# 527. An innovative design of actuation mechanism for active seat suspension of an off-road vehicle

**Noroozi Ghale Babakhani A.**[1]**, Ghobadian B.**[2]**, Salahshoor K.**[3]**, Mohtasebi S.**[4]
[1]Tarbiyat Modaress University,
**E-mail:** *noroozi@mapnaequipment.com*
[2]Tarbiyat Modaress University,
**E-mail:** *ghobadib@modares.ac.ir*
[3]Petroleum University of Technology,
**E-mail:** *Salahshoor@put.ac.ir*
[4]University of Tehran,
**E-mail:** *mohtaseb@ut.ac.ir*

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**Abstract.** In recent years human-machine interaction attracts scientific community attention because of human quality and health issues. Driver seat should be designed so that it would ensure occupational health as well as increase work efficiency. The aim of this research is to maintain seat height at constant level with regard to chassis excitation at different levels of frequency and amplitude by means of new design of pneumatic actuation circuit. Sinusoidal function was used for base vibration since almost all of excitation functions can be derived from it. System response shows in low frequency/high amplitude and high frequency/low amplitude chassis vibration, transmissibility decreased about 60% and 40% compared to solid suspension respectively.

**Keywords:** active seat suspension, off-road vehicle, pneumatic isolator, vibration.

## Nomenclature

| Notation | Description | Notation | Description |
|---|---|---|---|
| $k$ | Spring stiffness | $m$ | Mass |
| $\omega_n$ | Natural frequency | $\omega$ | Chassis excitation frequency |
| $y$ | Actual seat to chassis distance | $x$ | Actuator opened length |
| $L$ | Length of scissor link | $h$ | Chassis to ground distance |
| $y_{calc}$ | Desired seat to chassis distance | w | Accepted controller error |
| $\dot{x}_{calc}$ | Desired actuator velocity | $\dot{y}$ | Actual seat velocity |
| $\dot{h}$ | Actual chassis velocity | | |

## Introduction

Car suspension system has two tasks "Ride Comfort" and "Ground Traction" which influence each other adversely. As traction force tends to increase by stiffening suspension system, ride comfort decreases because of more ground unevenness transmissibility [3].

For most off-road soil working machines it's important to keep excavation depth constant, which is in conflict with suspension system. It is because the vibration of plowing blade results in non-uniform surface. Therefore there are two solutions to the problem: installation of vibration isolator between chassis and cabin or between cabin and seat. Since the first approach requires not only large spring and damper but also it imposes high costs, the second approach is preferred. Vibration isolator located between cabin and seat is cost-effective and occupies useless space [12].

According to standards ISO 2631-1 and VDI 2057-1, natural frequency of human body is about 1-80 Hz. There is an overlap between the mentioned range and the vibration frequency of vehicles while riding on uneven roads, which causes resonance. Side effects of this phenomenon are spine and muscle-skeletal disorder, vascular symptom and appetite decrement. So because of ordinary seat inefficient performance and threat to operator of an off-road machine, active suspension should be tested and adopted to substitute traditional one [1, 7].

Issue of active damping of harmful vehicle vibration appeared five decades ago. The main purpose of vibration dampers is not removal of vibration. It should serve as a bandwidth filter, i.e. it should remove harmful frequencies which cause resonance [12].

As it can be observed from fig.1, as ratio $\frac{\omega}{\omega_n}$ approaches to 1, the transmissibility value tends to increase.

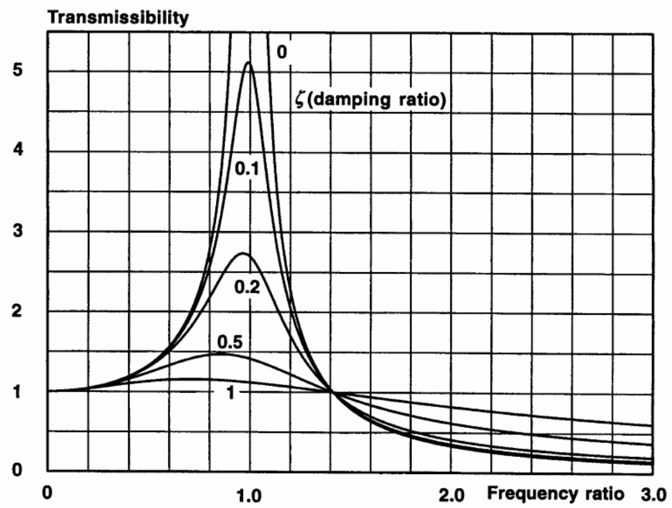



**Fig. 1.** Graph of chassis transmissibility as a function of frequency ratio ($\frac{\omega}{\omega_n}$) [6, 11]

Increasing damping ratio at resonance vicinity regions will reduce transmissibility but this situation causes increased transmissibility at higher vibration frequency. To overcome resonance, we can also diminish natural frequency ($\omega_n$). So frequency ratio and transmissibility will be increased and decreased respectively. Natural frequency is related to spring constant *K* and mass *m*. Since *m* is constant, the remaining variable which can affect natural frequency is spring constant *K*. Variation of this variable has some limitations. For example, to reach natural frequency about 1 Hz, the initial un-stretched length of spring would be about 0.5 m and lack of space will become a problem [10].

Using gas spring or gas accumulator enables designer to gain variable spring in suspension system. According to pressure of air trapped in accumulator, the equivalent spring stiffness will vary [2].

As mentioned earlier decreasing natural frequency in the range $\omega < \omega_n$ causes increase in transmissibility. To overcome this problem damping ratio should be increase to alleviate transmissibility but the deteriorative phenomena will occur at frequency in the region $\omega > \sqrt{2}\omega_n$ and transmissibility increases as indicated in fig.1. So damper should be active and passive in region $\omega < \sqrt{2}\omega_n$ and $\omega > \sqrt{2}\omega_n$ respectively [4].

Nieto et al present a variable orifice between gas accumulator and suspension system to control air flux flowing to gas spring. Amount of volumetric air which is transferred between gas spring and accumulator chambers determines stiffness of suspension. As orifice shifts to close position, amount of air transferred to secondary chamber decreases so pressure increases to higher points. This causes stiffening of the suspension system and vice versa [9].

The idea of hybrid suspension, which uses hydro-pneumatic interconnected system, previously used by industrial companies such as Mercedes S-series, Citroen and Toyota. The same research was done by Karnopp et al which is depicted in fig. 2 [2].

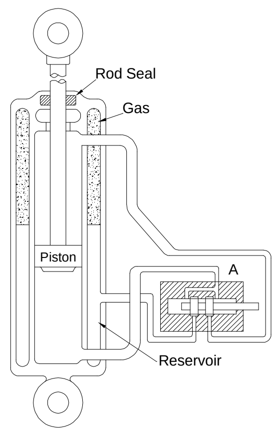



**Fig. 2.** Hydro-pneumatic interconnected suspension

Solenoid valve *A* controls flow and direction of exhaust oil from cylinder to two destinations. Variable orifice through valve *A*, directs oil to the other side of cylinder or to gas chamber or combination of them. Due to vibration intensity, performs mentioned functions [2].

Research of Liu et al indicates that semi-active dampers have better response than conventional ones. They used on-off damper in their research and compared system response at different frequencies on SDOF[1] model with sinusoidal base excitation. Since dual phase dampers have chatter at phase changing state, the system response is intermittent at mentioned

[1] Single degree of freedom

situation. Comparison of different dampers responses shows only in continuous damping the overshoot is minimized [8].

Almost all of mentioned industrial research works focused on controlling damping ratio regardless of stiffness. The aim of this research is not only to propose a cost-effective and simple actuation mechanism for seat suspension, but also more sophisticated in attenuating vibration at different frequencies and amplitudes. This is accomplished by controlling stiffness and damping ratio simultaneously.

## Pilot model description

Fig. 3 illustrates prototype model of active suspension system. The basic items of system are as follows:

- Induced vibration mechanism;
- Scissor mechanism;
- Pneumatic circuit;
- Control circuit.

## Induced vibration mechanism

This part of rig test is not only used as base frame for supporting other apparatus but also uses sliding-crank mechanism to vibrate chassis for simulating road irregularities. Electrical motor for turning crank mechanisms, equipped with variable gearbox hence the output crank revolution can be varied. 5 holes were drilled in crank to change connecting rod and slider position with regard to crank axis. So the vibrating slider can stimulate slider at different frequencies and amplitudes. Frequency range in experiment was varied between 0 to 3 Hz, while amplitude - between 0 to 10 cm as well.

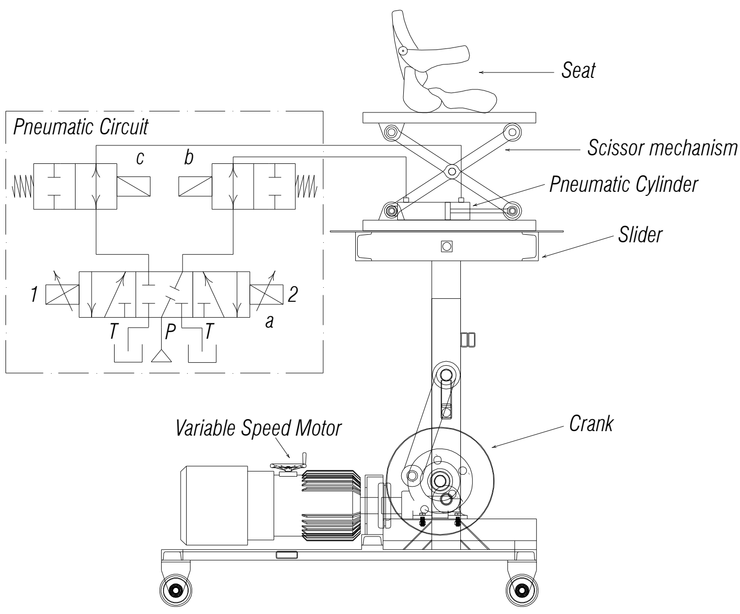
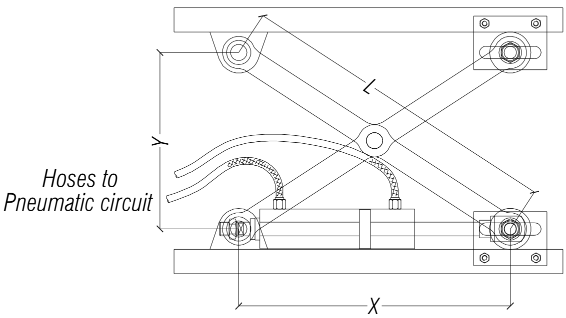


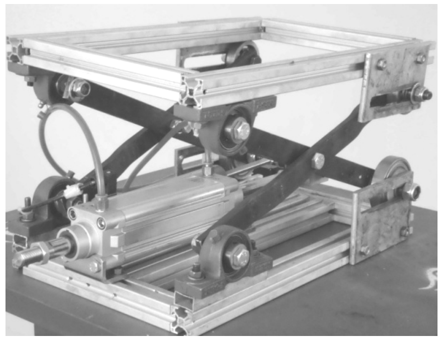

**Fig. 3.** Prototype model for pneumatic active suspension test

## Seat Scissor Mechanism

The purpose of using this mechanism is not only to control the vertical position of seat but also overcome the low speed of pneumatic cylinder. In other words, because of kinematical characteristics of this mechanism, the low speed of pneumatic cylinder will be compensated. Fig. 4 illustrates scissor mechanism in more detail.




**a)**



**b)**

**Fig. 4.** (a) schematics of scissor mechanism with dimensional designations (b) fabricated prototype

In scissor design two conditions should be satisfied. The required height and maximum velocity which scissor can reach. This was accomplished by means of simulation software and try and error methods.

As can be seen from fig. 4 the kinematic equation between pneumatic cylinder and scissor height is [5]:

$$\dot{y} = \frac{-x}{\sqrt{L^2 - x^2}}\dot{x} \tag{1}$$

## Pneumatic Circuit

An innovative design of pneumatic circuit must meet requirements such as overcoming inertial and frictional forces in the threshold movement of seat, and decelerating seat movement at the end of stroke. Fig. 3 provides schematic of pneumatic circuit. It consists of three major parts: solenoid a, b and pneumatic cylinder. Solenoids *b* and *c* are identical simple 2-2 valves but solenoid *a* is a 5-3 proportional valve. Port *P* is connected to compressor with 50 liter auxiliary tank and 7 bar pressure (Fig. 7) and port *T* is connected to free air. To equalize expansion and contraction speed of cylinder, piston with two connected shafts was used (Fig. 4a). Sizing of cylinder consists of stroke and diameter to satisfy height compensation as well as related velocity.

Operation of circuit consists of two stages. First stage is controlling cylinder velocity via proportional valve *a* and the second stage dedicated to deceleration of cylinder via valves *b* and *c* to avoid shock at the end of stroke (Fig. 3). Each valve in circuit is controlled by PLC which will be defined in the remaining of the article. PLC senses the velocity and direction of chassis movement and sends the related voltage to one of the ports of valve *a*. The stimulation voltage is proportional to speed of cylinder, i.e. the more voltage is applied to valve *a* ports, the more compressed air will pass through the valve and the more speed of cylinder we will expect. At the second stage of performance, as cylinder approaches to the end stroke, either valve *b* or *c* due to cylinder movement direction, is activated and closes air exhaust port. This causes instantaneous increase of system stiffness so cylinder is decelerated to stop at desired position. Implementation of this idea eliminates the need for accumulator. On the other hand, entrapped air in the enclosed chamber of cylinder will help cylinder to overcome the inertial and friction forces at the beginning of motion. Entrapped air will be warmed up but fortunately it will be discharged to ambient via port *T* of solenoid *a*. Mentioned remarks imply that solenoid *a* controls the damping ratio with variable rate but solenoids *b* and *c* control the stiffness.

## Extracting control model

Before programming the PLC, the control flow chart which defines the sequence of valve operation and actuating voltage should be extracted. For solenoid *a* we applied different voltage levels to port and measured the velocity of cylinder. In other words a look up table was constructed which specify each velocity to related voltage. Fig. 5 illustrates graph of applied voltage versus related velocity.

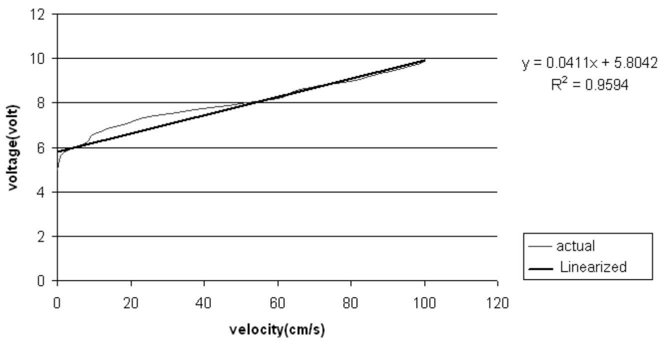



**Fig. 5.** Velocity vs voltage graph with recommended regression line

Best fit line at 95% acceptance level which is depicted in fig. 5, has a form of *voltage=F(velocity)* and will be used for PLC performance model.

For solenoids *b* and *c*, in each frequency and amplitude the best activation point is recommended via try and error. This point should satisfy desired deceleration rate and stop point. Finally a sophisticated model which predicts not only solenoid *a* action plan but also represents best phase changing from loose to coarse stiffness by solenoid *b* or *c*.

## Controller action flowchart

Major tasks of control system are to sense any chassis movement and actuate pneumatic system to compensate this non-homogeneous action. This can be achieved via controlling pneumatic cylinder position and velocity, which affect position and velocity of seat as well.

Two linear resistive sensors send position-related voltage of pneumatic cylinder and chassis instantaneously to the controller. Controller receives mentioned voltage after A/D conversion and then decides to open or close cylinder with regard to predefined model. A neural-network-shaped model was designed for controller. Flow chart scheme is presented in Fig. 6.

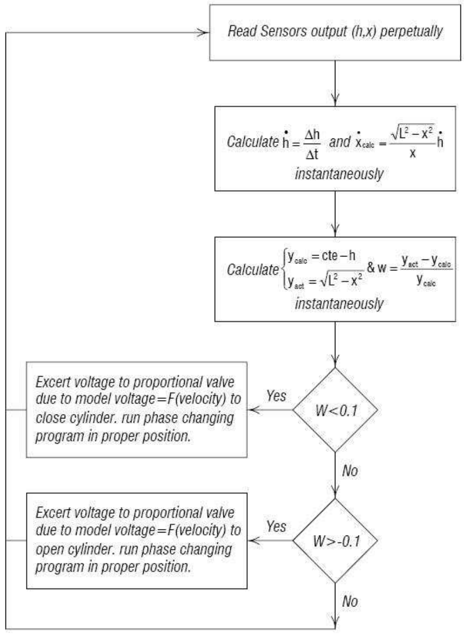



**Fig. 6.** Control flow chart scheme.

After processing actuation voltage by PLC due to mentioned model, the calculated voltage will be sent to solenoids after D/A conversion.

We are expecting both position and velocity control of seat. The flowchart in Fig. 6 indicates that both parameters are considered. If we nominate *h* for output value of chassis sensor, for ideal control, equation (2) should be satisfied:

$$y + h = cte \tag{2}$$

In other words, by altering *h* value, for road irregularities simulation, controller should change value of *y* to satisfy equation (2). By combining equations (1) and (2) we will have the appropriate velocity of pneumatic cylinder for control model:

$$\dot{x}_{calc} = \frac{\sqrt{L^2 - x^2}}{x}\dot{h} \tag{3}$$

Using experimental equation from fig. 5 the related voltage for solenoid activation will be extracted.

For reduction of valve sensitivity a working range was defined for the controller as per equation (4). This factor which is denoted as *w* is the touchstone for quality of controller performance. Values which lie in the range $-0.1 < w < 0.1$, cause idle operation of controller, but at remained range, the controller will act as indicated in flowchart in fig. 6:

$$w = \frac{y_{act} - y_{calc}}{y_{calc}} \tag{4}$$

## Experiments and analysis method

Since almost all of waves can be simulated by sinusoidal function, thus chassis induced vibration mechanism executes movement with 0-10 cm amplitude and 0-3 Hz frequency to chassis. The fabricated pilot model and its sub-systems are illustrated in Fig. 7. At mentioned frequencies and amplitudes we measured the response of the system by plotting growth rate of chassis and seat velocity versus each other. Differentiation from equation (2), represents that for optimal controller performance the velocity growth rate of seat should be equal to velocity growth rate of chassis but in reverse direction. So the best controller performance will be obtained if the response curve conforms with first-third quarter bisector.

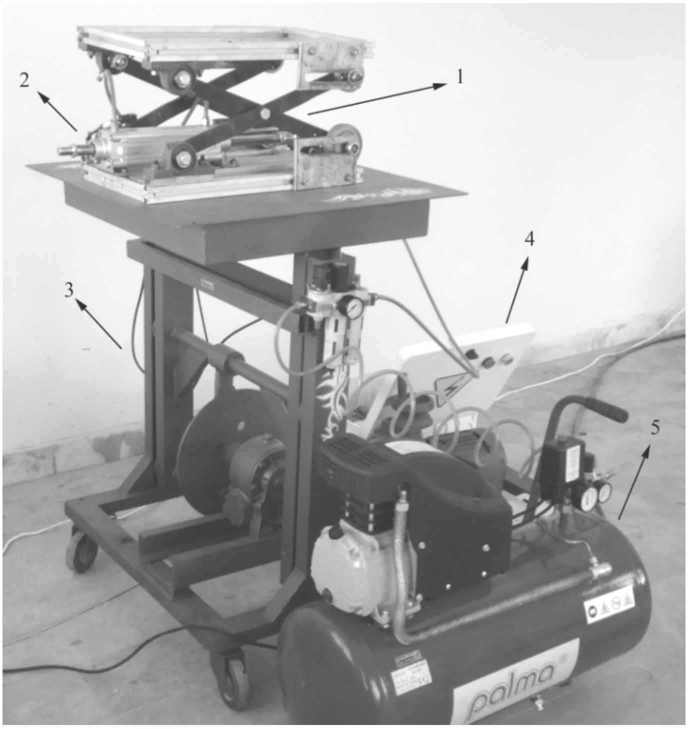



**Fig. 7.** Final assembled mechanism: (1) scissor mechanism, (2) pneumatic cylinder, (3) chassis with slider-crank mechanism, (4) electronic control circuit, (5) compressor with auxiliary tank

The performance analysis criterion of pneumatic control system is defined as deviation of response curve from first-third quarter bisector.

## Results and discussion

Results indicate that at low actuation voltage because of low passing of pressurized air through proportional valve, the inertial forces (friction and weight inertia) are high and we can observe some nonlinearity at low voltage region in fig. 5 graph.

To evaluate system response, we depict chassis position variation versus seat position variation (fig. 8). The more data is compatible with bisector, the better controller performance will be obtained. As can be observed in fig. 8 parts, the expansion/contraction cycles for each graph are not coincident with each other because of inertial forces.

Comparison of figs. 8a and 8b reveals that in steady frequency, with decreasing amplitude, the response reaction of system will be diminished. This can be interpreted as decreasing amplitude the required time for phase changing is much less than before. So the system reacts slower.

Comparison of figs. 8a and 8c reveals that in steady amplitude, with decreasing frequency, the response reaction of system will be increased. The same criterion can be concluded for present phenomena.

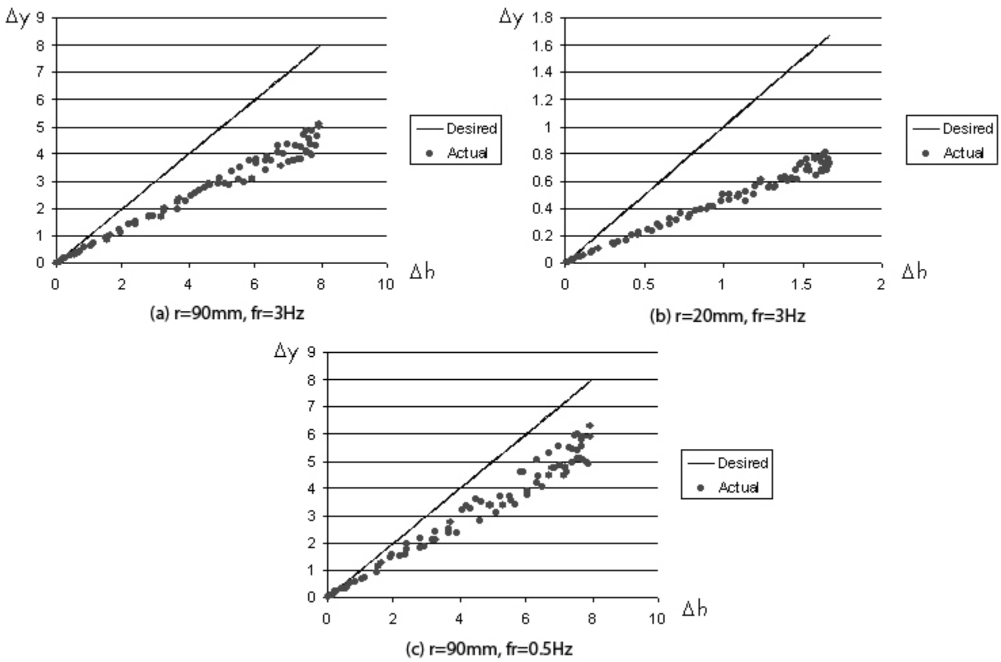



**Fig. 8.** Comparison of system response at different level of frequency and amplitude.
(a) high frequency-high amplitude response
(b) high frequency-low amplitude response
(c) low frequency-high amplitude response

Divergence of collected data in figs. 8a and 8c is determined by higher inertial force at lower frequency. As vibration direction in fig. 8b changes more rapidly than in fig. 8a, inertial forces are more intensive in recent state.

For numerical presentation of controller performance we derived best fit line for each graph of fig. 8. Table 1 summarizes presented conclusions.

**Table 1.** Comparison of system response at different situation

| Vibration amplitude (mm) | Vibration frequency (Hz) | Best Fit Line Slope (%) |
|---|---|---|
| 20 | 3 | 0.44 |
| 90 | 0.5 | 0.61 |
| 90 | 3 | 0.52 |

In comparison with obtained results of Liu et al experiments, it can be found that the major advantage of proportional damper, which was used in our experiment, is compatibility for various types of excitation unlike bi-state dampers. System response in Liu et al research work has phase lag due to damper low response. To eliminate this deficiency we used valves *b* and *c* to increase system response via releasing air entrapped in cylinder at related position.

## References


[1] **Bellmann M. A.** Perception of whole body vibration: from basic experiments to effects of seat and steering wheel vibrations on passenger's comfort inside vehicles. Dissertation in doctor of philosophy. University of Oldenburg. Germany. (2002).

[2] **Benmard R. and Levitt J. A.** Non-Linear Dynamic Modeling of an Automobile Hydraulic Active Suspension System. Mechanical Systems and Signal Processing. 8(5): 485-517. (1994).

[3] **Crolla D. A. and Abouel Nour A. M. A.** Power losses in active and passive suspensions of off-road vehicles. Journal of Terramechanics, 29: 83-93. (1992).

[4] **Duck M. and Goss G.** Investigation of tractor driver seat performance with nonlinear stiffness and on-off damper. Biosystem Engineering, 96: 477-486. (2007).

[5] **Harison H. R. and Nettleton T.** Principles of Engineering Mechanics (2$^{nd}$ ed.). London: Edward Arnold. (1994).

[6] **Hostens I., Deprez K. and Ramon H.** An improved design of air suspension for seats of mobile agricultural machines. Journal of Sound and Vibration, 276: 141-156. (2004).

[7] **Klooster S. J.** Vibration suppression and safety seat motion design of a hyper active seat. Thesis in master of science in Mechanical Engineering. Georgia Institute of Technology. USA. (2004).

[8] **Liu Y., Waters T. P., Brennan M. J.** A comparison of semi-active damping control strategies for vibration isolation of harmonic disturbances. Journal of Sound and Vibration, 280:21-39. (2005).

[9] **Nieto A. J., Morales A. L., Gonzalez A., Chicharro J. M. and Pintado P.** An analytical model of pneumatic suspensions based on an experimental characterization. Journal of Sound and Vibration, 313: 290-307. (2008).

[10] **Park S. T., Luu T. T.** An improved design for driver's seat suspension. Korus-2005: The 9th Russian-Korean International Symposium on Science and Technology. (2005).

[11] **Sciulli D.** Dynamics and Control for Vibration Isolation Design. Dissertation in doctor of philosophy. Virginia Polytechnic Institute. USA. (1997).

[12] **Uys P. E., Els P. S., Thoresson M.** Suspension settings for optimal ride comfort of off-road vehicles travelling on roads with different roughness and speeds. Journal of Terramechanics, 44:163-175. (2007).