Quarter Car Modeling - Active Suspension System

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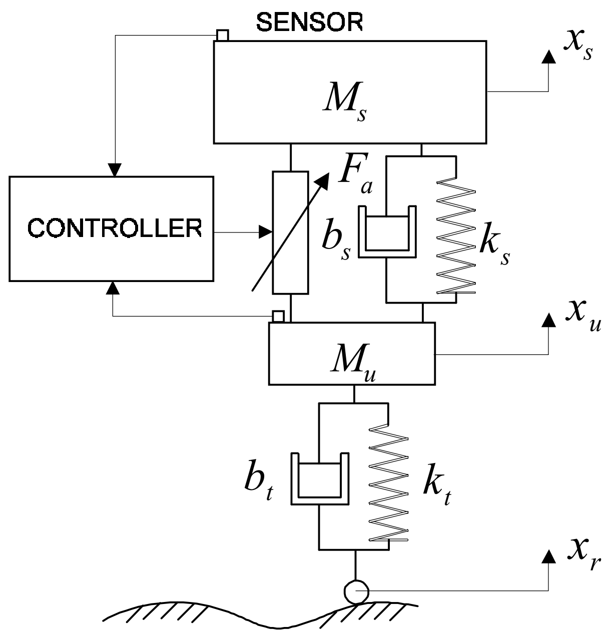
Problem:

"For road vehicle users, comfort is an important issue. To move from one place to another, road vehicles usually encounter various vibrations and shocks from ground, for instance, in traveling on a bumpy surface, or crossing over an obstacle. Prolonged exposures to vibrations cause some problems, such as pain and fatigue, for the passengers. To alleviate these problems, momentary loads from ground should be absorbed and damped out. Automotive suspension systems are intended to absorb and decrease the shocks and vibrations transferred from the ground to the passengers as well as the vehicle body. Passive suspension systems which consist of spring and damper components have been traditionally utilized on different types of vehicles, such as motorcycles, passenger cars, trucks and even bikes. Active suspension systems with separate actuators to apply controlled forces provide better ride A quarter-car model of a passenger sedan with active comfort and improved handling. The suspension system shown below represents the vehicle system at each wheel. It consists of a spring k_s, a damper b_s and a hydraulic actuator F_a. The tire stiffness and damping properties are also shown by k_t and b_t, respectively. The effective vehicle body mass is shown by M_s (sprung mass), and M_u (unsprung mass) represents the effective mass for the wheel and axle. The vertical displacements from the static equilibrium for M_u and M_s are shown by x_u and x_s, respectively. The road profile is represented by x_r. The suspension travel x_s - x_u is measured and compared to the set point (r = 0). The required actuator force is determined by the controller to eliminate the error, and thus, to reduce the vehicle oscillations. The actuator can provide a maximum force of 3000."



Parameter	Unit	Value
M_s	kg	300
M_u	kg	50
k_s	$\frac{N}{m}$	18000
k_t	$\frac{N}{m}$	180000
b_s	$\frac{Ns}{m}$	1200
b_t	$\frac{Ns}{m}$	0

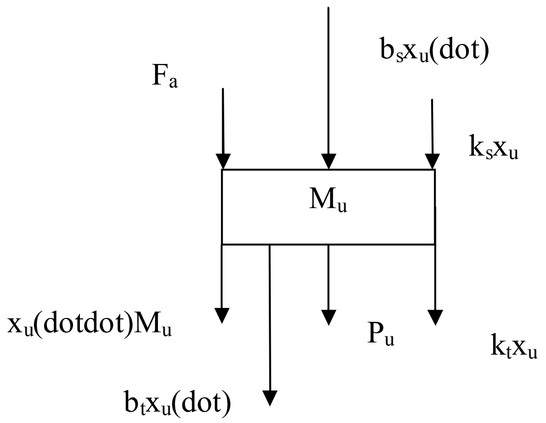
Suggested Solution:
Active suspension system:

- Spring k_s
- Damper b_s
- Hydraulic actuator F_a

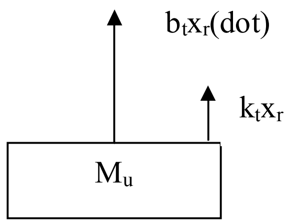
Tire properties:

- Stiffness k_t
- Damper b_t
- Effective vehicle body mass M_s
- Effective mass for wheel and axle M_u

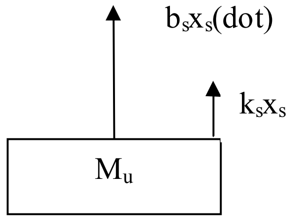
Perform force analysis:
Forces apply to M_u due to M_u's movement only:



Forces apply to M_u due to x_r only:



Forces apply to M_u due to x_s only:



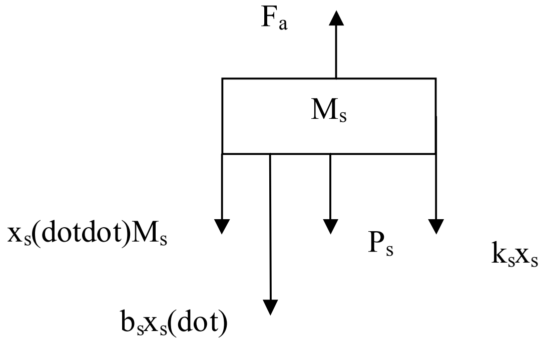
Equation for M_u:

$$F_a + b_s(\dot{x_u} - \dot{x_s}) + k_s(x_u - x_s) + b_t(\dot{x_u} - \dot{x_r}) + P_u + k_t(x_u - x_r) = -M_u\ddot{x_u}$$

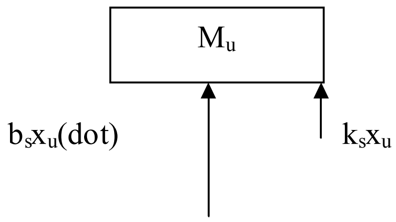
Hence:

$$\ddot{x_u} = \frac{F_a + b_s(\dot{x_u} - \dot{x_s}) + k_s(x_u - x_s) + b_t(\dot{x_u} - \dot{x_r}) + P_u + k_t(x_u - x_r)}{-M_u}$$

Forces apply to M_s due to M_s's movement only:



Forces apply to M_s due to x_u only:



Equation for M_s:

$$F_a + b_s(\dot{x_u} - \dot{x_s}) + k_s(x_u - x_s) - P_s = M_s\ddot{x_s}$$

Hence:

$$\ddot{x_s} = \frac{F_a + b_s(\dot{x_u} - \dot{x_s}) + k_s(x_u - x_s) - P_s}{M_s}$$

PID Controller for F_a:

$$P + I\frac{1}{s} + D\frac{N}{1 + N\frac{1}{s}}$$

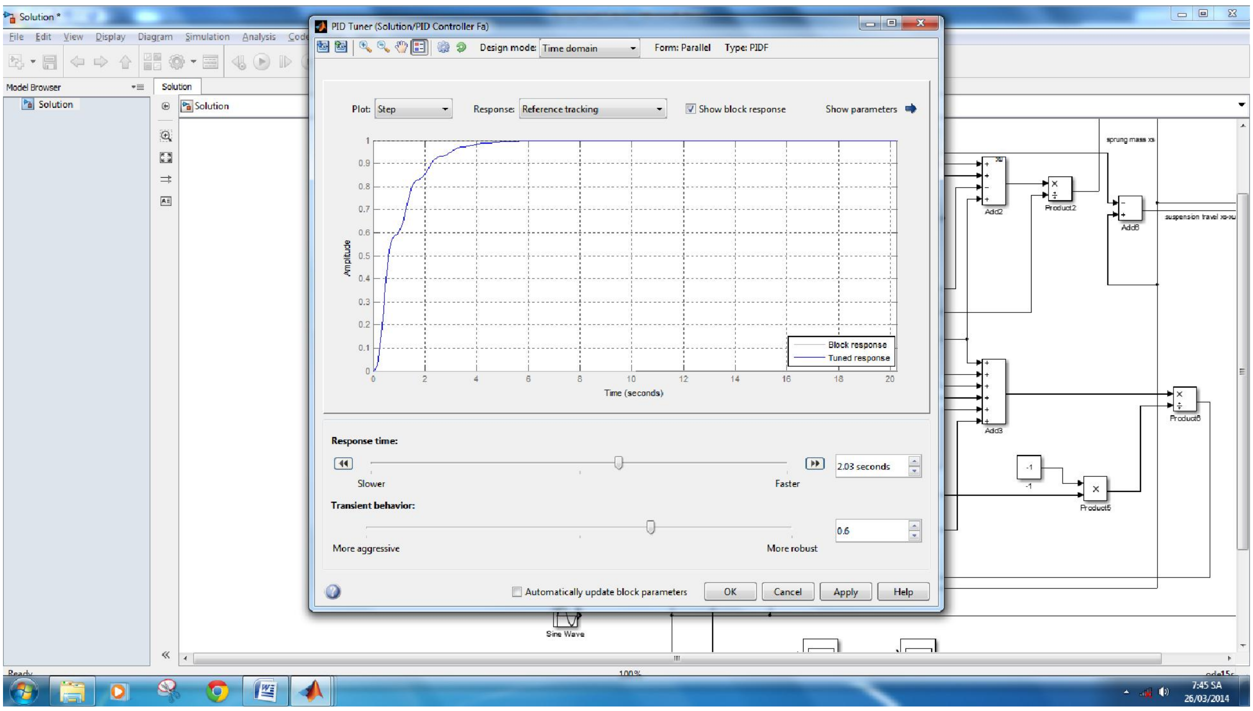
Initial condition:

- P = 1
- I = 1
- D = 0
- Filter Coefficient (N) = 100
- Upper saturation limit: 3000 (N)
- Lower saturation limit: 0 (N)

Simulate system on various road profiles, plot suspension travel (x_s - x_u), sprung mass (x_s) displacement and acceleration (x_s dot dot), tire deflection (x_r).

PID Tuning: tuning PID parameters to achieve desired characteristics:

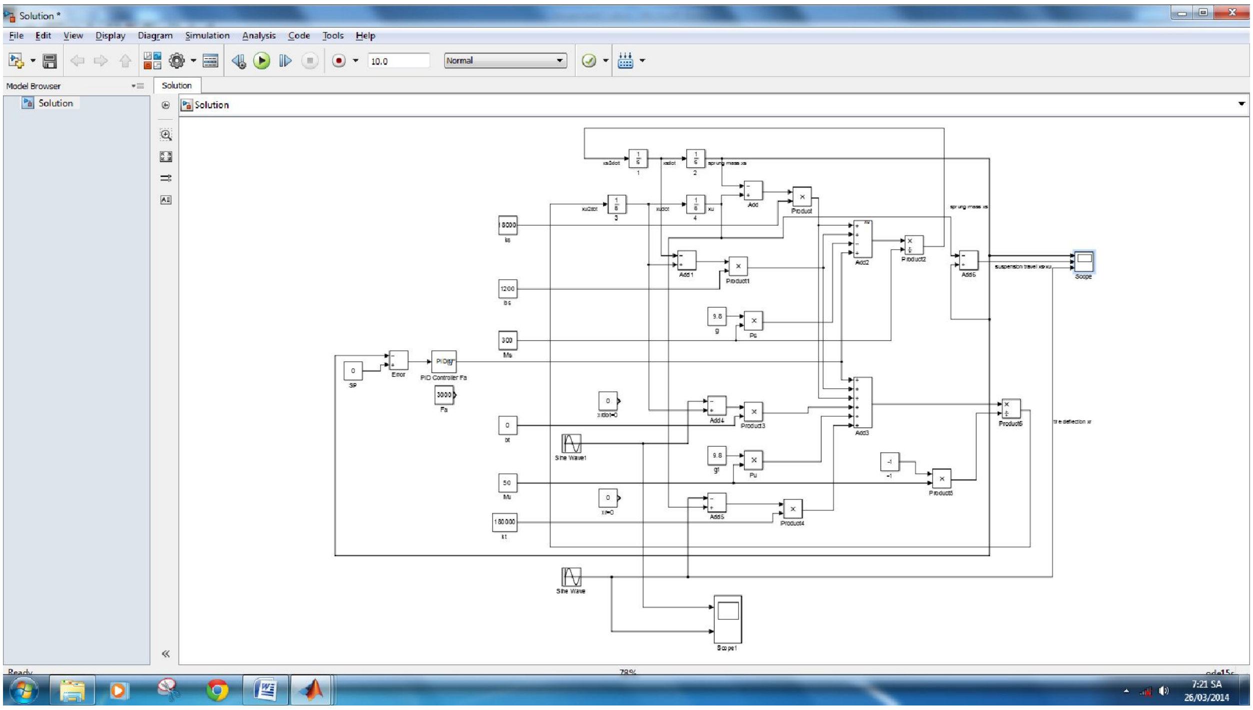
- Response time: 2.03s ~ 2s
- Robustness: 0.6



Assumption:

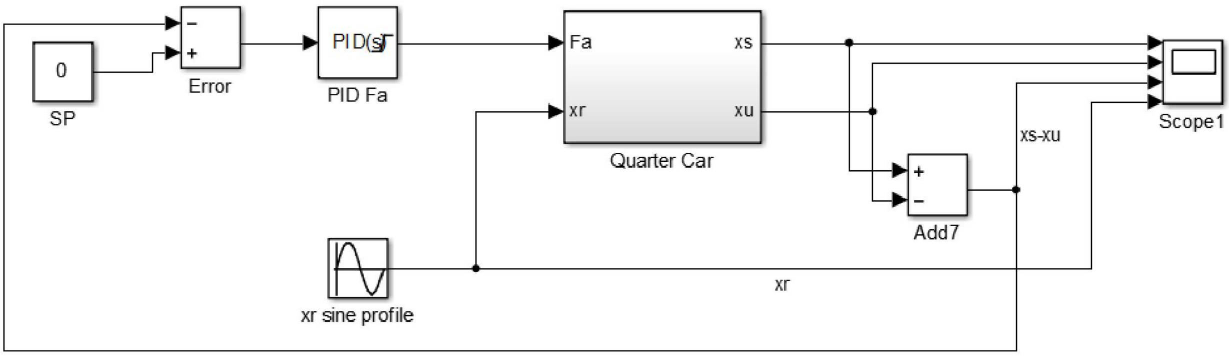
- Tire'shape is not affected by system mass
- PID controller has the following parameters: P = 0; I = 17490; D = 0; N = 100 (filter coefficient)
- When PID controller is not activated, a constant F_a = 1500 N is applied

Model

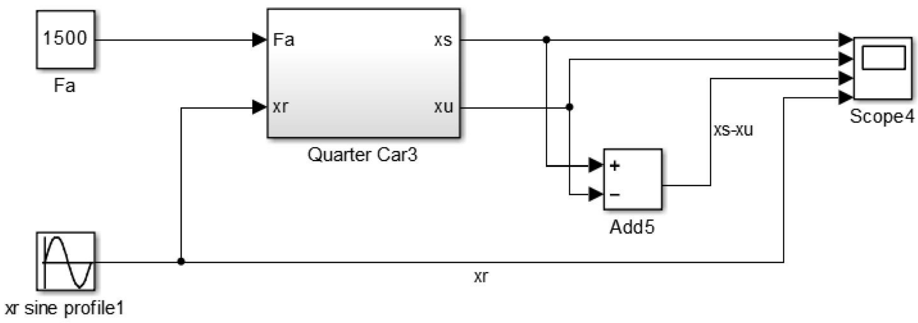


Simplified model using block:

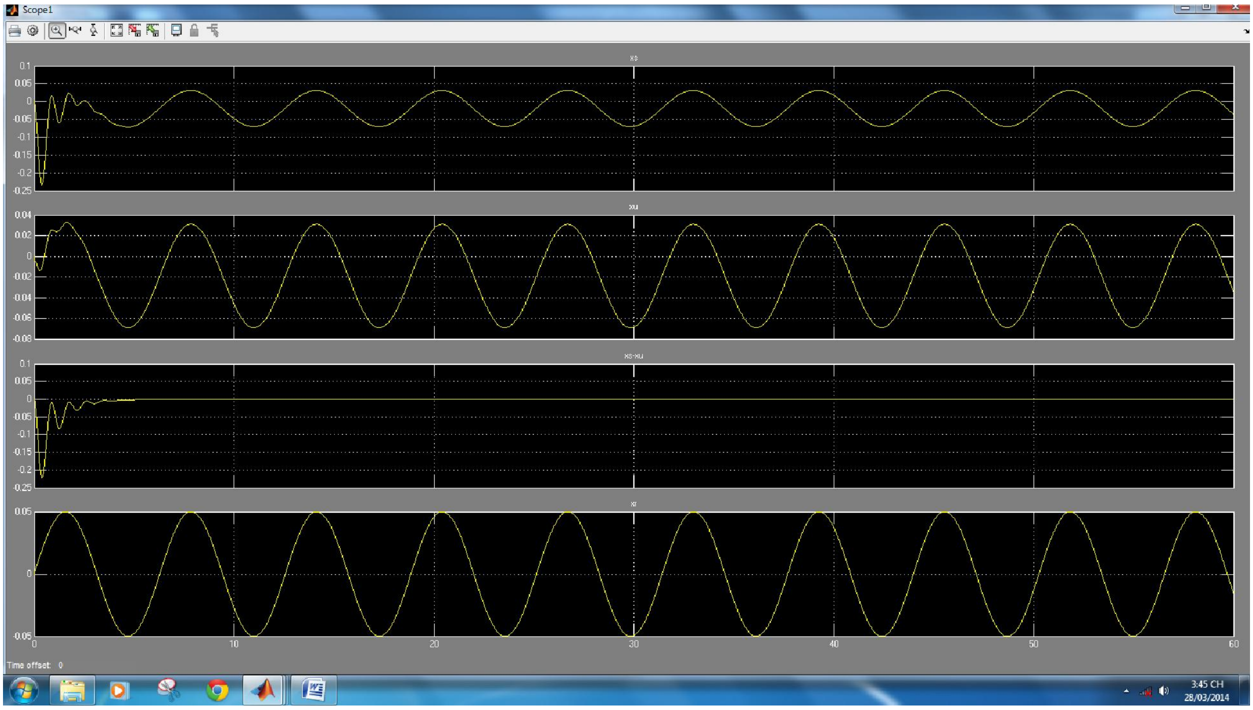
PID with feedback:



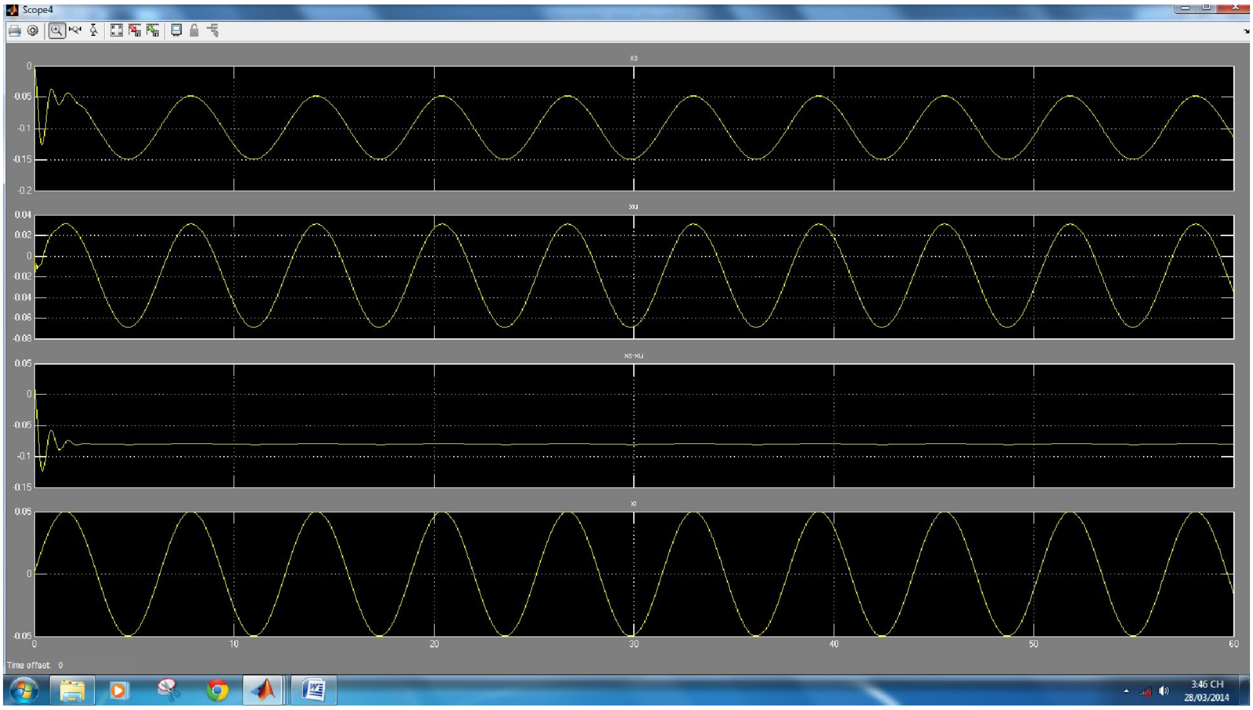
Constant F_a:



Sinusoidal bumpy with amplitude = 5 cm (max), frequency = 1 rad/s

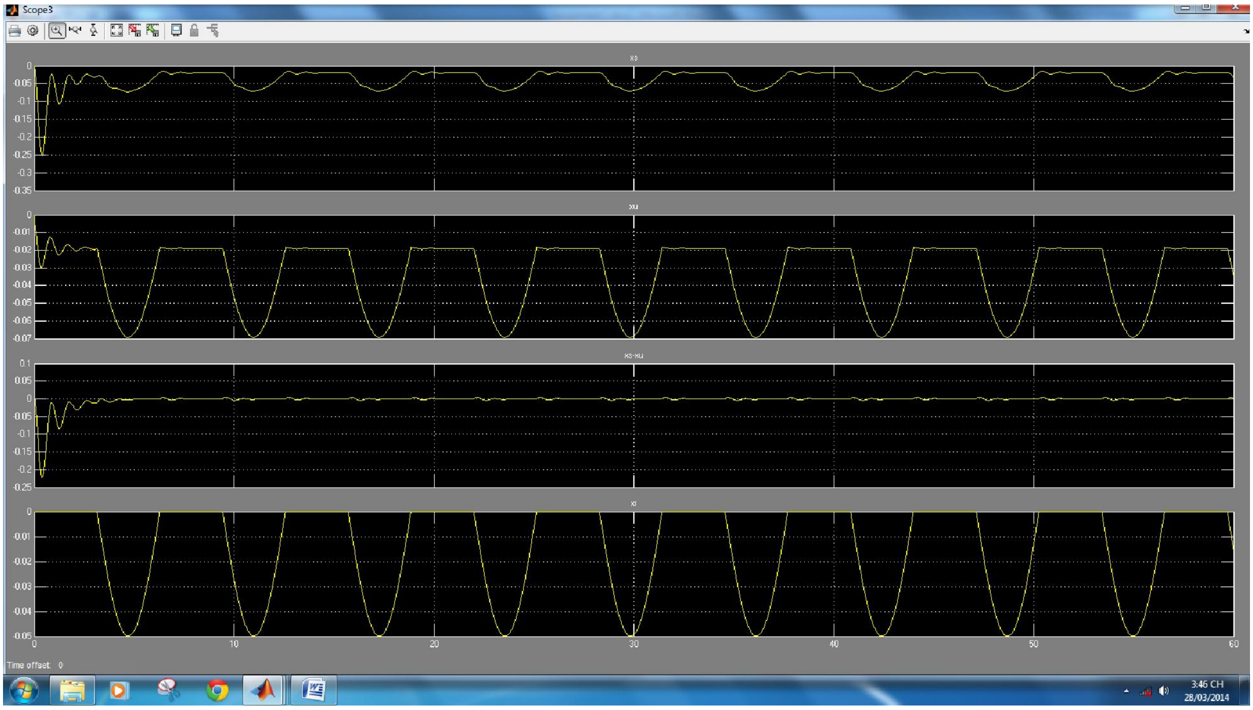


Simulation result with PID controller

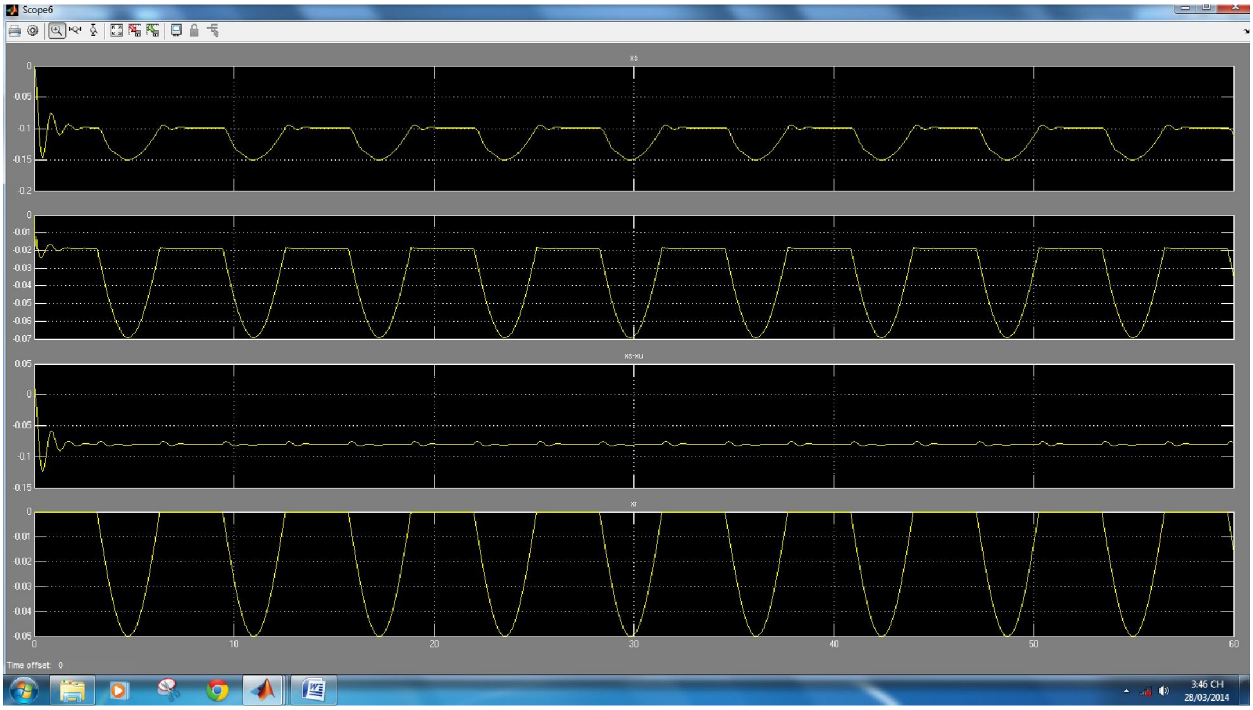

Simulation result without PID controller

Without PID controller, before settling, displacement of sprung mass is equal to about half of that when PID controller is applied. Similarly, suspension travel when not applying PID controller is smaller than when applying PID controller. It takes less time to achieve stable state when not using PID controller than when using PID controller. Tire defections are similar for both cases. For this case, overall without PID controller, the system has better performance (less vibration).

Pot-hole road profile



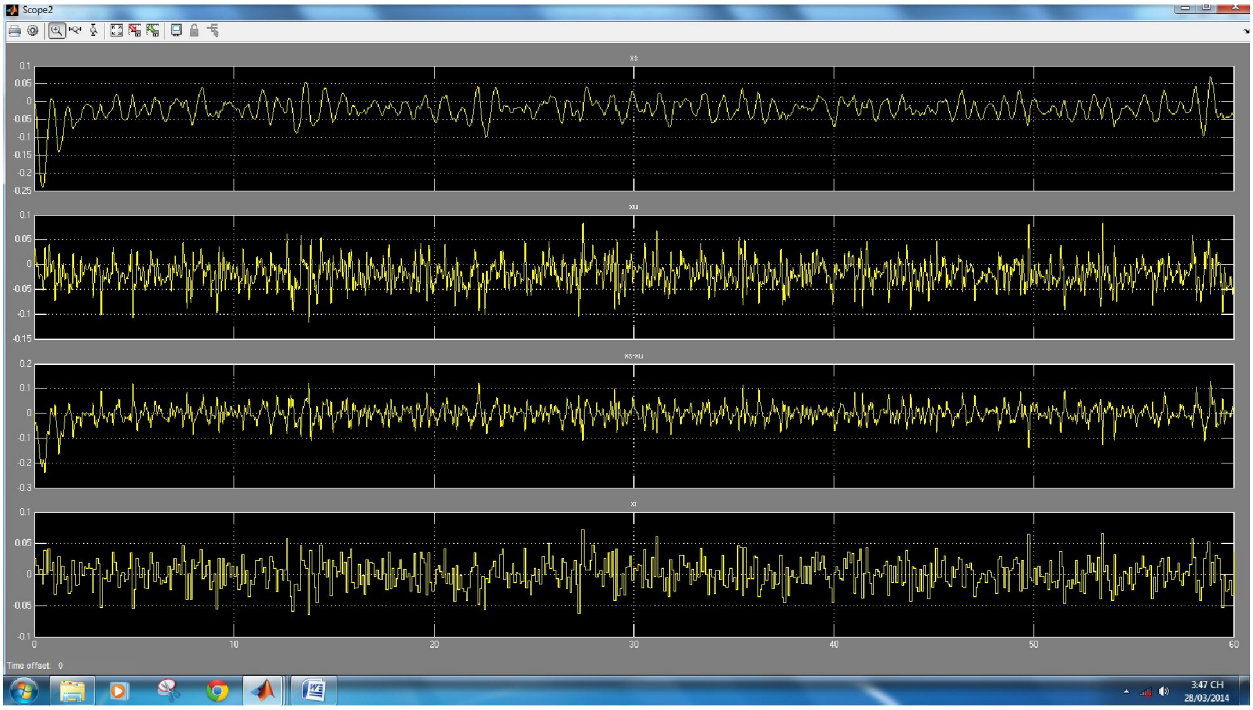
Simulation result with PID controller



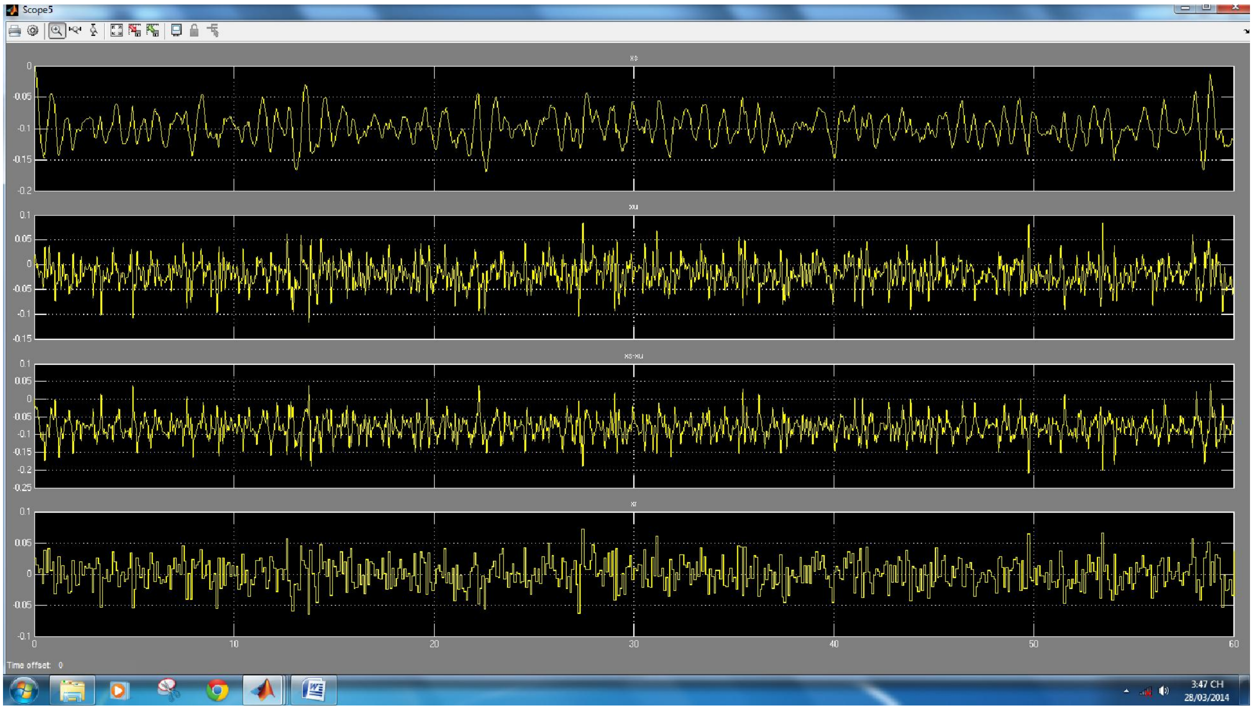
Simulation result without PID controller

Sprung mass displacements are similar for both cases. However suspension travel is smaller when using PID controller. Both suspension travel and sprung mass displacement are closer to 0 when using PID controller. Overall PID controller has better performance.

Random road profile: mean = 0; variance = 0.0005; sample time = 0.1s



Simulation result with PID controller



Simulation result without PID controller

Through observation, simulation results for both cases are similar except there are small offset between results of the simulations. Performance of both systems is similar.

Conclusion:

- Sinusoidal bumpy road: Without PID controller, performance is better.
- Pot-hole road: With PID controller, performance is better.
- Random road: Both are similar.

Suggestion:
Different controller shall be used or the PID parameters shall be further fine tune to obtain the best performance.