Simulation and Control of a Hydro-pneumatic Suspension system

Ibrahim A. Badway, Mohamed Ib. Sokar, Saber Abd Raboo

Abstract— this paper involves both analytical modelling, simulation and experimental study of a hydro-pneumatic suspension system. The motivating behind this work was to build a fully controlled and stable suspension system to increase ride comfort and to obtain excellent performance in rough terrains. The active hydro-pneumatic suspension, composed of a proportional directional valve, an accumulator attached to an actuator for the purpose of absorbing shock pulsation in addition to controlled system.

A PI-controller operates the valve to achieve the desired suspension performance. A mathematical model is derived and a simulation program was carried out using Matlab/Simulink software. To validate the simulation results, an experimental test rig of a quarter car suspension model is designed and examined. All the input signal and measured values are monitored and controlled using the Labview software that used to perform online simulation for the system. The program takes account of fluid mechanics, friction and nonlinearities of various sub-elements. The system behavior and oscillation rates are studied and tested for different kinds of terrains mathematically and experimentally. The comparison of results obtained from the simulation model and experimental test-rig shows a better conformity for a different excitation road profile. The maximum displacement resulting from the simulation and experimental results of the hydro-pneumatic system are within ± 12 % of the reference value.

Index Terms— Hydropneumatic Suspension, modelling, Control, Simulation, PI-controller, Labview.

1 Introduction

The development of automobile suspension systems has been evolved due to benefit from simulation tools and co-evolution of electronics and microprocessor control to get the most efficient and appropriate suspension system for cars and vehicles.

Suspension systems are generally classified into passive, semi-active, or active. This classification depending on the amount of external power required for the system to perform its function [1]. A traditional passive system comprised of springs and dampers. Semi-active suspensions use spring and controlled damper, the damper consumes small amounts of energy to adjust the damping level. Active suspensions have a controllable force actuator powered from an external energy source. Active suspension systems can potentially generate any desired force and so are much more flexible than conventional passive systems.

This paper starts with concise survey and discussion of related topics of automobile suspension designs and controls. A mathematical modelling, dynamics and Simulink model simulation of the active hydro-pneumatic suspension are introduced. The design and characterization of the computer controlled suspension test-rig, road profile and excitations are explained. The results for two states of car suspension are taken for the stationary car position and a passenger ride in the car, which consider an input.

Many researchers have investigated semi-active and active suspension by applying varying control techniques. A comparison of passive and active hydro-pneumatic suspension by using Matlab/Simulink was done by [2]. The results indicate that acceleration of the sprung mass decreased by 79.5% compared to passive suspension. Virtual car model analysis is used to compare the performance of active and semi-active systems such as work done in [3]. Others designed experiment, test rig and implemented Neuro active force controller [4 and 5], fuzzy logic controller [6, 7 and 5], or H_∞ controller method [8] to study active suspension.

A combination of a new hierarchical control strategy using a combined control scheme of a genetic algorithm, based self-tuning PID controller and a fuzzy logic controller was proposed by [9] on a virtual active hydro-pneumatic suspension prototype. The results of the designed prototype improved riding comfort characteristics of vehicles. Reference [10] Applied linear black box models, then identified it by using frequency domain identification techniques.

2 Hydro-pneumatic Suspension Analysis

2.1 Hydro-pneumatic System Description

The hydraulic circuit shown in Fig. 1, includes the power unit, on the right, that generates flow energy from the pump to the excitation circuit which used to simulate road ad even instant bumps. The upper part of this circuit that include 4/3 proportional valve which is used to control the level of the car via a

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ISSN 2229-5518

controller and a set of sensors. The suspension system consists of hydraulic actuator, orifice and hydro-pneumatic accumulator. The control program is done by Labview software on a laptop computer.

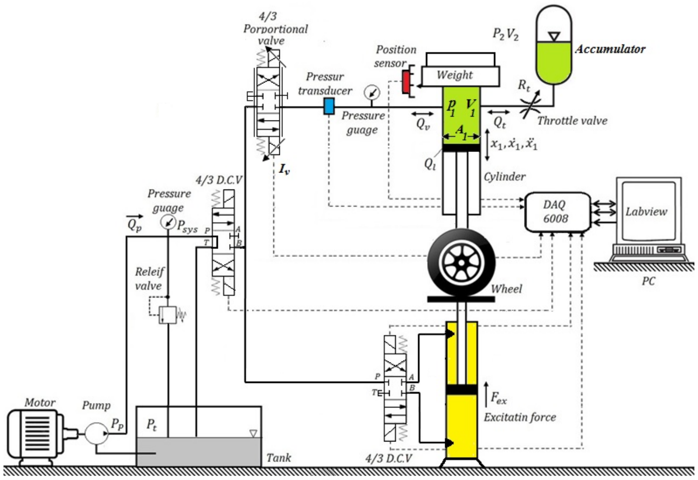


Fig. 1. General setup of the hydro-pneumatic suspension system

2.2 Mathematical Modeling

The mathematical modeling of an actual system is very important in engineering fields as it facilities the understanding of the performance and operation of a system. It also allows the determination of certain characteristics of the system, and can give important information on operating conditions with the use of relatively simple and inexpensive procedures.

The mathematical modeling equation is derived from the main components of the hydro-pneumatic transmission system to build the Simulink model. The position of the sprung mass or the piston, x_1 will consider equal to zero at the reference position and positive if it is "above" that position, negative when it is "below" it.

The rate of cylinder pressure is related to the proportional valve flow rate Q_v, flow to the actuator which equal to the product of Piston area A_1 and suspension velocity $\dot{x}_1$.

$$\dot{P}_1 = \frac{\beta}{V_1}\left(Q_v - Q_t - A_1\dot{x}_1\right) \quad (1)$$

The acceleration of the sprung mass, m can be expressed as a function of the excitation force F_{ex}, piston pressure P_1, and viscus friction F_{fr};

$$\ddot{x}_1 = \frac{1}{m}\left(F_{ex} - mg - F_{fr}\,\dot{x}_1 + P_1A_1\right) \quad (2)$$

The supply pressure P_s, for the proportional directional valve is sufficiently high and steady. If the pressure across the valve is stable, then a simplified first order relationship between valve flow Q_v and control current can be used, as follows

The flow throughout the throttle valve

$$Q_t = \frac{\left(P_1 - P_2\right)}{R_t} = A_1\dot{x}_1 \quad (3)$$

The control valve flow Q_v, valve current i_v, and valve pressure drop are related by the following simplified equation:

$$Q_v = \begin{cases} C_q\, w\, sign\,(i_v)\sqrt{\dfrac{P_s - P_1}{\rho}} & i_v \ge 0 \\ C_q\, w\, sign\,(i_v)\sqrt{\dfrac{P_1 - P_T}{\rho}} & i_v < 0 \end{cases} \quad (4)$$

The hydraulic accumulator is considered a gas spring in the suspension. It is pre-charged at $P_o = 7bar$ at the nominal volume V_o. When the cylinder is compressed, the flow of oil directs to accumulator through the throttle valve, raising the gas pressure of the accumulator from P_2 to P_3. This produces a stiffness force on the cylinder. Assuming adiabatic state change of the Nitrogen in the gas side of the accumulator.

$$P_3 = \frac{P_o V_o^k}{\left(V_o - A_1x_1\right)^k} \quad (5)$$

The gas stiffness can be calculated by differentiating the gas force with respect to cylinder displacement as follows;

$$K_s = \frac{dF}{dx} = \left[\frac{P_o V_o^k}{\left(V_o - A_1x_1\right)^k}\right]A_1^2 \quad (6)$$

2.3 Simulation Model

According to the derived mathematical model, the MATLAB/Simulink simulation model relating the system variables was built as shown in Fig. 2, the model includes the friction subsystem.

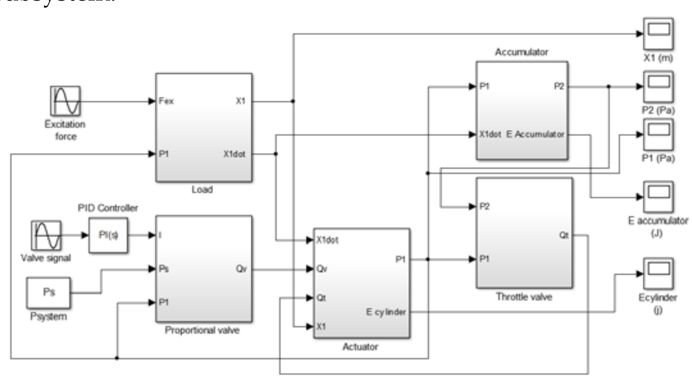


Fig. 2. The Simulink model of the quarter car model's suspension

A multi-domain physical systems (i.e. Mechanical, hydraulic, and electrical components) was built by Simscape simulation model in Simulink® as shown in Fig.3. A Simscape model assure the robustness of designed a Simulink model. It is much easier rather than complicated mathematics in Simulink. The results of Simulink and Simscape were found similar.

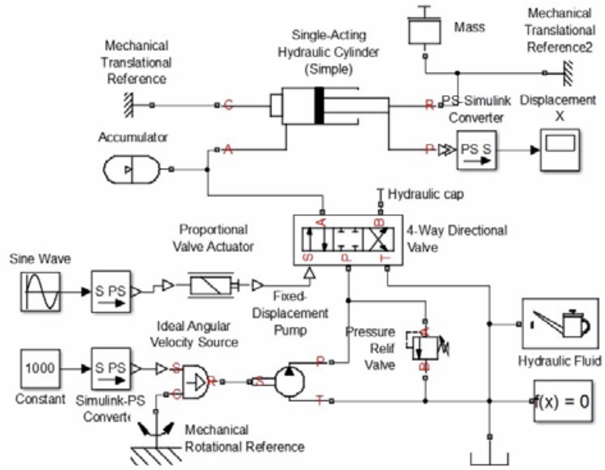


Fig. 3. The Simscape simulation model

2.4 Control Strategy;

The control strategy comprises of an adaptive PI - controller to keep the sprung mass of car body close to the predefined reference point and to minimize acceleration of the body. In the control loop the vertical displacement of the sprung mass is considered as the control target. The vertical displacement is measured by Linear Position Transducer (LPT), this value will be compared to the reference vertical displacement. The difference between the reference and actual deflection is manipulated by the controller to actuate the proportional valve. Then a command signal is developed for the suspension actuator. The in and out flow from the valve enable the hydraulic actuator and accumulator to overcome the excitation road force. The effectiveness of the controller is examined when the control system is subjected to road bumps, pothole and rough road conditions.

3 Experimental Test Rig

3.1 General Description

The test-rig of the hydro-pneumatic suspension is a good example of Mechatronics system. It includes mechanical, electrical, hydraulic, microcontroller, sensors, DAQ and Labview software will be explained.

3.2 Test-rig Components and Operation

The detailed construction and components of the test rig are shown in Fig. 4. The control hardware of the suspension system includes main part; besides sensors, filters, amplifiers, ready-made commercially integrated circuits (microcontrollers), D/A, and A/D converters. The hydraulic circuit road exciting that simulates road roughness has a separate control unit to provide the actuator with external disturbance, according to the simulated conditions of road during car movement (e.g. Sinusoidal, random, step or impulse signal). It consists of a hydraulic cylinder, a directional control valve, a hydraulic power source.

(1) Hydraulic tank.
(2) Pressure relief valve.
(3) Pressure gauge.
(4) Control valve for road cylinder.
(5) Control valve for hydraulic system.
(6) Control valve for proportional valve
(7) Weight at right.
(8) Weight at left.
(9) Pressure gauge.
(10) Accumulator.
(11) Wheel.
(12) Damper.
(13) Proportional control valve.
(14) Pressure sensor.
(15) Position sensor.
(16) Mechanical roller.
(17) Double acting cylinder.
(18) Throttle valve.
(19) AC motor for hydraulic system.
(20) AC motor for wheel.
(21) Control unit 1.
(22) Control unit 2.
(23) Laptop computer.

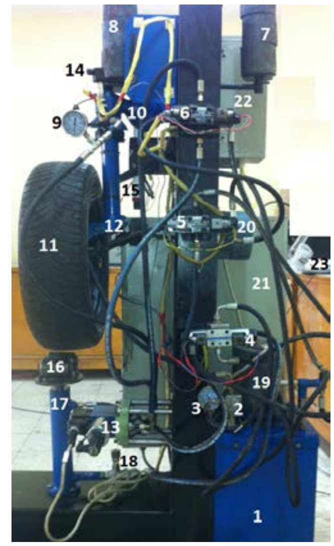


Fig. 4. General setup of the hydro-pneumatic suspension system

The hydro-pneumatic suspension unit consists of linear actuator, accumulator, and proportional valve, DAQ cards and controller. The main parameters of the quarter car system used in test rig are listed in Table 1. In the test rig, there is a place to add additional weights on the suspension system in step of 10 kg. The electric current signal supplied to the proportional valve is originated from the Ni-DAQ analog channel. The control signal coming from DAQ card is an analog output in volt ranged from (0 to 5 Vdc); but the control signal of proportional valve is in current. Thus, an interface electronic card, called signal insulated transmitter shown in Fig. 5, is required to convert from 0 - 5 Vdc to 4 – 20 mA.

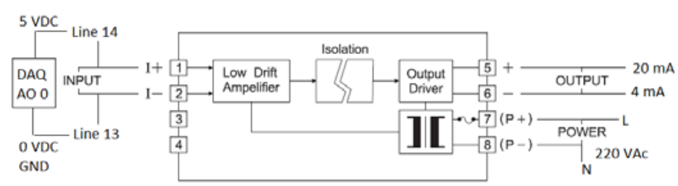


Fig. 5. Signal isolated transmitter and wiring connection

The cylinder pressure is measured by Precision Pressure Transmitter. Since, the output signal reading is in current from 4 to 20 mA. A resistance with a constant value (240 Ohm) is connected to the pressure transducer to convert the output current to volt to be suitable for the DAQ card. A Linear Position Transducer (LPT) supplied with 12 Vdc is used to measure the vertical displacement of the suspension, this signal is converted into a digital signal by A/D converted and sent to the DAQ card to control the suspension deflection. All measured signals are filtered and processed by the control unit finally transferred to the Labview program and stored in a separate file for each run.

To start operation, the Labview interface program must be first activated. Then the PI-controller will adjust the car displacement to the reference point. After steady state conditions are satisfied, different weights are added on the suspension system to check the ability of the control system to reject this disturbance. In two cases, the first case when the car parking and a passenger ride it. The second case, during car movement at different road bumps.

TABLE 1
THE MAIN PARAMETERS OF THE SYSTEM

Parameter	Value	Unit
Sprung body mass	250	kg
Usprung wheel mass	60	kg
Diameter of actuator (absorber)	4.5	cm
Length of acuator	40	cm
Accumulator size	0.4	Litre
Precharge pressure	7	Bar
Maximum pump pressure	150	Bar
Pump size	14	cm^3
Hydraulic oil density	790	Kg/m^3
Bulk modulus	2×10^9	N/m^2

3.3 Interface card DAQ and Labview

The signal acquired by the pressure and displacement sensors is converted by A/D converter and fed to the main controller unit. A Data acquisition card (6008) is connected to the Laptop via USB port. The Labview program performs online simulation by sending and receiving data to the controlled system. A picture of the virtual system builds in Labview is shown in Fig. 6

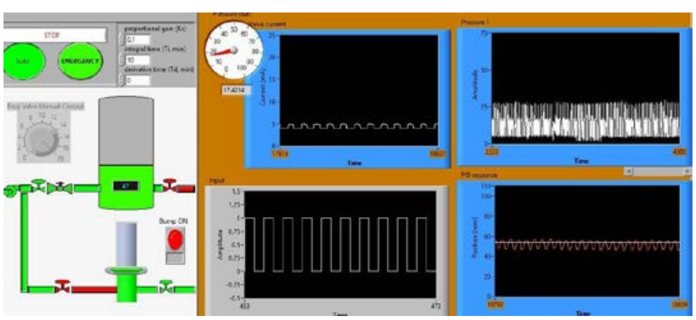


Fig 6. General setup of the hydro-pneumatic suspension system

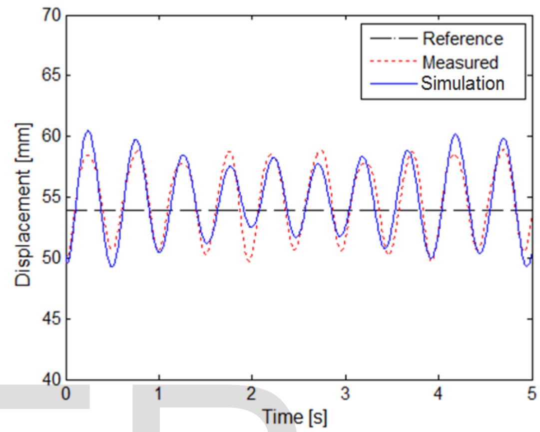
4 EXPERIMENTAL AND SIMULATION RESULTS

Since, the international conventions and standards for road classes are classified as A (very good), B (good), C (average), D (poor) and E (very poor) in. This study was carried on the class D road which is suitable for the most Egyptian roads.

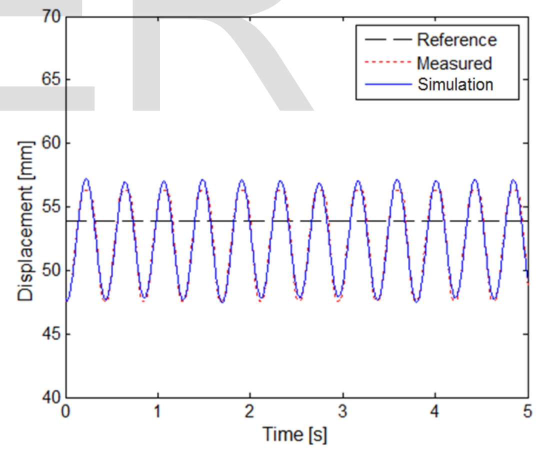
After the control system is designed, the controller performance is examined in time domain with random and square road displacement input. At each run, the responses of cylinder pressure, proportional valve current and suspension deflection are measured to evaluate the controller performance.

4.1 Model Validation

To check the effectiveness of the simulation model, the experimental results are performed by adding two additional weights on the quarter car model, namely 60 and 80 kg, and compared to the simulation results during as shown in Fig. 7.



(a) At +60 kg load



(b) At +80 kg load

Fig. 7. General setup of the hydro-pneumatic suspension system

4.2 Performance Results

The results shown here are taken in two conditions. He first condition, when the car is in standby mode (i.e. Engine idle without car movement) and passengers riding the car, these added weights are distributed along the car suspension system, Assuming the share of the weight on the quarter car model ranges from 60 to 80 kg. In this case the controller reacts to the added weight on the sprung mass and reset the suspension deflection again to its reference position of static equilibrium as shown in Fig. 8. This cannot be controlled in a passive system, whereas a fixed suspension deflection appears that proportional to the added weight on the car body.

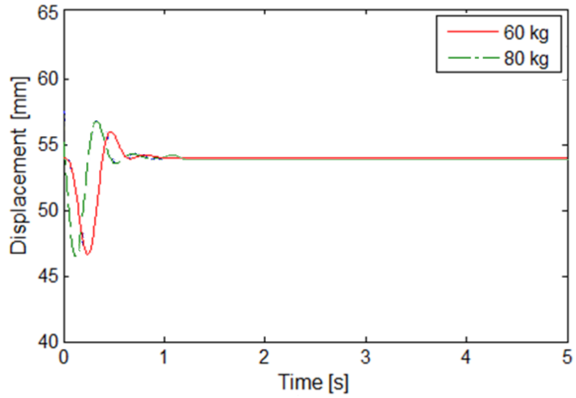
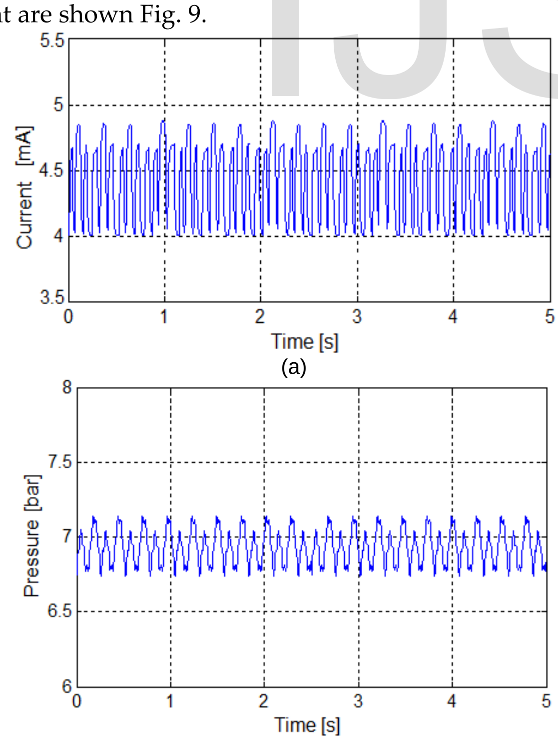


Fig. 8. Disturbance rejection of car body when subjected to additional weight

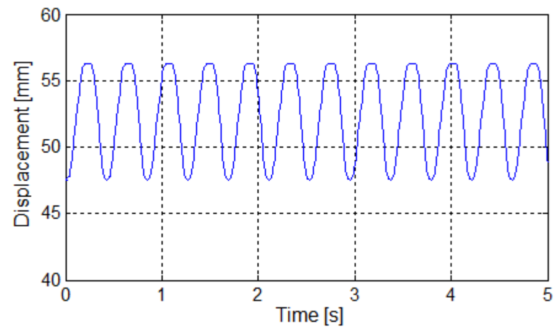
Contrary to the above stationary idle car condition, during car movement, the suspension system is subjected to different input excitation forces from the road roughness, even bumps or potholes. The system performance is studied by running the experimental test-rig with an additional passenger load, in this case only one weight is enough to use, namely 80 kg. The changes in valve current, cylinder pressure and suspension system displacement are measured via suitable sensors installed on these test-rig components.

The experimental results at the car movement under plus weight are shown Fig. 9.



(a)

(b)



(c)

Fig. 9. Experimental results of the controlled System at 60 kg (a) proportional valve current, (b) cylinder pressure, and (c) suspension displacement

Simulation results showed that the implemented controller of the hydro-pneumatic suspension under study is effective and reacts to the road excitation profile, and operate the system smoothly reduces the oscillation of the suspension system. Also, the controlled system is able to reject disturbance input to the hydro-pneumatic suspension system.

4.3 Analysis of the Results

The results obtained of hydro-pneumatic vehicle suspension at stationary and driving conditions are carried out by using Matlab/Simulink simulation model and experimental environment on D-level random roads.

Comparing the simulation results to the experimental one shown in Fig. 5 indicate a good conformity between them. This means that the simulation model is trusted in studying the performance of the system under study.

The result obtained in Fig. 6, display the possibilities of the control system to reject and disturbance load on the system by keeping the suspension system at a predefined reference value. Also, this curve shows that he controlled system has a superior damping effect on the excitation load force which provides better road holding capability.

The performance curves of the system shown in Fig. 7, shows the dependences of system variable on the input signal of the road profile. The amplitude and speed of the excitation force have a great effect on the system performance and suspension deflection. The maximum displacement resulting from the simulation and experimental models of the hydrodynamic system is within ± 12 % of the reference value.

5 Conclusion

The objective of this research was to study the performance and explore the potential of hydro-pneumatic suspension system to use in a vehicle driven on rough roads.

The mathematical model is derived from the basic equations of the system components. The simulation model is created by a Matlab/Simulink is built from the governing equation. The control strategy applied in the model comprises of an adaptive PI - controller. The main task of the controller is to keep the sprung mass of car body close to the predefined reference point and to minimize acceleration of the body under varying excitation force or load disturbance. A quarter car test rig model is constructed. The control hardware of the system includes sensors, filters, amplifiers, microcontrollers, D/A, and A/D converters.

The results of the Simulink simulation model are compared to the measured experimental results obtained from the test-rig for different weights. The comparisons indicate the closeness of the measured to the simulated results. This indicates the ability of the simulation model to study and analysis the performance of the hydro-pneumatic suspension system at various road excitation. The results show also that controlled system has a superior damping effect on the excitation load force which provides better road holding capability enabling rid comfort. The maximum displacement resulting from the simulation and experimental model of the hydro-pneumatic system are within ± 12 % of the reference value.

Acknowledgment

This study was supported by the faculty of engineering at Shubra, Benha University, Egypt and department of Mechatronics and Robotics Engineering, Egyptian Russian University in Cairo. The authors gratefully acknowledge these support universities and the anonymous reviewers who provided invaluable references and suggestions for this study.

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