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# Modeling and simulation of active hydro-pneumatic suspension system through bond graph

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## 1. Introduction

The modeling of physical systems is of great importance within all engineering fields because it allows us to understand the behavior of the system without having to experiment on it. 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. Moreover, it is an essential tool for the design of control strategies, which are very important at industrial level [1].

Nearly all the systems that an engineer may work with are dynamic, i.e., things are usually changing so that there is no status quo or lasting steady state. To be sure, many of important facets of a design may be based on steady-state consideration, but a new device or system will fail if it cannot withstand transient peak loads, respond quickly enough to a changing input, or operate without violent oscillations when disturbed. Dynamic analysis can predict such problem before a system is built; a system analysis which does not include the effects of significant dynamic phenomena is very likely to be worthless [2, 3]. Therefore, it is of utmost importance to have models that are able to predict the dynamic behavior of the system. One of these models, which is the focus of the present paper, is the Bond Graph (BG) model.

The Bond graph model was first developed in 1961 at MIT, Boston, by Paynter [4] and further by Karnopp, Rosenberg [5] and Thoma [6].

BG are a multidisciplinary and unified graphical modeling language which provides, from this point of view, a convenient and useful tool for model builder conception. BG is a modeling and simulation tool, providing many possibilities. It allows both a causal and a behavioral system analysis. From the behavioral point of view, BG tool allows to deal with the enormous amount of equations describing the dynamic behavior of different phenomena which occur in the system. It allows, independent of the physical nature of the studied system, precisely by its graphical nature, to display the exchange of power in a system, including storage and transformation and the instrumentation diagram (sensor location in the real process). Furthermore, BG is subject to evolution, that is to say, the model can be refined by adding graphically more elements like thermal losses or inertia and storage effects, without having to start all over again. This property is very attractive and is one of the main advantages of the BG method [7]. An exhaustive review of the BG modeling and its characteristics may be found in [8-11]. For a comprehensive review of the applications of BG, the reader is referred to [12].

The aim of the present paper is to use BG method in modeling of hydro-pneumatic suspension. A hydro-pneumatic spring consists of two fluids acting upon each other, usually gas over oil. A compressible gas, such as nitrogen is used as the springing medium, while a hydraulic fluid is used to convert pressure to force. In a pneumatic or air spring, the external force directly compresses the gas, whereas in a hydro-pneumatic suspension, hydraulic fluid is used [13-17].This system consists of mechanical, hydraulic, thermo-fluid and control subsystems. Therefore, it is a multidomain system and BG is a good candidate to model it.

The organization of this paper is as follows: section 2 deals with a brief introduction to hydro-pneumatic suspension systems. In the third section, the present system is described. In section 4, BG model of an active hydro-pneumatic suspension system and its equations are presented. In section 5, the simulation results and validation studies are reported. Finally, the conclusions drawn from the current work is presented.

## 2. Hydro-pneumatic suspension

### 2.1. Historical overview

Hydro-pneumatic suspensions have been introduced on battle tanks in the 1950's. The first hydro-pneumatic struts were fitted to a prototype tracked vehicle, as a result of research done by two German companies; Frieseke and Hopfner from Erlangen, and Borgwald from Bremen, into the use of compressible fluids in suspension systems [13].

This type of suspension system is popular due to its nonlinear characteristic and versatility. The nonlinear characteristic causes the spring rate to increase as the load is increased. It also reduces body roll and pitching, results in more constant wheel loads and usually eliminates the necessity for a sophisticated bumps top. Many controllable suspension systems make use of hydro-pneumatic springs because the hydraulic fluid can easily be channeled through ducts, orifices and valves. By adding or removing hydraulic fluid, the vehicle dynamics and the ride height can be altered [13].

The main objective of hydro-pneumatic suspension is to eliminate the body roll of a car when cornering at a high speed. Additionally, the hydraulic system can enable self-leveling, variable ride height, and assisted jacking. Furthermore, it can provide the hydraulic power to assist the braking systems and the power steering. In some car models, the high-pressure hydraulic system also operates the clutch and gear change. Generally speaking, the idea of using a hydraulic circuit as a suspension instead of springs and dampers is quite effective, and has been in use for half a century. The first automotive company to use this kind of suspension was Citroen [14].

Main components are: reservoir, high-pressure pump, main accumulator, the "load-bearing" shock-absorbers, the height control valves.

2.2. The suspension subsystem

The suspension subsystem is fed directly from the main accumulator. The fluid feed immediately splits into front and rear branches, each passing through a height control valve (Fig. 1). When each valve is activated, high-pressure fluid inflates the pair of load-bearing shocks. When the valve is in the neutral position, the pressure level remains constant between the pair. When the valve is deactivated, the fluid in the shock pair drains directly back to the reservoir.

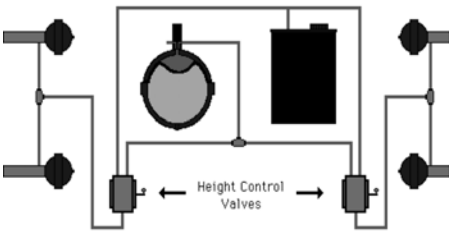


Fig. 1 The schematic of the suspension subsystem

Sharing the pressure between left and right shock absorbers proved to have many benefits. The tendency to equalize pressure between them accomplished horizontal self-leveling, even at high speeds. But it proved more advantageous to have pressure separated fore and aft. This was accomplished through the independent height control valves on the front and rear axis. If the load on the rear of the car increased, the rear valve would be activated and a greater volume of high-pressure fluid would be allowed into that pair.

**3. Setup of the intended system**

The focus of this paper lays on an Active Hydro-Pneumatic suspension (AHP). At the heart of each AHP suspension strut, there is a force controller, which is responsible for tracking a certain desired force (calculated by other, higher level or "outer loop" controllers). The innermost control task is to set and track certain pressure in the cylinder, which translates into the force then exerted by the piston. The reference force (pressure) is the result of several higher (outer) control loops that, for instance, compensate the cars tendency to roll in corners or pitch with changing longitudinal acceleration [15].

It is important to note that everything discussed here concerns a single wheel only. In the car, however, (except for the pump) four of these systems are required, one for each wheel.

3.1. Components

The setup of the hydro-pneumatic system is shown in Fig. 2.

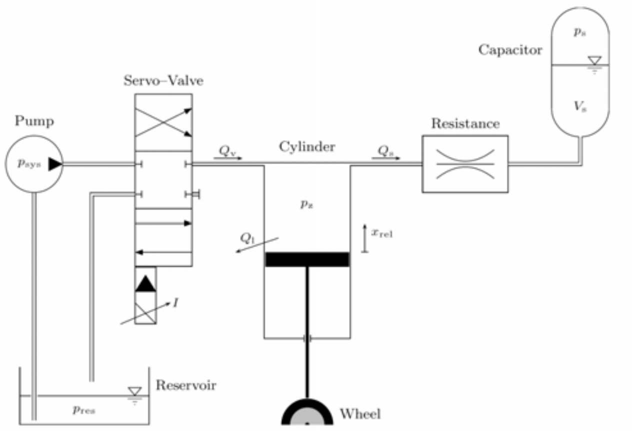


Fig. 2 Basic set-up of the system

Here, one suspension strut is made up of a:

-cylinder, which is connected directly to the wheel;

-hydraulic capacitor or "gas spring", consisting of two chambers, one connected to the oil circuit, the other, separated by a membrane, contains gas;

-laminar resistance between the cylinder and the capacitor;

-hydraulic pump connected to the car's engine;

-4/3 servo valve which controls the in- and outflow of oil to and from the system.

Pressures in the cylinder and capacitor ($p_z$ and $p_s$), the system pressure $p_{sys}$ and the pressure in the reservoir $p_{res}$ are measured by suitable sensors and are available for the use in controllers.

This could be the oil leaving the system through some worn out fittings, especially in the valve, where it may be that a certain amount of oil does not flow into the system, but directly into the reservoir for instance. We then have the control current for the valve $I$, which, for positive $I$, injects oil into the system, and for negative $I$ allows oil to leave it. The position of the plunger is $x_{rel}$; it is zero at the neutral (middle) position, positive if it is "above" that position, negative when it is "below" it. We also allow for some external force $F_{ext}(t)$, which could result from the car running over a bump on the road for instance.

As mentioned above, a certain pressure in the cylinder translates (via the effective surface of the plunger) into a force. This force, diminished by some friction, will accelerate the body of the car sitting on top of the cylinder.

3.2. Inner control loop

We shall now take a quick look at the existing controller in the inner loop. By its structure it is a PI controller. Its input is the difference between current and reference force exerted by the cylinder, and its immediate output is a desired oil flow into (or out of) the system. As the actuator is the valve, which takes a specific control current and "translates" it into the wanted oil flow, an inverse valve model is used to determine the necessary control current needed.

3.3. Outer control loops

The cascaded control system of AHP suspension contains two "layers". Several (parallel) components calculate the desired force (or reference force) on the outer layer, which is then to be set and tracked by the inner loop.

## 4. Bond Graph model

The modeling of the system has been performed through BG simulation technique which is an effective tool for modeling and simulation of physical systems. It facilitates the exchange, storage and dissipation of energy among interacting physical elements efficiently. The bonds of the model portray the paths of the exchange of power within the constraint structure and atomic elements. It is to be noted that all bond graphs including the present one are lumped element representation [16].

BG model of the system shown in Fig. 3 is described as follow.

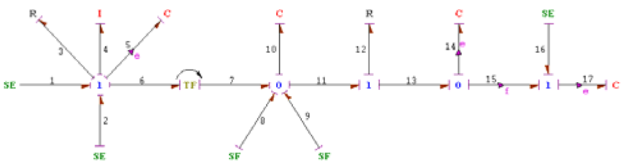

Fig. 3 BG model of the system

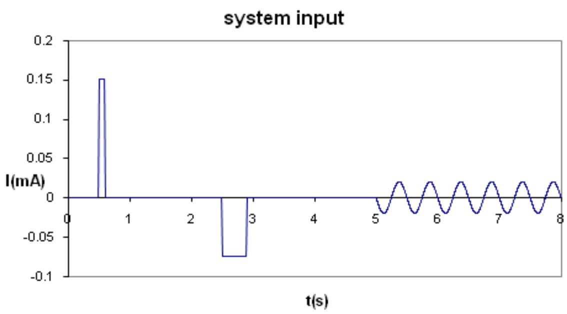

Fig. 4 System input

The force exerted to the vehicle from the road is denoted by $SE_1$, which is considered an external force equal to $F_{ext}$. $SE_2$ is the weight of the vehicle and $I_4$ denotes a quarter of the vehicle inertia. Friction coefficient between the cylinder wall and the piston is denoted by $R_3$. Element $C_5$, which is an activate bond, is an observer that monitors piston displacements. Element *TF* is used to convert force to pressure, and velocity to volume-flow-rate. In fact, this element transfers energy from mechanical domain into the hydraulic domain. The volume flow rate of hydraulic oil, $Q_v$ in Fig. 2, is considered as $SF_8$. This parameter is considered as an input to the system, and is calculated via Eq. 6. $SF_9$ is considered as leakage flow rate, $Q_l$ in Fig. 2, which is neglected in the present study. Oil compressibility is defined by element $C_{10}$. Element $R_{12}$ is the resistance due to pressure drop in the orifice located between the cylinder and the capacitor. The oil volume entering the capacitor is described by element $C_{14}$ which is activated because the hydro-static pressure of oil in the capacitor is neglected in the present study. The last portion of the bond graph in Fig. 3 is the nitrogen tank, for which the ideal gas law is considered to prevail. As seen in Fig. 3, the entire system states are $q_5$, $p_4$, $q_{10}$, and $q_{14}$. These are, respectively, the piston displacement, the piston speed, the in-cylinder oil volume, and the oil volume inside the capacitor.

In the aforementioned model, the control subsystem has not been considered and the controller output ($I$) has been regarded as a model input (Fig. 4), which motivates the valve. As mentioned earlier, for positive $I$ values, oil is injected into the system, and for negative $I$ values, oil is allowed to leave the system.

4.1. The governing equations of the system

The system equations derived from the BG model are described as follows:

First equations

$$\dot{q}_5 = f_5 = f_4 = \frac{P_4}{I_4} \rightarrow \dot{q}_5 = \frac{P_4}{I_4} \tag{1}$$

Second equations

$$\dot{P}_4 = e_4 = e_1 + e_2 - e_3 - e_6\,;\; e_1 = SE_1 = F_{ext}\,;$$

$$e_2 = SE_2 = -mg\,;\; e_3 = F_s = -F_{fr}\,;$$

$$e_6 = A_z e_7 = A_z e_{10} = A_z \frac{q_{10}}{C_{10}}\,;$$

$$\dot{P}_4 = F_{ext} + F_{fr} - mg - \frac{A_z}{C_{10}} q_{10} \tag{2}$$

Third equations

$$\dot{q}_{10} = f_{10} = f_7 + f_8 + f_9 - f_{11}\,;\; f_7 = A_z f_6 = A_z f_4 = \frac{A_z}{I_4} P_4\,;$$

$$f_8 = Q_v\,;\; f_{11} = f_{12} = \frac{1}{R_{12}} e_{12} = \frac{1}{R_d}(e_{11} - e_{13})$$

$$e_{11} = e_{10} = \frac{q_{10}}{C_{10}}\,;\; e_{13} = -SE_{16}\,;\quad f_{11} = \frac{1}{R_d}\left[\frac{1}{C_{10}} q_{10} - (-SE_{16})\right];$$

$$P_a V_a^{\,K} = P_g V_g^{\,K} \rightarrow P_g = P_a \left(\frac{V_a}{V_g}\right)^K;\quad SE_{16} = -P_a \left(\frac{V_a}{V_M - q_{14}}\right)^K;$$

$$\dot{q}_{10} = \frac{A_z}{I_4} P_4 + Q_v - Q_l - \frac{1}{R_d}\left[\frac{q_{10}}{C_{10}} - P_a \left(\frac{V_a}{V_M - q_{14}}\right)^K\right] \tag{3}$$

Fourth equations

$$\dot{q}_{14} = f_{14} = f_{12};$$

$$\dot{q}_{14} = \frac{1}{R_d}\left[\frac{1}{C_{10}} q_{10} - P_a \left(\frac{V_a}{V_M - q_{14}}\right)^K\right] \tag{4}$$

In the above equations, friction force ($F_{fr}$) and oil flow rate from valve ($Q_v$) are defined as in [15]

$$F_{fr} = -\frac{F_{c_2}}{\pi/2} tan^{-1}(-K_1 x_2) - \frac{F_{m_2}}{\pi/2} tan^{-1}(-K_2 x_2) +$$
$$+ d_{V_2} x_2;$$

$$\rightarrow F_{fr} = -\frac{F_{c_2}}{\pi/2} tan^{-1}\left(-K_1 \frac{P_2}{I_2}\right) - $$
$$-\frac{F_{m_2}}{\pi/2} tan^{-1}\left(-K_2 \frac{P_2}{I_2}\right) + d_{V_2}\frac{P_2}{I_2} \quad (5)$$

$$\left.\begin{array}{l} Q_v(I,P_z) = \begin{cases} K_v Sat(I)\sqrt{P_{sys} - P_z} & I \ge 0 \\ K_v Sat(I)\sqrt{P_z - P_{res}} & I < 0 \end{cases} \\ Q_v(I,P_z) = \begin{cases} K_v Sat(I)\sqrt{P_{sys} - \dfrac{q_{10}}{C_{10}}} & I \ge 0 \\ K_v Sat(I)\sqrt{\dfrac{q_{10}}{C_{10}} - P_{res}} & I < 0 \end{cases} \end{array}\right\} \quad (6)$$

Where *Sat* (*I*) is a limited function acts on input signal of the valve. Saturating *I* at $\pm I_s$ as the opening fraction of the valve is, of course, physically limited. In the present study input signals as shown in Fig. 4 are under saturation limit.

4.2. System parameters

Oil pressure in the cylinder

$$P_Z = \frac{E}{V_{Z_0}} V_Z$$

In bond graph view

$$e_{10} = P_Z ; \rightarrow e_{10} = \frac{1}{C_{10}} q_{10} ,\ q_{10} = V_Z ;$$

$$\rightarrow C_{10} = \frac{V_{Z_0}}{E} = 4.08\times 10^{-13} \quad (7)$$

$$P_{Z_0} = 35\times 10^5 ;$$
$$\rightarrow q_{10_0} = C_{10} e_{10_0} = 4.08\times 10^{-13} \times 35\times 10^5 ;$$
$$\rightarrow q_{10_0} = 1.428\times 10^{-6} ;$$
$$P_a V_a{}^K = P_{s_0} V_{s_0}{}^K ;$$
$$\rightarrow V_{s_0} = V_a \left(\frac{P_a}{P_{s_0}}\right)^{\frac{1}{K}} = 1.13\bullet 10^{-4}\left(\frac{42}{35}\right)^{\frac{1}{1.36}} =$$
$$= 1.3057\times 10^{-4} \quad (8)$$

Therefore accumulator volume and initial volume of nitrogen:

$$V_M = V_{s_0} = 1.3057\times 10^{-4}$$

Finally, system parameters are shown in Table 1.

Table 1

System parameters

| | | | |
|---|---|---|---|
| $M = I_4 = 365$ kg | $K = 1.36$ | $E = 2\times 10^8$ Pa | $A_z = 10.2\times 10^{-4}$ m$^2$ |
| $R_d = 2.08\times 10^9$Pa/(m$^3$/s) | $Q_1 = 0$ m$^3$/s | $Kv = 5.9\times 10^{-7}$ m$^3$/s/Pa$^{1/2}$A | $Psys = 180\times 10^5$ Pa |
| $P_{res} = 10^5$Pa | $Va = 1.13\times 10^{-4}$ m$^3$ | $Pa = 42.6\times 10^5$ Pa | $K_1 = 1937$ s/m |
| $F_{c2} = 150$ N | $d_{v2} = 20$ N/(m/s) | $F_{m2} = 100$ N | $K_2 = -50$ s/m |
| $V_{Z0} = 8.16\times 10^{-5}$ m$^3$ | $V_M = 1.3057\times 10^{-4}$ m$^3$ | $C_{10} = 4.08\times 10^{-13}$ m$^3$/Pa | $F_{ext} = 7158.4$ N |

4.3. Initial condition

The above set of equations is amenable to numerical methods after defining a proper set of initial conditions. The vector form of the initial conditions is presented in Eq. 9

$$X = \begin{bmatrix} q_{5_0} \\ P_{4_0} \\ q_{10_0} \\ q_{14_0} \end{bmatrix} = \begin{bmatrix} 0 \\ 0 \\ 1.428\times 10^{-6} \\ 0 \end{bmatrix} . \quad (9)$$

**5. Results**

Figs. 6, 7 and 8 are, respectively, the piston position $x_{rel}$, the oil pressure $P_z$, and the nitrogen pressure $P_s$ in the system versus time.

It is observed in these figures that during the first time interval, pressure (in both cylinder and capacitor) builds up to a certain level. Oil is forced into the system, but at the beginning (as the piston is not moving) most of the oil has to go into the capacitor. The increase of $P_z$ and

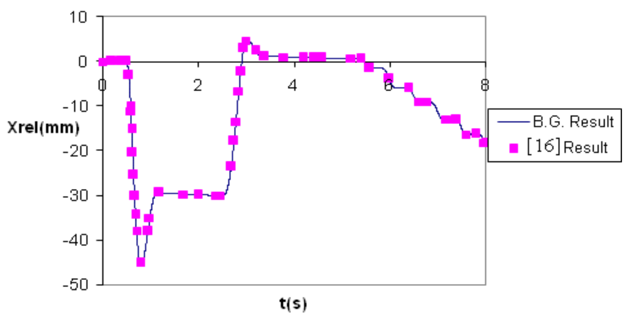


Fig. 5 Piston position $x_{real}$

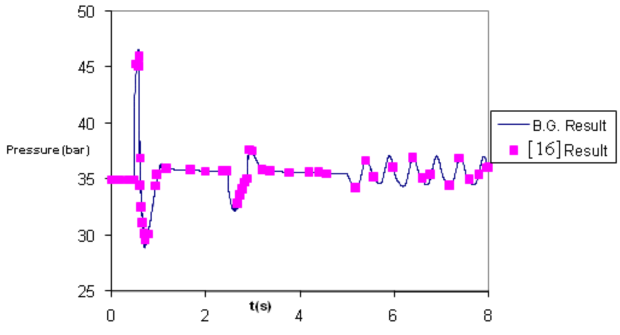


Fig. 6 Oil pressure $P_Z$

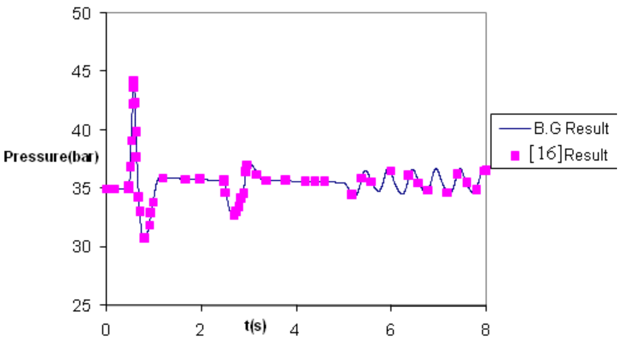


Fig. 7 Nitrogen pressure $P_S$

$P_s$ starts to create a force which is greater than that gene–rated by the external force. Then the plunger is accelerated outwards. When it starts moving, the available volume in the cylinder increases, resulting in oil flow back from the capacitor, and the subsequent decrease in pressure. After the plunger (and the considered part of the car mass) starts moving, it comes to a stop due to stoppage of the oil flow. In the second time interval, associated to the second pulse, the opposite of the above description is relevant (i.e. oil is taken from the capacitor resulting in a lack of pressure, which in turn results in deceleration, as the external force of the car is bigger than that generated by the piston).

For the sinusoidal part above descriptions are true but there is not too much to discuss for the sinusoidal part, because oil is entered into or is left from the system frequently in low periods.

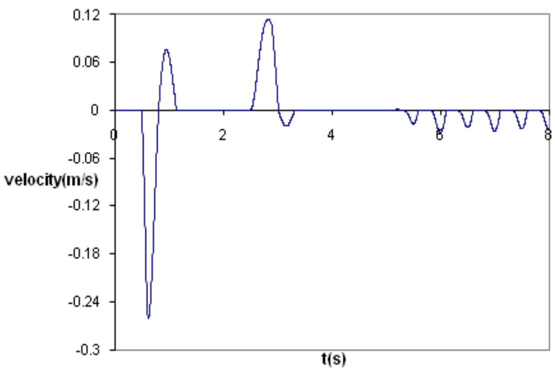


Fig. 8 Piston velocity $V$

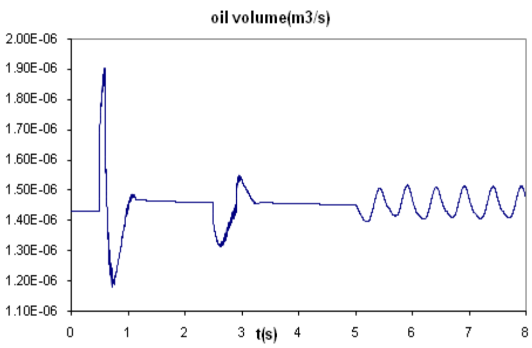


Fig. 9 Oil volume in the cylinder $V$

Moreover, by solving the state equations, the piston speed and the oil volume inside the cylinder may be calculated as shown in Figs. 9 and 10. As mentioned before, the downward movement of the piston increases the cylinder volume, affecting both oil volume and reservoir pressure to decrease, as is seen in these figures. For upward movement of the piston, the opposite of the above description is relevant again.

## 6. Conclusions

An introduction is made to the main principles of active hydro-pneumatic suspension system and how the combination of fluid, gas and several accumulators in a suspension can provide both a good level of comfort and good handling. Bond graphs have been introduced to model physical systems in a domain independent way. Domain independency stems from the fact that physical concepts are analogous for the different physical domains. As a typical hydro-pneumatic suspension system includes various phenomena in the field of hydraulic, thermo-fluid, mechanical and control, the BG approach may render itself a convenient choice for analyzing such systems. In this study a submodel for one wheel is created. The model uses adiabatic ideal gas compression assumption for spring force and dynamic friction submodel for the friction force. The simulation results have been in accordance with others results. Overall, the present results show that the bond graph is indeed a suitable method for such applications.

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AKTYVIOS HIDRO-PNEUMATINĖS PAKABOS SISTEMOS MODELIAVIMAS IR IMITAVIMAS NAUDOJANT GRAFŲ TEORIJĄ

R e z i u m ė

Šiame straipsnyje aprašytas aktyvios hidro-pneumatinės pakabos sistemos modelis. Tai yra palyginti naujas pakabos projektas, kuriame sistema veikia be klasikinių mechaninių elementų, tokių kaip spyruoklės ir amortizatoriai. Vietoj hidraulinės sistemos panaudotas plunžerinis cilindras, tekėjimo ribotuvas, hidropneumatinis akumuliatorius ir galingas hidraulinis siurblys kartu su greitaveikiu valdymo vožtuvu. Kiekvienos aktyvios hidro-pneumatinės pakabos sistemos pagrindas yra jėgos matuoklis, ribojantis veikiančią jėgą. Sistemos modeliavimas atliktas naudojant imitavimą grafų teorijos technika. Visi sistemos komponentai buvo pakeisti jų grafų teorijos pakaitalais. Pagrindinės lygybės buvo sudarytos ir išspręstos naudojant grafų teorijos modelius. Grafų teorijos rezultatai palyginti su kitais rezultatais ir nustatyta, kad grafų teorijos išvados sutampa su kitų modelių išvadomis. Taigi, grafų teorijos taikymas yra dar vienas geras būdas sudėtingų sistemų komponentams ir fiziniams modeliams projektuoti.

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MODELING AND SIMULATION OF ACTIVE HYDRO-PNEUMATIC SUSPENSION SYSTEM THROUGH BOND GRAPH

S u m m a r y

This article presents a model for active hydro-pneumatic (AHP) suspension system. This is a relatively new suspension design, in which the system works without the classical mechanical parts of a suspension, such as steel springs and dampers. Instead, a hydraulic system is used which consists of a plunger cylinder, a flow resistance, a hydro-pneumatic capacitor and a strong hydraulic pump together with a fast response servo valve. At the heart of each AHP suspension system, there is a force controller, which is responsible for tracking a certain desired force. The modeling of the system has been made using the bond graph simulation technique. All of the components of the system have been replaced by their bond graph counterparts. The governing equations are written in terms of bond graph models, and are solved simultaneously. Bond graph results are compared to the results of others. The bond graph method succeeds in reproducing the same outputs as the other methods. However, the ability of the bond graph modeling in adapting to new changes in the system components and physical models makes it a good choice for complex systems.

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МОДЕЛИРОВАНИЕ И ИМИТИРОВАНИЕ АКТИВНОЙ СИСТЕМЫ ГИДРО-ПНЕВМАТИЧЕСКОЙ ПОДВЕСКИ С ИСПОЛЬЗОВАНИЕМ ТЕОРИИ ГРАФОВ

Р е з ю м е

В этой статье представлена модель активной системы гидропневматической подвески. Это сравнительно новый проект подвески, в котором система работает без классических механических элементов, таких как пружины и амортизаторы. Вместо гидравлической системы использован плунжерный цилиндр, ограничитель течения, гидропневматический аккумулятор и мощный гидравлический насос с быстродействующим сервоклапаном. Основой каждой активной системы гидропневматической подвески является ограничитель действующей нагрузки. Для моделирования системы использовано имитирование техникой теории графов. Все элементы системы были заменены дубликатами теории графов. Основные уравнения составлены и решены используя модели теории графов. Результаты теории графов сопоставлены с другими результатами и установлено соответствие выводов теории графов с выводами других моделей. Таким образом использование теории графов для проектирования компонентов сложных систем и физических моделей является еще одним хорошим предложением.

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