easy installation and understanding. It can be said that it plug-in software. When it comes to the flexibility we can say that the proposed model can describe the vehicle movement during different rates of accelerations. Taking into account the environmental conditions such as aerodynamic drag force and rolling resistance force in addition to the climbing or grade force. Moreover, due to its easy parameter tuning, it can be said that the proposed model is flexible. It is noteworthy to mention that the model is an intelligent design because it uses the optimization techniques to enhance the performance and the stability of the vehicle while operation. In addition, the proposed model can easily describe the vehicle process from the power source i.e. the torque provided by the engine to its destination which is the tier. With a speed controller which represent as one of the Advanced Driver Assistant Systems ADAS which is growing fast in development to provide in the first safety for the driver and road. Furthermore, the pedestrians in addition to the traffic flowing smoothly, fuel economy and environmental protection from greenhouse gases. The traditional vehicle dynamic model for speed control is taking in account the aerodynamic drag force and rolling resistance. Neglecting all the nonlinear motion of the vehicle, but in this model a full dynamic model with tire model which represent nonlinear dynamics is being implemented for more realistic to the model and taking in account the longitudinal tire force and slip ratio with other forces during modeling the speed controller.

### 1.1. cruise control system

The recent cruise control system first developed by Ralph Teetor (1890-1982), this statement was reported at the magazine of Popular Science in 1954 which presented the system as a great and important innovation towards automatic driving. The speed limiter or regulator has great popularity with name

Cruise Control or Tempomat [9]. The cruise control consisted from upper level controller and lower level controller as can be seen in fig (1.1), the aim for the upper level controller is find the vehicle desired speed. While the aim of the lower level controller is to find the required real time vehicle throttle input that track the desired speed [10]. For the upper level controller which is the important part of the controller design, when the driver is set a desired speed the vehicle must converge to this speed with fast rise time, no overshot, and stable response.

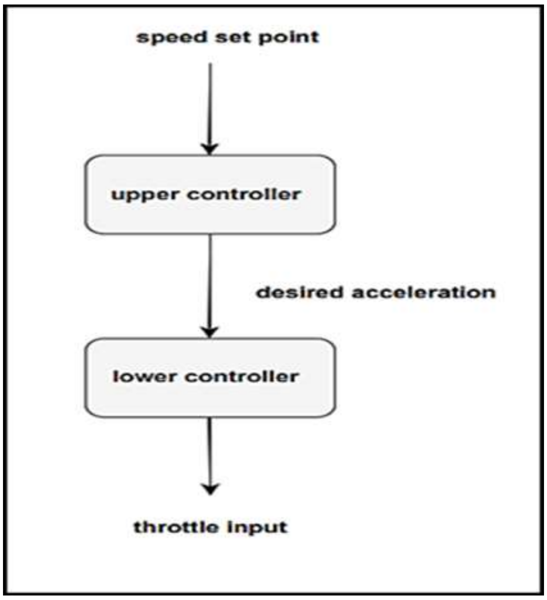


**Fig (1.1) Cruise control system structure[3]**

### 1.2. Typical Model Design

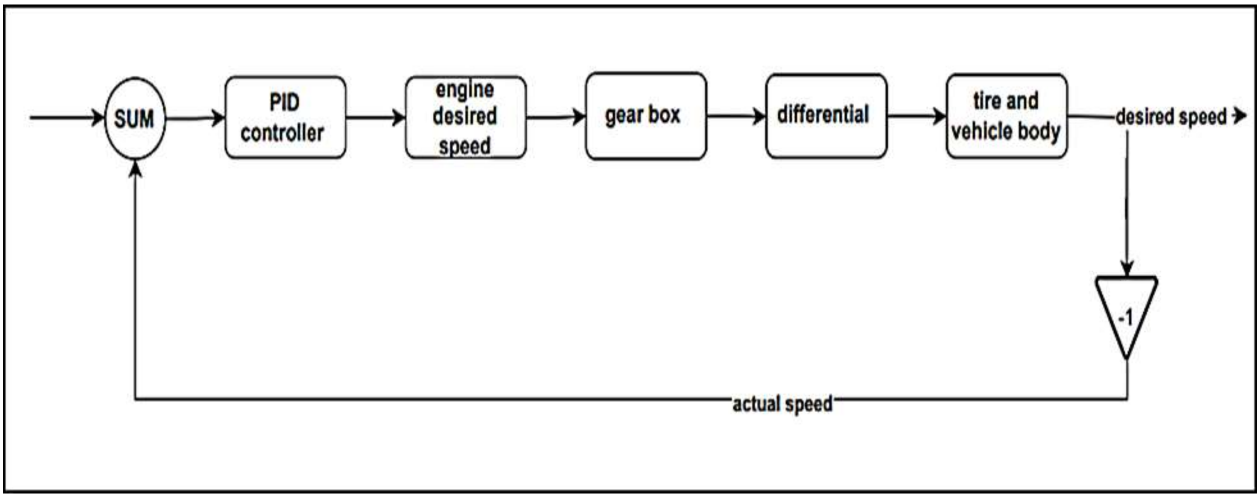


**Fig (1.2) the proposed longitudinal vehicle model**

The vehicle drivetrain consists of the engine (power source), driveshaft and gear reduction which represented by the gear box, and finally the differential in the drive axle, and axle shafts which represented by the differential which connect directly to the vehicle wheels. The vehicle dynamics model is simple and represented to be rigid so no internal losses can be caused due to the gearbox or power source is being counted, the initial vehicle movement is causing by the torque. The torque in turn is generated from the vehicle power source which is ICE internal combustion engine. The torque reaches the gearbox which is set or engaged on a specific degree that corresponds to the vehicle speed. In general, the vehicle gear ratio must be set on degree 3 or higher while the speed control is activated. Then the gearbox is coupled to the vehicle driving wheels through the differential. The differential is allowing the left and right wheels to spin at different speeds. After that the torque is reached to its final destination which is the vehicle body and tire. After finishing the mechanism of the vehicle dynamics, the vehicle speed control is applied which represent by PID controller to tracking the desired speed, to specify the desired speed, an actuator who adjust the throttle position on particular position to get the desired speed from the engine, by applying the speed controller which fed with the error signal that is generated from comparing the actual speed signal with the desired speed signal. After the controller being set for the appropriate value, the error will be minimized with stable signal value is generated to get the desired speed. For the control design preparations the wind velocity equals zero and the ground is flat Θ of the road is zero.

**2- Vehicle modeling**

**2.1. Vehicle powertrain**

In this suction the vehicle components which responsible for the vehicle motion include the power source engine which is torque produced, gearbox and differential which they are the vehicle powertrain and responsible for convert the torque and delivered it to the wheels are being explain.

**2.1.1 Internal combustion engine ICE**

The internal combustion engine is a device that produces energy by using fluid combustions, the combustion process carried out in a specific way to produce high pressure combustion products. These products can be expanded in a piston or turbine to produce energy that accelerate the vehicle [11]. The Internal Combustion Engines ICE is uncompetitive device in the field of high energy supply and wide range of uses

The approximate second order equation (3.1) can be used for engine torque estimation taking in concern the characteristics of the engine torque curve

$$Te = 528.7 + 0.152R - 0.0000217R^2 \qquad (2.1)$$

Where $R$ is the engine angular speed in revolution per minute RPM

The engine RPM can be estimated by

$$R = \eta g . \eta f . Wf \qquad (2.2)$$

Where $\eta g$ specific gear ratio for current drive , $\eta f$ differential ratio and, $Wf$ wheel angular velocity

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**.1.2. Gear box**

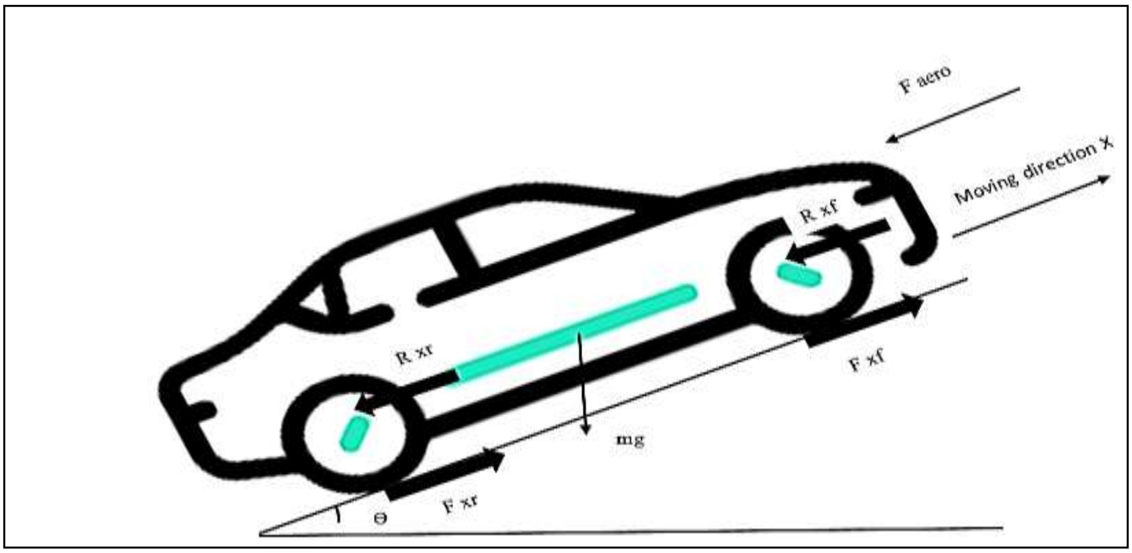
The engine total traction cannot be used unless an additional output converter be used because the maximum traction is limited due to the adhesion limits, the output converter aim to convert the engine characteristics to be as close as possible to ideal traction hyperbola. That is the reason behind using the gearbox. By using a gearbox the internal combustion engine potential power will be better.

**2.1.3. The differential**

It is a number of gears where arrange in a certain way to allow the two shaft rotating at different speeds. The differential is located on the vehicle rear axle (in rear drive axle). It allows the rear right and left wheels to rotate at different rates during curves [12]. When the vehicle is moving in curved road the external tire must rotate more than the inner tire to achieve the rotation correctly.

2.2. **Longitudinal vehicle dynamics**

The vehicle mechanical system uses the engine torque for feeding the drive axel with traction force, for sequence process next the mechanical power is born and transferred to the vehicle mass casing the vehicle moving, in the vehicle dynamics the vehicle velocity is being calculated from the equilibrium of the acting forces on the longitudinal vehicle axis including the longitudinal tire force.




The vehicle total traction force or vehicle acceleration is equaled the sum of the external resistance forces plus the longitudinal tire forces

$$Facc = Ftire + Faero + Froll + Fclimb \quad (2.3)$$

Where $F_{tire}$ is the longitudinal tire force for front and rear tire, $F_{roll}$ is the rolling resistance force, $F_{aero}$ is aerodynamic drag force, $F_{acc}$ is the acceleration force, $F_{climb}$ is the gravitational force with road angle, m is the vehicle mass, and $\ddot{X}$ is the vehicle acceleration

**2.2.1 Aerodynamic drag force**

When an object is moving during a substance or fluid a resistance force trying to counteract it motion, just like a moving vehicle through the air the resistance force will be the aerodynamic drag force. This force represents all the interactions between the vehicle body and the air. The aerodynamics drag force is calculated depending on this formula

$$Faero = \frac{1}{2}\rho CdAf(Vx + Vwind)^2 \quad (2.4)$$

Where $\rho$ is the mass density of the air which equal to 1.225Kg/m$^2$, $Cd$ is the aerodynamic drag coefficient, $Af$ is the projected vehicle front area to the travel direction, $Vx$ longitudinal vehicle velocity, $V\ wind$ is the velocity of the wind which is positive for the head wind and for tail wind is negative.

For estimation the vehicle front area the formula below (3.6) is being implemented for the vehicle mass rag 800-2000 Kg [3].

$$Af = 1.6 + 0.00056(m - 765) \quad (2.5)$$

**2.2.2. Rolling resistance**

When the tire is rotating, it is being deformed vertically each time when it touch the grounds during its rotation. The contact patch is continuously deformed and reshaped to its original shape. Due to the normal load, the tire material deflects in vertical direction as it goes through the contact patch and springs back to its original shape continuously. Due to the internal damping of the tire material, the energy spent in deforming and reshaping is not completely recovered. This loss of energy can be represented by a force which acts on the tires called the rolling resistance. Which is the force acting in an opposite direction of the vehicle movement. In other words it is the magnitude of the force needed to overcome the friction force generated between the tire and the road [3].

the rolling resistance force can easily calculated by multiplying the rolling resistant coefficient Crr by the sum of normal forces to the plane taking in consideration the bank angle of the road .

$$Rx = Crr.m.g.\cos(\theta)$$

Where $Crr$ is the rolling resistance coefficient, $M$ is the vehicle mass, $g$ is the earth gravity which equal to 9.81, $\theta$ is the road angle

**2.2.3. Climb resistance**

The grade resistance (climb resistance) is one of the resistance forces. The vehicle must override this force in order to let the vehicle move forward.

$$Fg = m.g.\sin(\theta) \quad (2.7)$$

Where $M$ is the vehicle mass, $g$ is the gravity factor, $\theta$ is the road angle

**2.3. Tire dynamics and tire forces**

In this section the details of dug-off tire model is being discussed in addition to the mathematical equations to calculate the tire forces and other factors. The dug off tire model is an analytic model which being used to calculate the road- tire forces by analysis the longitudinal and lateral tire dynamics. The dug-off tire model represents the tire behavior which is in steady state with uniform pressure distribution in vertical direction inside the tire.

**2.3.1. Tire angular acceleration**

The tire angular acceleration must be calculated to estimate the tire angular velocity which is the integrate value for the angular velocity.

$$\dot{\mathrm{W}}w = \frac{1}{Iw}Twheel - \frac{Reff}{Iw}Fx \quad (2.8)$$

$$\mathrm{W}w = \int \frac{1}{Iw}Twheel - \frac{Reff}{Iw}Fx \quad (2.9)$$

Where $I_w$ tire mass moment of inertia, $T_{wheel}$ is the torque act on the wheel which comes from the engine or electric motor, $F_x$ the force acting on the tire in longitudinal direction which comes from the ground, $R_{eff}$ it is the effective tire radius, $\dot{\mathrm{W}}w$ is the tire angular acceleration.

**2.3.2. Tire slip ratio**

The dug-off tire model for nonlinear motion will be used for estimating the tire forces. The slip ratio is being calculated as shows in equation (3.12) during acceleration mode

$$\sigma_x = \frac{Reff \times \omega_w - V_x}{Reff \times \omega_w} \quad (2.10)$$

Where $Reff$ is the effective tire radius, $W_w$ is the tire angular velocity, $Vx$ the velocity of center of the tire which is equal to the vehicle velocity

**2.3.3. Longitudinal tire forces**

The dug-off tire model is being used to calculate the longitudinal tire forces during acceleration.

$$\mathrm{Fx} = C6\frac{6x}{1+6x}f(\lambda) \quad (2.11)$$

Where $C6$ tire stiffness, $6x$ is the tire slip ratio, $F(\lambda)$ is a function which used in the dug-off tire model

If ( $\lambda$ )< 1 then

$$f(\lambda) = (2-\lambda)\lambda \quad (2.12)$$

If ( $\lambda$ )>= 1 then

$$f(\lambda) = 1 \quad (2.13)$$

λ is being calculated according to the formula below

$$\lambda = \frac{\mu Fz(1+6x)}{2(C66x)} \qquad (2.14)$$

Where $\mu$ is friction coefficient between the tire and ground, $Fz$ is the tire normal force, $C6$ is the tire stiffness, $6x$ is the tire slip ratio.

## 2.4. PID speed control

The PID controller is widely used as its parameters are simple to adjust. Thus, many control systems especially controllers with feedback use PID controllers. Despite of its simple adjustment, it can be considered a very efficient controller [13]. A PID controller is very useful kind and is capable of solving wide range of problems. More than 95% of all industrial control problems are solved by PID control, besides the system output response with PID controller become stable and reliable due its ability of linearization. The PID controller contains three terms: (1) proportional KP, (2) an integral KI and (3) derivative KD. The past signal represented by error integral KI, the proportional term KP represent the present signal and the future is being represented by the derivative KD which is the linear extrapolation of the error[14]. The time domain output equation for the controller is:

$$u(t) = KPe(t) + \mathrm{KI}\int e(t)\,dt + KD\,\frac{de(t)}{dt} \qquad (2.15)$$

## 2.5 Practical swarm optimization method PSO

This algorithm is an optimization algorithm that is being developed in 1995 by Dr. Kennedy and Dr. Eberhart. PSO depends on the behavior of fish school or bird swarms within a flock. In this method the swarm is assumed to have a certain size, with each particle starting position at a random location in multidimensional space [15]. Each particle in the swarm of the PSO algorithm has two vectors which are the position vector and the velocity vector.

The vectors updated consequently depending on the fallowing equations [16].

$$vi^d(t+1) = w.vi^d(t) + c1.r1.(pbesti^d - xi^d) + c2.r2.(gbesti^d - xi^d) \qquad (2.16)$$

$$xi^d(t+1) = xi^d(t) + vi^d(t+1) \qquad (2.17)$$

Where $c_1, c_2$ are two constants responsible for the acceleration, $r_1$, $r_2$ are two random numbers which uniformly distributed in the range [0,1], w is the weight of inertia which is important to balance between the search past direction and current direction., $\mathrm{pbest}^d$ is the $i$th particle previous position, $\mathrm{gbest}^d$ is the best global particle which has been discovered in entire population, $Vi^d$ is the velocity, $Xi^d$ is the position

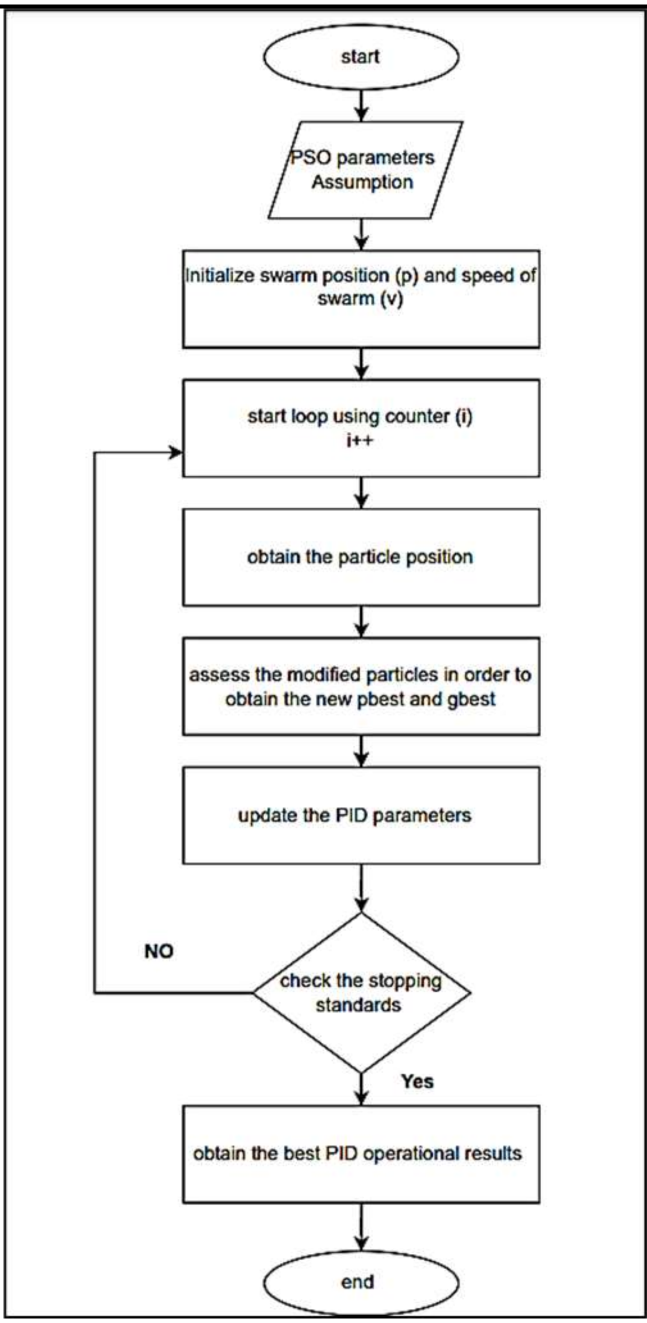



**Fig (2.2) flow chart sketch for PSO algorithm**

## 3- Simulink and results

### 3.1. Proposed Model Components

The full dynamic model with tire model is being modeled to implement the PID speed controller. The vehicle parameters for proposed model are listed below.

**Table (3.1) the parameters of the vehicle for proposed model**

| Vehicle parameter | Value | Unit |
|---|---|---|
| Vehicle mass | 1470 | Kg |
| Effective tire radius | 0.31 | M |

| | | |
|---|---|---|
| Earth gravity | 9.81 | m/s$^2$ |
| Rolling resistance coefficient | 0.012 | |
| the aerodynamic drag coefficient Cd | 0.29 | |
| ρ the mass density of the air | 1.225 | Kg/m$^3$ |
| Tire stiffness | 40000 | N/degree |
| Gear ratio | 0.83 | |
| Differential ratio | 2.82 | |
| Engine efficiency | 97% | |
| Tire mass | 42 | kg |
| Tire –road coefficient μ | 1 | |
| Ɵ theta road angle | zero | Degree |
| Wind velocity | Zero | |

Figure (3.1) illustrates the the suggested model. It has the vehicle drive train (power source) and the rolling resistance, aerodynamic drug force and climbing resistance, besides the dug-off tire model is being used to calculate the tire angular acceleration, the tire angular velocity, and the tire slip ratio are being estimated to calculate the tire forces. the output of these forces after finding the yield of them. This yield value represents the vehicle acceleration. The integration of this value represents the vehicle velocity in the longitudinal direction. In order to discover the distance passed by the vehicle, one can integrate the velocity. This is because the distance is the first integration of the velocity while it represents the second integration of the acceleration.

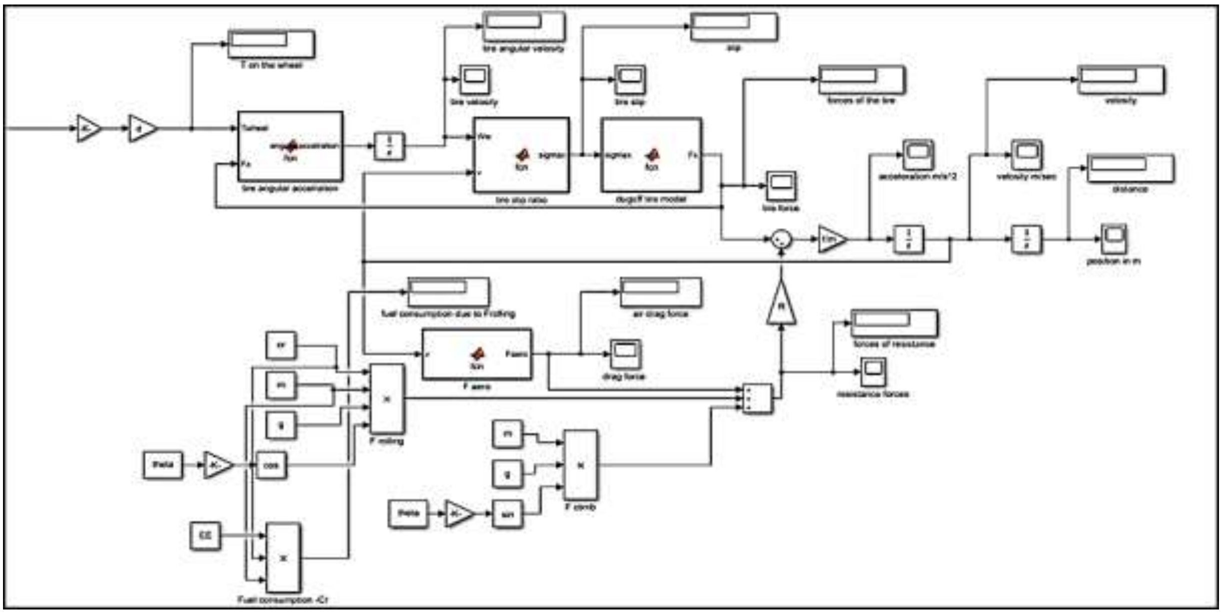



**Fig (3.1) the full dynamic vehicle model with tire model**

The speed controller or cruise control system is one of the fundamental systems in modern vehicles. The speed controller is represented by PID controller which feedback with the error signal between the actual vehicle velocity and desired velocity. Then the PID controller being set on particular value which make the error eliminate to generate the desired speed. This study will explore the full dynamic model for PID speed control with manual tuning once and again with using the PSO optimization algorithm. The figures (3.2) and (3.3) are explain the full dynamic model with PID speed controller and PID based on PSO optimization algorithm respectively.

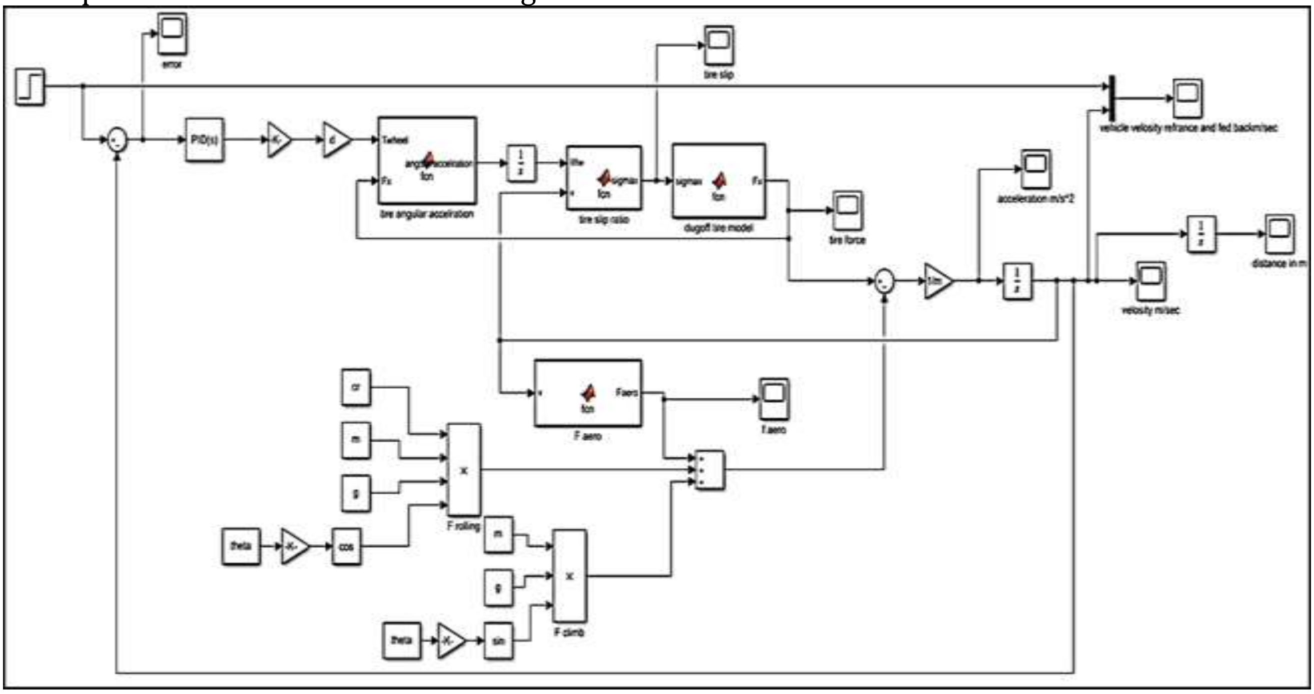



**Fig (3.2) the full dynamic vehicle model with tire model and speed controller**

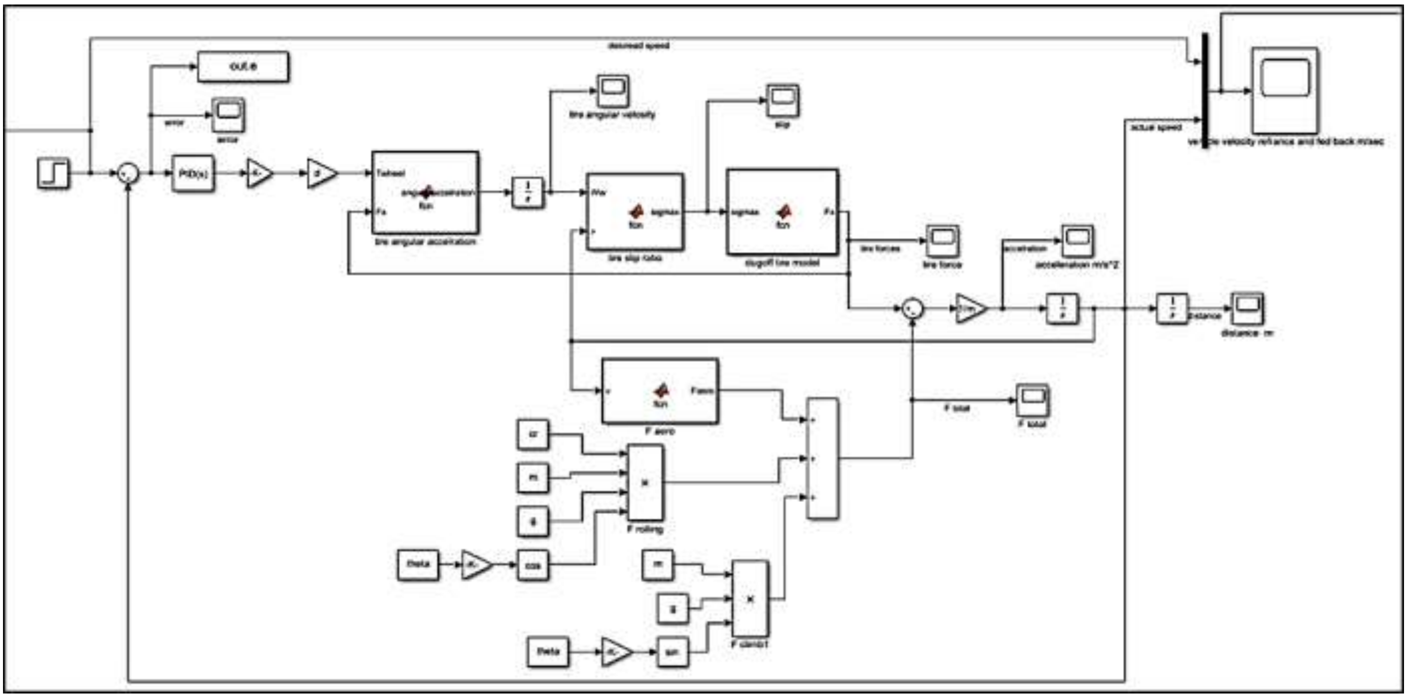



**Fig (3.3) the full dynamic vehicle model with tire model in addition to speed controller and PSO optimization algorithm**

**3.2 Results**

in this sub suction the influence of changing the model parameters on the model performance and related factors are explain in fallowing tables and charts .

**3.2.1. Influence of difference Cd values on the vehicle dynamicity**

The relationship between the Cd aerodynamic drag coefficient on the vehicle velocity, tire velocity and the distance is given here in this subsection. As can be noticed from the table (3.2) there is a direct relation between the Cd aerodynamic drag coefficient and the vehicle velocity. In other words, when the value of the Cd aerodynamic drag coefficient is lower the vehicle velocity becomes higher i.e. it is an inverse relationship. As for the tire velocity the same effect is realized. In turn the vehicle distance will be affected by this relationship.

**Table 3.2 Influence of difference of Cd on the vehicle parameters**

| cd | Velocity m/s | tire angular velocity m/s | Distance m |
|---|---|---|---|
| 0.15 | 85.41 | 276.1 | 3.87E+04 |
| 0.2 | 73.97 | 239.1 | 3.42E+04 |
| 0.35 | 55.92 | 180.9 | 2.66E+04 |
| 0.4 | 52.3 | 169.7 | 2.50E+04 |
| 0.5 | 46.78 | 151.1 | 2.26E+04 |
| 0.8 | 36.98 | 119.5 | 1.81E+04 |

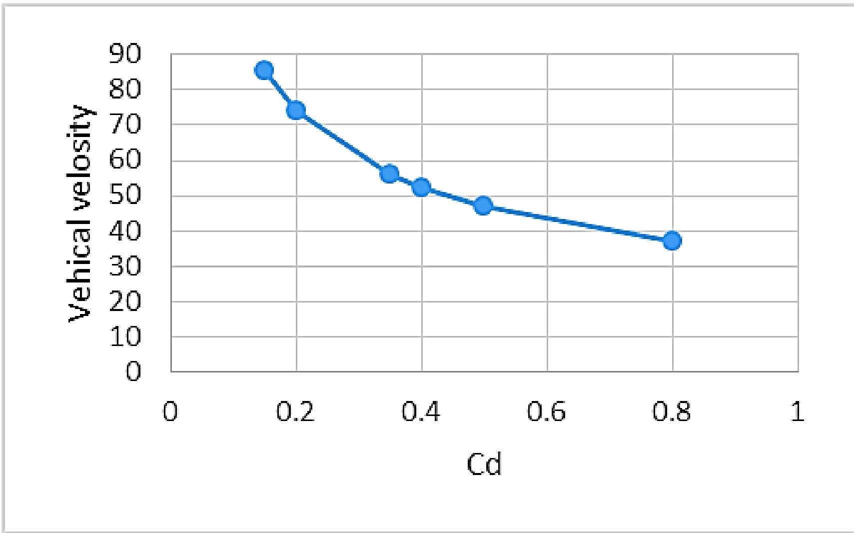



**Fig (3.4) the influence of variation of Cd on the vehicle velocity**

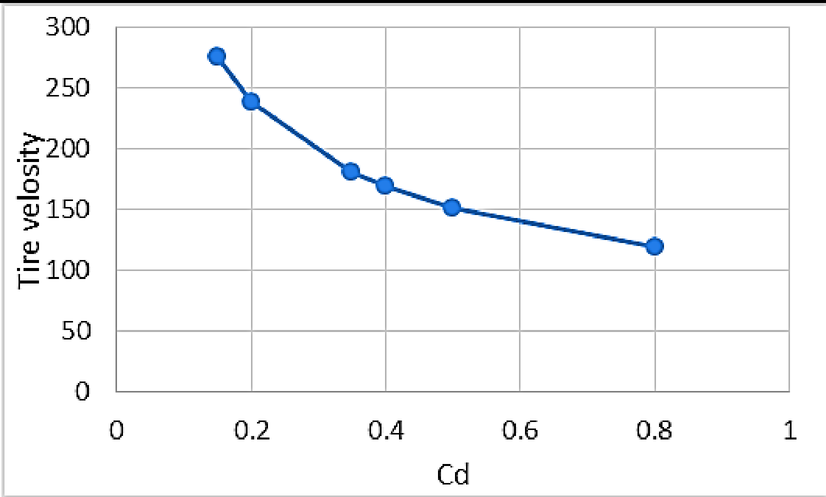



**Fig (3.5) the influence of variation of Cd on the vehicle tire velocity**

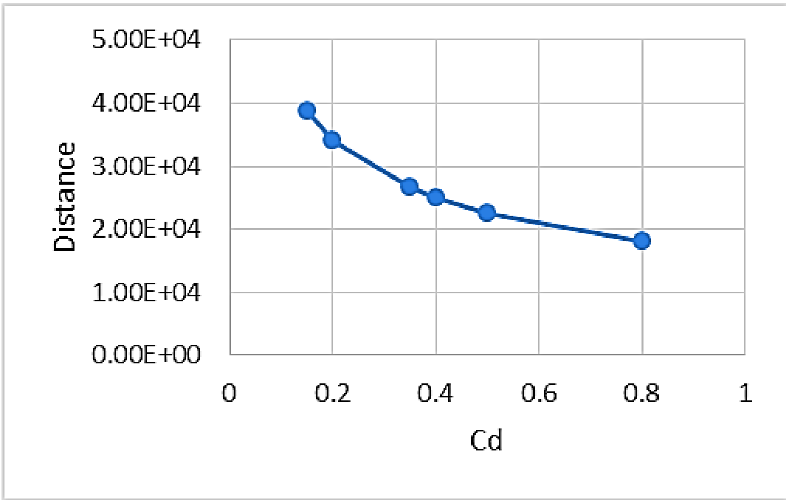



**Fig (3.6) the influence of variation of Cd on the distance the vehicle reach**

### 3.2.2. Influence of difference of Cr on the vehicle parameters

The Cr rolling resistance coefficient is considered the basic factor that affects the fuel consumption. The relationship between the Cr rolling resistance coefficient on the vehicle velocity, tire velocity, the rolling force and the fuel consumption can be realized in the table (3.3) there is an inverse relation between the Cr rolling resistance coefficient and the vehicle velocity differently speaking, when the value of the Cr rolling resistance coefficient is less, the velocity becomes more. For example, when Cr rolling resistance coefficient 0.012, it can notice that the vehicle velocity becomes 61.42, simultaneously when the value of this coefficient equals 0.035, the vehicle velocity becomes 53.23. The Cr rolling resistance coefficient has the same impact on the tire velocity. In turn the vehicle distance will be affected by this relationship.

**Table (3.3) the influence of difference of Cr on the vehicle parameters**

| Cr | Tire angular velocity | Vehicle velocity | Rolling force | Fuel consumption |
|---|---|---|---|---|
| 0.012 | 199.1 | 61.42 | 1.73E+02 | 17.11 |
| 0.015 | 195.1 | 60.43 | 2.16E+02 | 21.39 |
| 0.02 | 188.8 | 58.73 | 2.88E+02 | 28.52 |
| 0.025 | 184 | 56.96 | 3.61E+02 | 35.65 |

| | | | | |
|---|---|---|---|---|
| 0.03 | 178 | 55.15 | 4.33E+02 | 42.78 |
| 0.035 | 172.1 | 53.27 | 5.05E+02 | 49.91 |

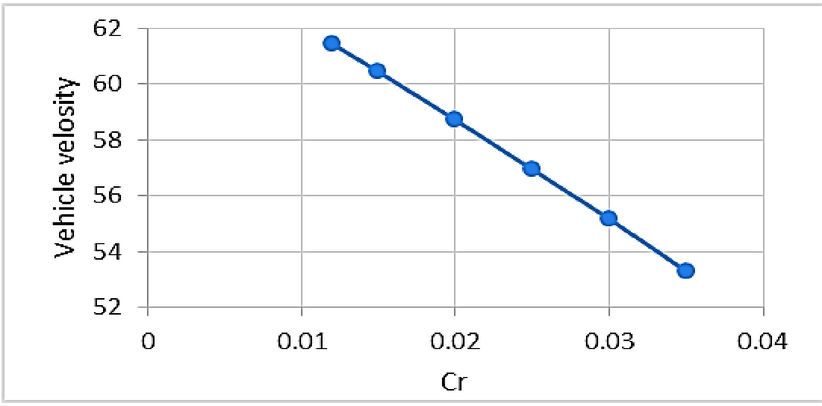



**Fig (3.7) the influence of variation of Cr on the vehicle velocity**

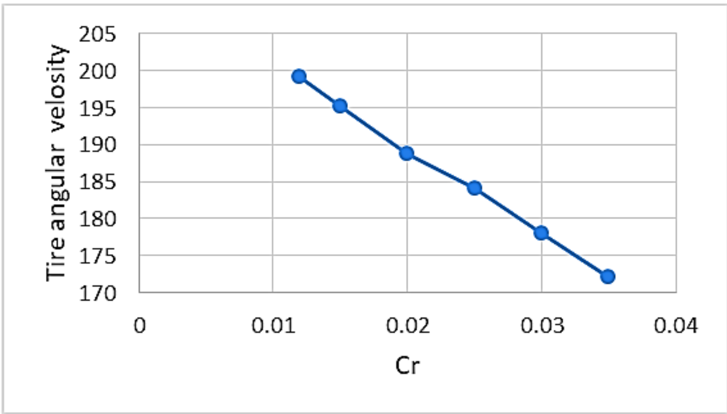



**Fig (3.8) the influence of variation of Cr on the tire angular velocity**

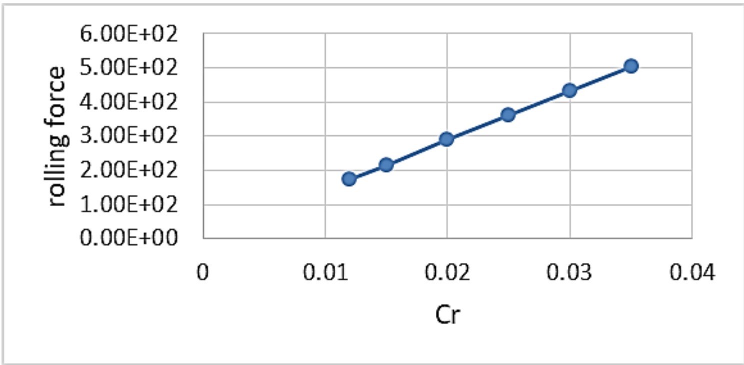



**Fig (3.9) the influence of variation of Cr on rolling resistance force**

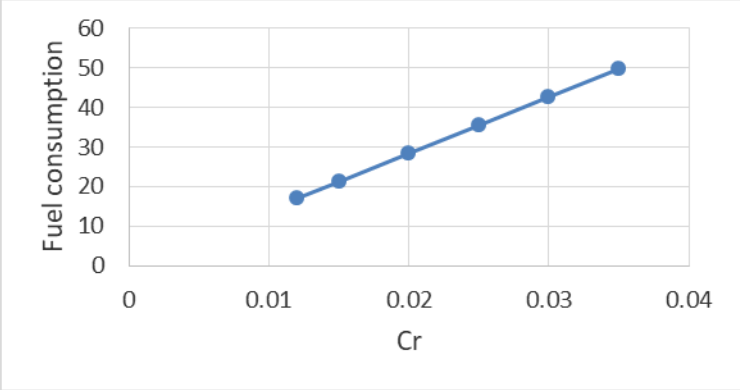



**Fig (3.10) the influence of variation of Cr on fuel consumption**

### 2.3.3. Influence of speed variation on vehicle parameters

The table (3.4) shows the impact of the vehicle's speed increment on the F drag force and the total resistance forces. For instance, when the vehicle velocity is 61.42 m/s it can be noticed that the values of the F aero (1337N) and the total resistance forces (1510N). On the other hand, when the vehicle velocity is 100.9 m/s it can be realized that the parameter's values of the F aero (3605N) and the total resistance forces (3778N). Obviously that the increment of vehicle velocity lead to an increment in F drag force since it depends on the vehicle force, furthermore, it can be seen that the F drag force represents a big percentage of total resistance forces.

**Table (3.4) the influence of vehicle speed difference on the vehicle parameters**

| Vehicle velocity m/s | F DRAG N | Total resistance force N |
|---|---|---|
| 61.42 | 1337 | 1510 |
| 76.84 | 2092 | 2265 |
| 89.63 | 2847 | 3020 |
| 100.9 | 3605 | 3778 |
| 110.9 | 4360 | 4533 |
| 137.6 | 6712 | 6885 |

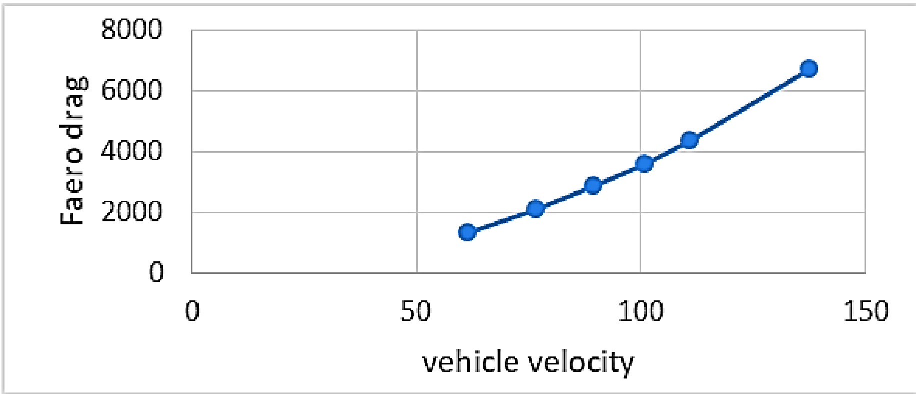



**Fig (3.11)** the influence of increasing in vehicle velocity to air drag force

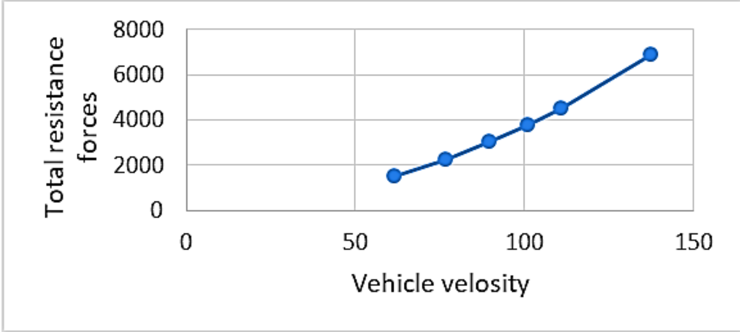



**Fig (3.12)** the influence on increasing velocity on the total resistance forces

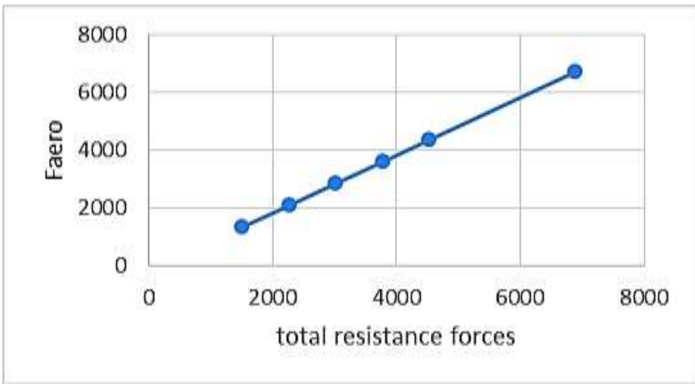



**Fig (3.13)** the relationship between the total resistances forces with the Faero

### 3.4. Full vehicle dynamic model for PID speed control

The full dynamic model is being designed with tire model which represent the contact point between vehicle and ground. The nonlinear motion must be nullified or at least minimized due to the fact that this kind of motion might cause an instability in the vehicle behavior while running.

Although tire model can give more reality to the model and give a wide look on the impact of the controller on the vehicle behavior, adding the tire model can be considered as a nonlinear motion. It can give reality to the model especially when considering the slip ratio. When the controller is added to the vehicle's model and when it runs above gear 3, the tire slip ratio becomes less as the longitudinal motion involved in speed control are very gentle. The observation of the slip ratio is considered very important factor as a controller performance measurement. The model is being tested in three cases: the first case is without controller, the second is with a manual tuning for PID controller and finally the PID controller is being accompanied with PSO optimization component. In the following subsections these three cases are explained in details.

#### 3.4.1. without speed controller

The vehicle model showing a results without speed controller that the amount of torque input is not equal to the speed output. In this case there is a high variation between the two values i.e. the torque input and the speed output.

The fig (3.15) illustrate the vehicle speed 138 m/s in open loop maneuver, while fig (3.16) shows the vehicle acceleration, this scope describes that the vehicle speed still increasing until reach the steady state zone in almost 100 seconds. The fig (3.17) The tire slip ratio is keeping in increase without any limitations or regulation.

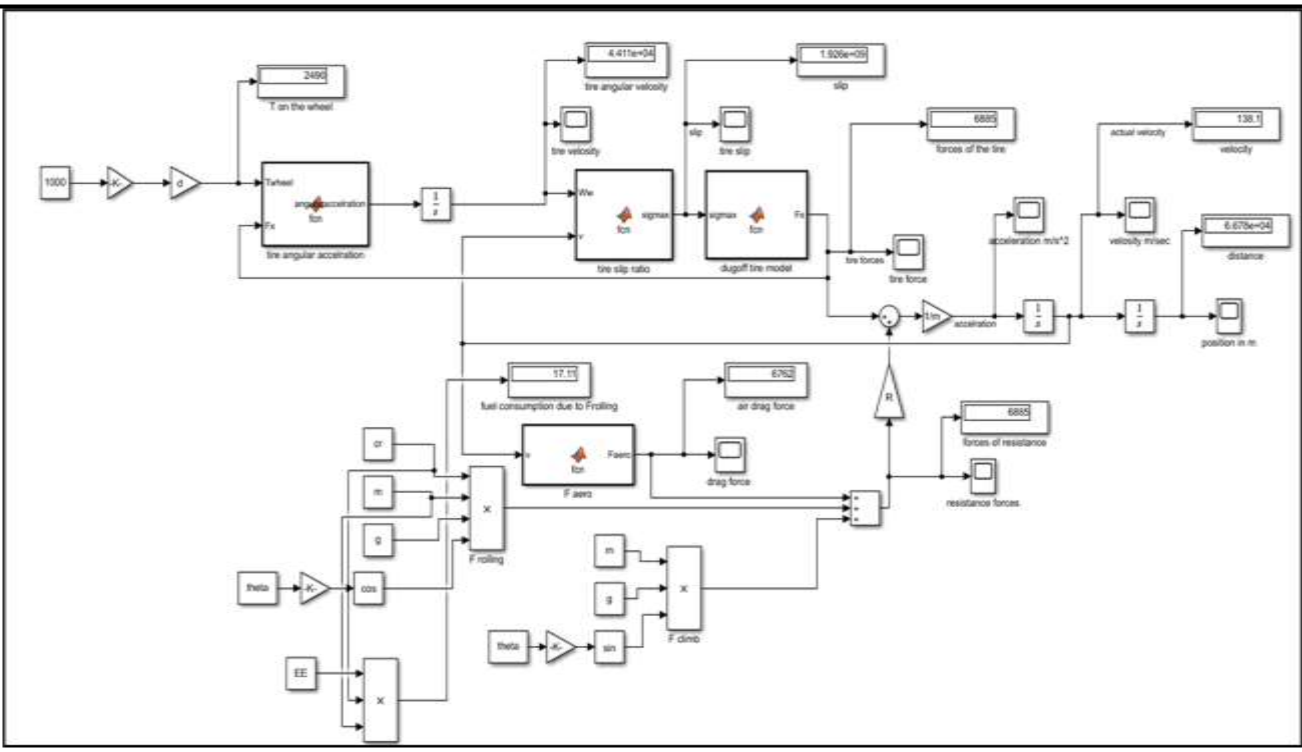



Fig (3.14) the full dynamic model for speed testing before applying PID speed control with speed 138 m/s

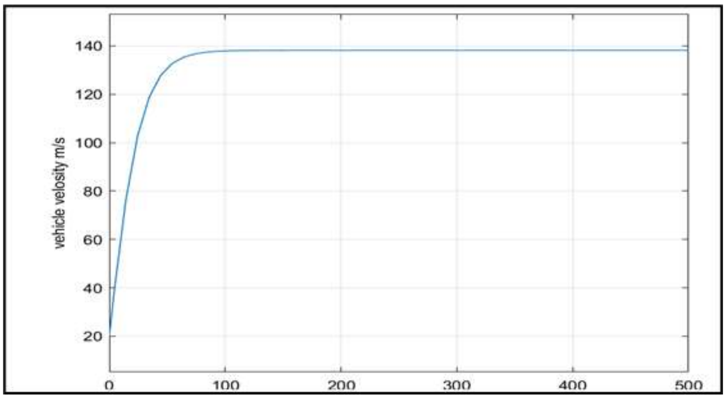



Fig (3.15) the vehicle speed 138 m/sec without speed control

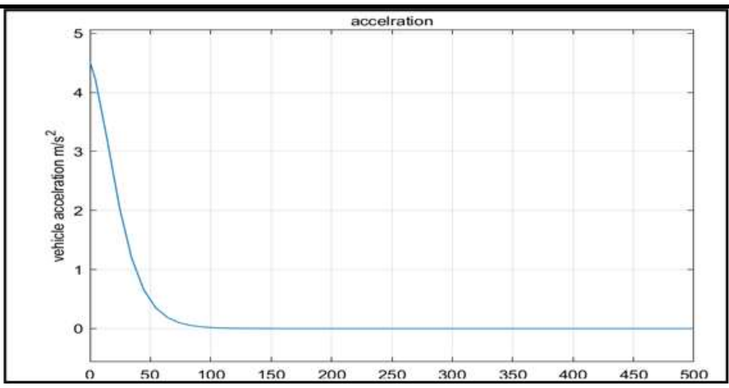



**Fig (3.16) the vehicle acceleration with speed 138 without PID speed control**

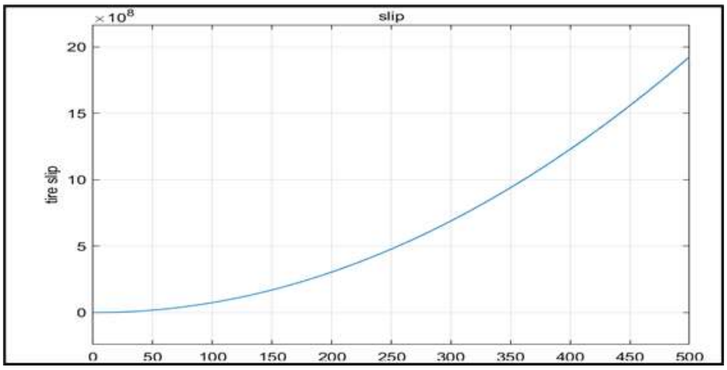



**Fig (3.17) the vehicle slip ratio when vehicle speed is 138m/s without PID controller.**

### 3.4.2. The speed control with manual tuning

The PID controller is applied to the full longitudinal vehicle dynamic model and tuned depending on trial-and-error method. The fig (3.18) shows the scope for desired speed with the step response. The controller which feedback with error signal is trucked the desired speed. After almost 150 sec the desired speed 50m/s is set. As can be seen in fig (3.19) the over shot is very small between the desired speed and the peak time which almost 2, the settling time is relatively long 140 s. the peak time is 60 s and rise time is 40s.

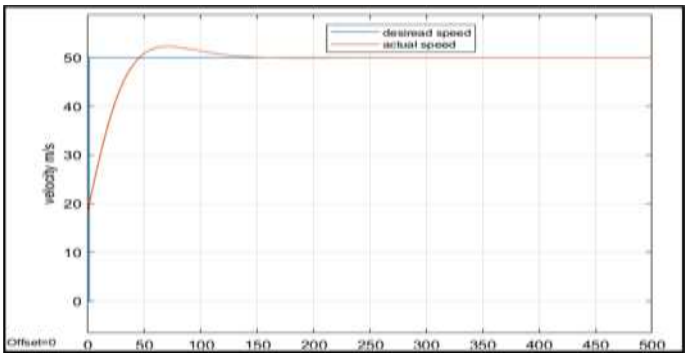



**Fig (3.18) the desired speed step input with actual vehicle speed response with manual tuning for PID speed control with velocity 50 m/s**

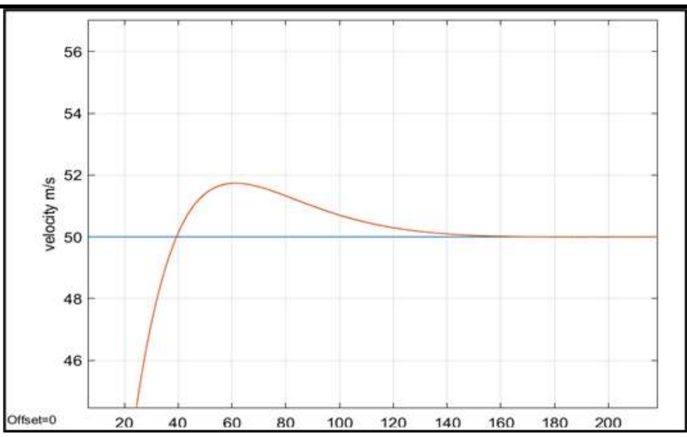



**Fig (3.19) close view to the vehicle speed PID controller with manual tuning**

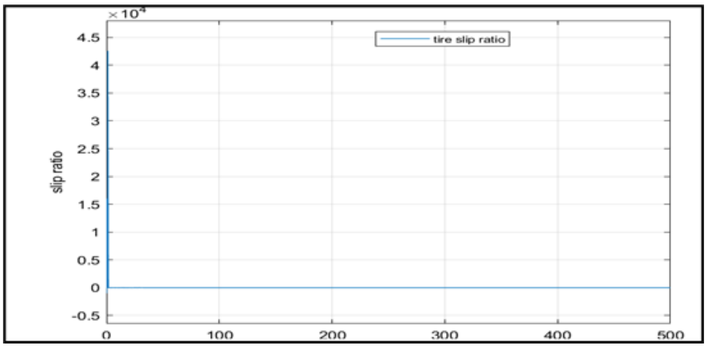



**Fig (3.20) tire slip ratio PID speed control with manual tuning**

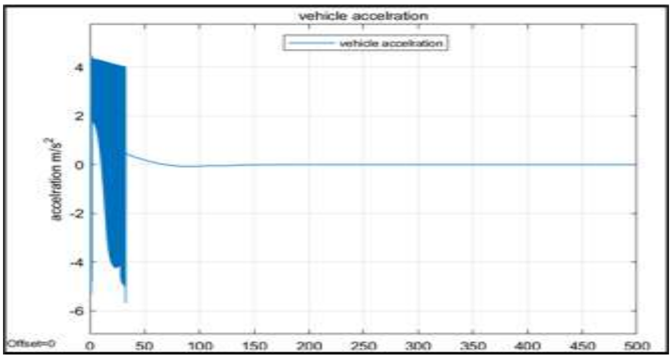



**Fig (3.21) the vehicle acceleration with PID speed control with manual tuning.**

Fig. (3.20) shows the tire slip ratio for the vehicle when PID controller is applied with manual tuning. The slip ratio becomes very small compared with the vehicle model without the PID controller. When the controller applied, the longitudinal motion of vehicle in the forward direction becomes very gentile. For that reason the tire slip ratio becomes very small. Since the brakes are not used in this mode, the slip ratio become almost zero. This indicates that the mean difference between the actual velocities of the wheel excel and the equivalent tire rotational is almost zero.In Fig. (3.21) it can be noted that the acceleration after almost 150 seconds becomes zero which is the normal performance because there is no change of vehicle speed in both magnitude and direction therefore, the acceleration must equal to zero.

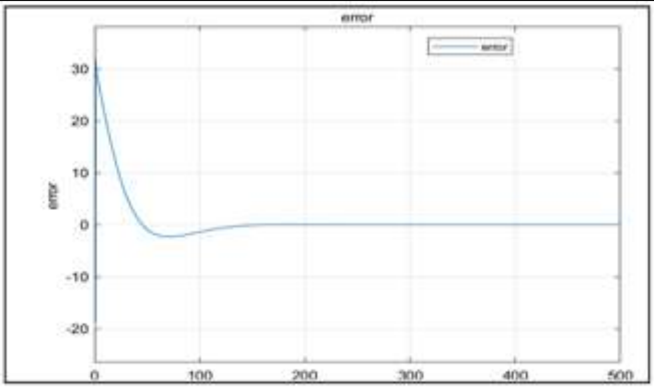



**Fig (3.22) the error signal with PID speed control with manual tuning**

In fig (3.23), the error signal shows that the PID speed controller is tracking the desired speed, and with almost 150s the steady state error become zero. This performance indicates that the vehicle controller, according the manual tuning, is working with little error percentage within 150 second.

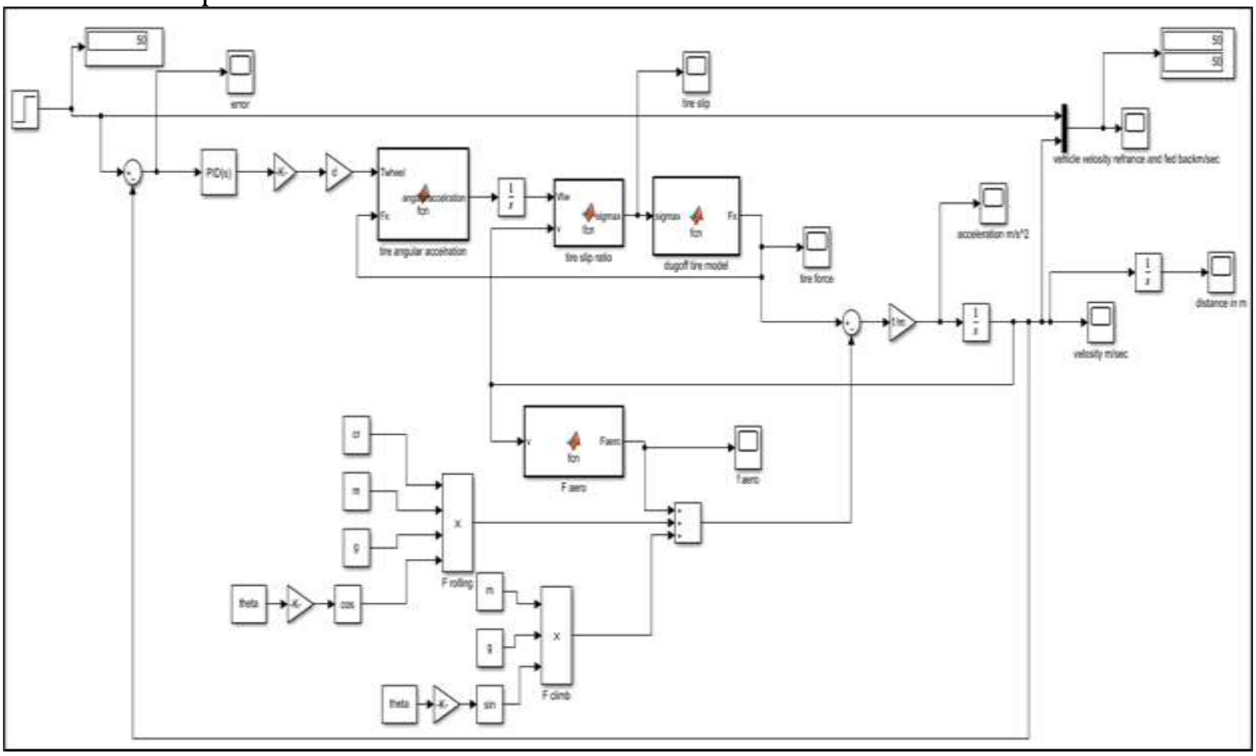



**Fig (3.23) the vehicle model with PID speed control the controller is tracking the desired speed which is 50 m/s total Simulink time is 500 sec**

**Table (3.5) the controller values for manual tuning**

| Tuning | Desired speed | KP | KI | KD |
|---|---|---|---|---|
| Manual | 50 m/sec | 12.4 | 0.52 | 6.8 |

### 3.4.3. The speed control based PSO optimization

The figure (3.24) illustrates the effect of using the optimization using the PSO. It can be noted that the utilization of PSO help the system to be stabilize earlier than using the PID controller alone. This can be observed in the figure (3.24) that both the desired speed and the actual speed that been reached to the matching point much earlier than the case of the controller was used without optimization. As another observation, it can also be realized from the same figure that both speeds has been matched exactly all over the vehicle run time. In contrast to the other case where there is no optimization used, an instability regions has been detected while running which indicate to an instability in operation. The fig (4.32) shows that there is no over shot in the PID performance besides no steady state error and faster raise time which almost 17 s.

To recap, one can mention that the optimization search can be used as an efficient tool that help to do the following two objectives:

1- Reaching the desired speed in a competitive time.

2- Reduce the instabilities that can be detected when the optimization is not used.

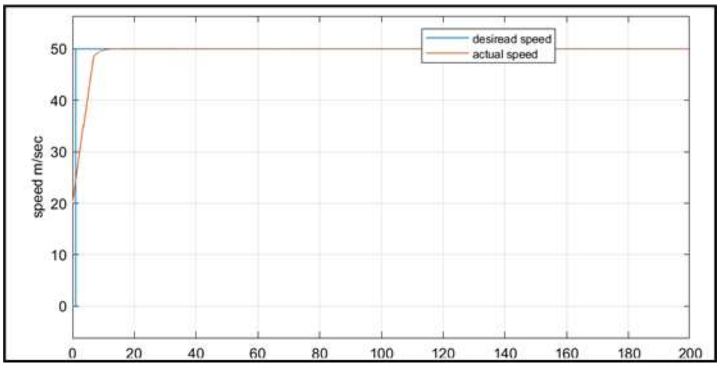



**Fig (3.24) the desired speed step input with actual vehicle speed response with PSO for PID speed control velocity 50 m/s**

In both figure (3.25) and figure (3.26) show the slip ratio and error carvers while running with PID controller that is equipped with the PSO component. In figure (3.25) the slip ratio can be seen to be minimized fast during the running time. Similarly, the fig (3.26) error signal has also been zeroed while running. The minimization of both values can be considered as an indication to the efficiency of using the optimization with the PID controller in this context.

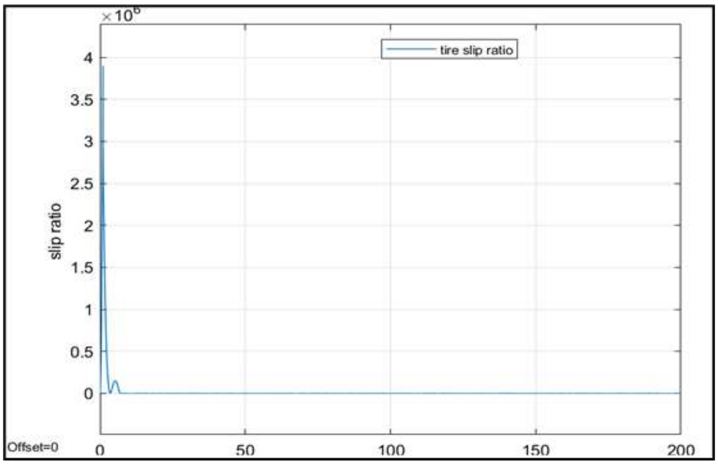



**Fig (3.25) the slip ratio with PSO for PID speed control with velocity 50 m/s**

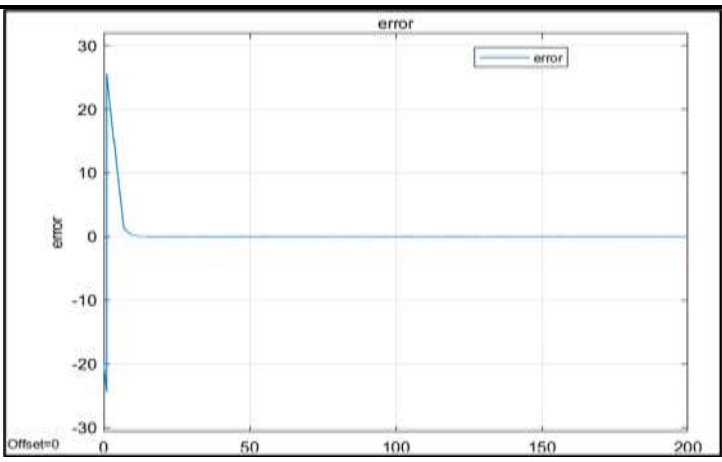



**Fig (3.26) the error signal with PSO for PID speed control for velocity 50 m/s**

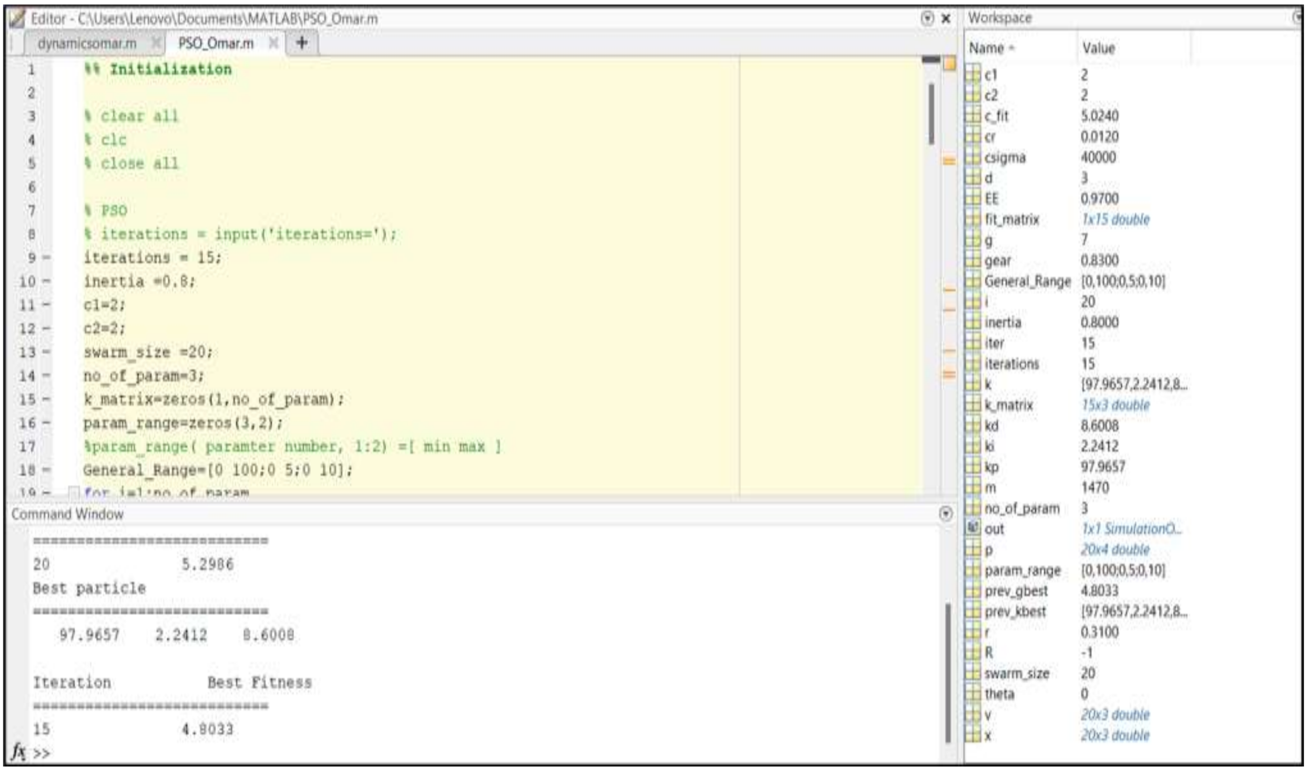



**Fig (3.27) MATLAB code that used to generate PSO solutions has been screenshot**

**Table (3.6)** the parameters of the PSO for optimal solution with desired speed 50m/s

| Parameter | Value |
|---|---|
| Iterations | 15 |
| Inertia | 0.8 |
| C1,c2 | 2,2 |
| Swarm size | 20 |
| KP | 97.9657 |
| KI | 2.2412 |
| KD | 8.6008 |
| KP range | 0-100 |
| KI range | 0-5 |
| KD range | 0-10 |

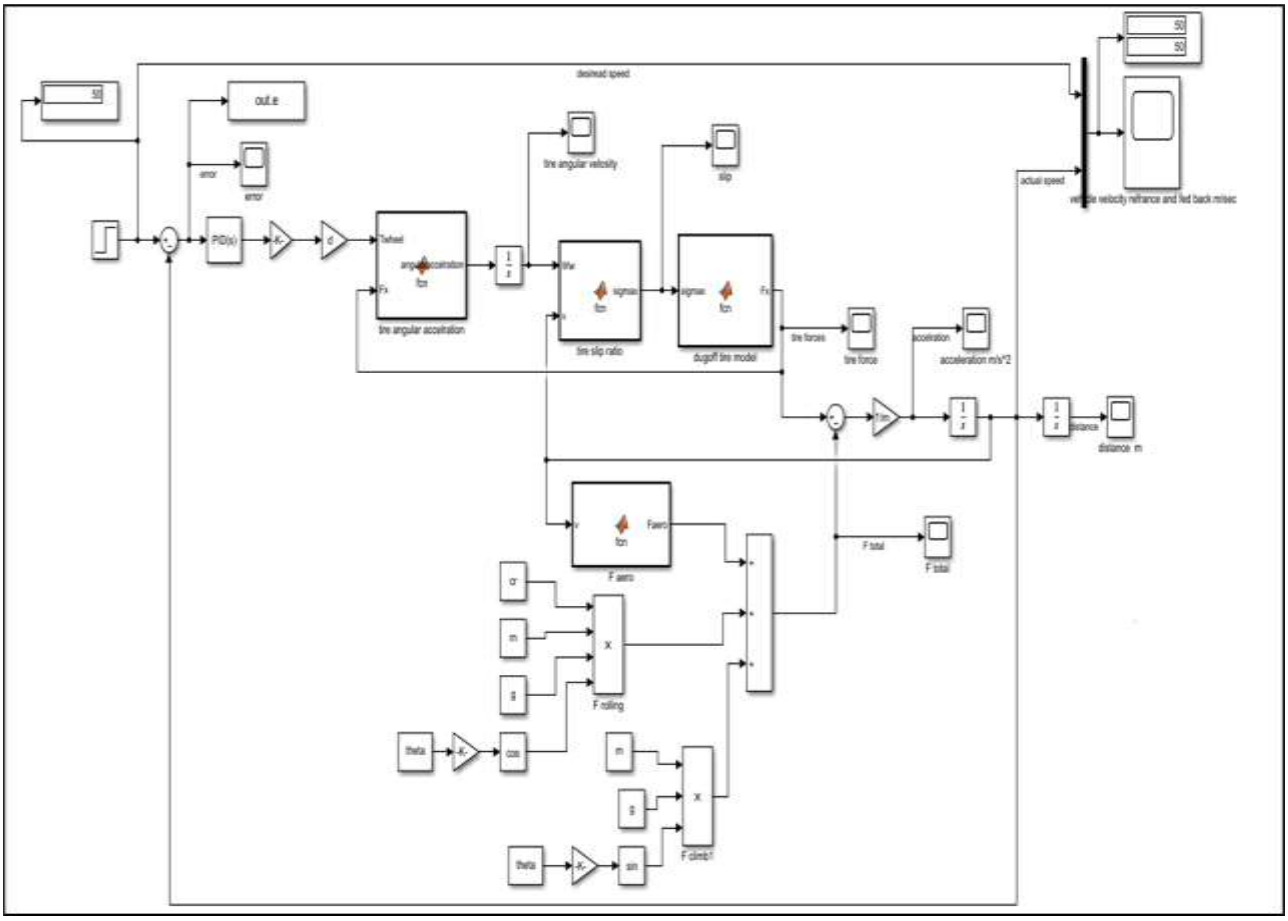


**Fig (3.28)** the PSO working window with the optimal solution for 50m/s desired speed

## 3- Conclusion

In this study full dynamic model has been modeled for PID speed controller based PSO, the proposed model has been modeled in MATALB/ Simulink software environment. The study is being developed according fallowing steps

1- A full dynamic model which represent the vehicle longitudinal motion in forward direction is being developed, the model consists of three subsystems: the first subsystem is composed of the (i) engine, (ii) gear box and (iii) differential. The second represents the resistance force. The third is the tire model. The output of this system is the vehicle speed. The model is being tested under different aspects and variation values. This dynamic model represents a general model that can be used for any type of vehicle to calculate the vehicle speed and other forces by updating the vehicle parameters very easily.

2- The tire model is being added to the model in order to give a reality performance to the model besides to calculate the tire forces and slip ratio.

3- The full dynamic model is being experimented by changing one parameter and keep the others constant for many times, to explain the dynamic model behavior. Later the data has been transmitted to the excel sheets and charts to explain the relationship between the model parameter, as shows in related tables and charts.

4- The full model is being modeled for applying speed control system by PID control which is representing to be a required system due to important in enhancing the driving and environment besides less the driving fatigue and tasks on the driver. The model is being tested for improving more than one tuning speed.

5- The PID controller which is consist of three parameters proportional KP, integral KI, and derivative KD, is being tuned for two times the first with manual tuning trial and error, the second with using practical swarm optimization algorithm PSO, the code of the algorithm is being downloaded from MATLAB and updated for this study purpose.
6- By comparing the two tuning method for PID controller, the PID based PSO shows an efficient improvement in the performance of the controller with no overshoot, no steady state error, and faster rise time with 20 seconds earlier. Generally the PSO optimization method enhance the overall controller performance.

## 4- Future work

1- Updated the proposed model to fit another vehicle kinds like electric or hybrid electric vehicles.
2- Using another optimization algorithm for tuning the PID controller to compare the performances.
3- Updating the proposed model to discuss the longitudinal and lateral vehicle motion.

## References

[1] S. S. James, S. R. Anderson, and M. Da Lio, "Longitudinal Vehicle Dynamics: A Comparison of Physical and Data-Driven Models under Large-Scale Real-World Driving Conditions," *IEEE Access*, vol. 8, pp. 73714–73729, 2020, doi: 10.1109/ACCESS.2020.2988592.

[2] E. Onieva, J. Godoy, J. Villagrá, V. Milanés, and J. Pérez, "On-line learning of a fuzzy controller for a precise vehicle cruise control system," *Expert Syst. Appl.*, vol. 40, no. 4, pp. 1046–1053, 2013, doi: 10.1016/j.eswa.2012.08.036.

[3] M. Abe, *Vehicle Dynamics and Control*. 2015. doi: 10.1016/b978-0-08-100390-9.00001-4.

[4] *Downloaded from SAE International by Beihang University, Saturday, May 06, 2017*.

[5] M. Nagai and P. Raksincharoensak, *Vehicle Dynamics of Modern Passenger Cars*. 2019.

[6] A. Zulu, "Towards explicit PID control tuning using machine learning," *2017 IEEE AFRICON Sci. Technol. Innov. Africa, AFRICON 2017*, pp. 430–433, 2017, doi: 10.1109/AFRCON.2017.8095520.

[7] E. A. Yfantis and W. Culbreth, "A Data Driven PID Control System," *2020 10th Annu. Comput. Commun. Work. Conf. CCWC 2020*, pp. 580–585, 2020, doi: 10.1109/CCWC47524.2020.9031230.

[8] J. Yang, X. Liu, S. Liu, D. Chu, L. Lu, and C. Wu, "Longitudinal tracking control of vehicle platooning using DDPG-based PID," *2020 4th CAA Int. Conf. Veh. Control Intell. CVCI 2020*, no. Cvci, pp. 656–661, 2020, doi: 10.1109/CVCI51460.2020.9338516.

[9] M. Walch, K. Lange, M. Baumann, and M. Weber, *Autonomous driving*. 2015. doi: 10.1145/2799250.2799268.

[10] D. Xiao, L. Li, P. F. Li, and J. Su, "Modeling and co-simulation of stop and go cruise control system," *Adv. Mater. Res.*, vol. 945–949, pp. 1486–1492, 2014, doi: 10.4028/www.scientific.net/AMR.945-949.1486.

[11] RC Flange, "Internal Engines," in *AirPollution88*, 1988, pp. 226–289. [Online]. Available: http://resolver.caltech.edu/CaltechBOOK:1988.001

[12] C. Singh, L. Kumar, B. kumar Dewangan, Prakash kumar sen, and K. Shailendra Bohidar, "A Study on Vehicle Differential system," *Int. J. Sci. Res. Manag.*, vol. 2, no. 11, pp. 1680–1683, 2014, [Online]. Available: http://ijsrm.in/v2-i11/15 ijsrm.pdf

[13] Y. Kebbati *et al.*, "Optimized self-adaptive PID speed control for autonomous vehicles To cite this version : HAL Id : hal-03442081 Optimized self-adaptive PID speed control for autonomous vehicles," 2022.

[14] ˚ K. J. A. and R. M. Murray, *Feedback Systems*, Version v2. 2009.

[15] E. S. Rahayu, A. Ma'arif, and A. Çakan, "Particle Swarm Optimization (PSO)

Tuning of PID Control on DC Motor," *Int. J. Robot. Control Syst.*, vol. 2, no. 2, pp. 435–447, 2022, doi: 10.31763/ijrcs.v2i2.476.

[16] Z. Xiang, D. Ji, H. Zhang, H. Wu, and Y. Li, "A simple PID-based strategy for particle swarm optimization algorithm," *Inf. Sci. (Ny).*, vol. 502, pp. 558–574, 2019, doi: 10.1016/j.ins.2019.06.042.