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MODELLING A SELECTIVE HYDROPNEUMATIC SUSPENSION ELEMENT

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Abstract

The aim of this study was to create a model of a selective hydropneumatic suspension element (gas spring and a fluid damper). The model is intended as a basis for a full vehicle selective suspension simulation model. When defining and testing the damping rates for different kinds of terrains, advanced numerical simulation models are very valuable. Also, when investigating different kinds of control algorithms for selecting the optimal damping rate, a model of a selective shock absorber is required.

The model was created using MATLAB/Simulink block diagram representation. The model takes account of fluid mechanics, thermodynamics, friction and non-linearities of various sub-elements. Also the electrical system of the driving actuator was modelled. The parameters of the model were chosen so that the model could be scaled for different sizes of shock absorbers. A set of parameter values were measured from an experimental hydropneumatic suspension system and the behaviour of the model and the real system were compared.

As a result, a parameterized, non-linear model of a selective hydropneumatic suspension element was obtained. The model serves as a sub-model for full vehicle simulation models and provides an input for a control algorithm to select the damping rate.

INTRODUCTION

A wide variety of off-road vehicles make use of hydropneumatic suspension. Hydropneumatic suspension has typically a low space claim, high degree of integration and the possibility to include a leveling system without complicated mechanisms. Most of the present hydropneumatic systems are totally passive, but controllable systems also exist. Passive systems are designed for a certain range of operating conditions. The performance attained is a sort of comprise between handling and comfort. Controllable systems are aimed to provide better handling without the penalty of decreased vibration isolation.

Extensive research has been carried out in the field of controllable suspension to overcome the restrictions of fully passive systems [4,5]. Research has mostly been concentrated on semi-active and active systems. However, active systems are still rare in serial production vehicles due to their high price and energy consumption. Semi-active systems are more common than active ones, but they are still associated with high-end models. Automatically controlled selective suspension provides an low-cost alternative for semi-active and active suspension. Passive-selective suspension systems have been around for decades, but the vehicle driver has been liable for the selection of the suspension state (damping, spring rate or both). However, by making use of sophisticated control electronics, it is possible to use complex reasoning algorithms in suspension state selection.

This study concentrates on the modeling of a multi-state hydropneumatic suspension system. The model is comprised of a hydropneumatic spring, variable hydraulic valve and an actuator for the valve. Contrary to electrical systems, hydraulic systems are typically highly non-linear and more prone to parameter variations. These aspects lead to the need for numerical simulation models which take non-linearities into consideration. The model under consideration is developed in the MATLAB/Simulink environment. The block diagram representation provides a straightforward way to describe a complicated system like a hydropneumatic suspension.

THE MODELLED SYSTEM

Hydropneumatic suspension system consists of a hydraulic cylinder, accumulator and damping valve. The movement of the suspended object is converted to fluid flow by means of a hydraulic cylinder. Supporting spring force is achieved by gas compression in the hydraulic accumulator and the damping force is generated due to pressure loss in the damping valve. The system is illustrated in figure 1 - a.

The force generated by the hydropneumatic suspension element is the sum of forces achieved by the accumulator and damping valve due to pressure loss. An additional force generated by the hydraulic cylinder sealing system has to be taken account. Each of the force components will establish an independent system and can be modelled as its own block.

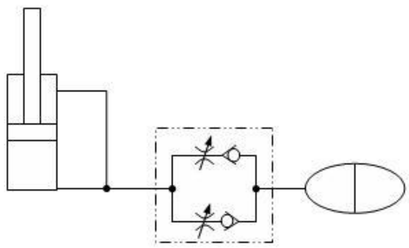

Figure 1 - -a. The system modelled

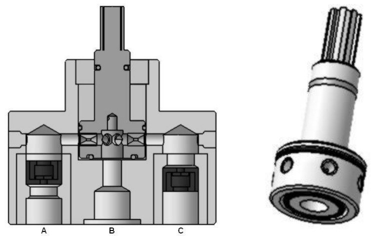

Figure 1 - b. The selective damping valve.

The modelled selective damping valve is illustrated in figure 1-b. Rotating movement of the rotor enables to choosing the active orifice. Two opposite orifices are used at the same time (See fig. 1 – a). The rotating movement is achieved by means of a DC – motor.

SPRING FORCE

The spring force is achieved by means of gas compression in the hydraulic accumulator due to fluid flow. In this paper the gas compression process is assumed to be adiabatic and can be represented with the equation 1 [3]. In equation 1, P_0 and V_0 are the initial gas pressure and gas volume, and P_1 is the gas pressure when gas volume has been changed to V_1. In adiabatic process the isentropic exponent κ has value of 1.4.

$$P_0 V_0^{\kappa} = P_1 V_1^{\kappa} \tag{1}$$

The spring model is based on the assumption that the gas specific volume remains constant during the action and the system is fully restricted. This would mean that there is no energy leaving as heat from the system and there is no energy brought as heat to the system. These assumptions are not quite exact but they are very close. The velocity of the suspension element is so high that no significant amount of heat energy will leave the system through the cylinder walls to surroundings.

The fluid volume in the hydraulic accumulator depends on the displacement of the hydraulic cylinder. Solving P_1 from equation 1 and writing V_1 in terms of rod displacement x the equation 2 can be found.

$$P(x) = P_0 \cdot \left(\frac{V_0}{V_0 + Ax} \right)^{\kappa} \tag{2}$$

The initial condition is convenient to choose as the static equilibrium point of the system. In this condition (x = 0) the gas pressure, P_0, can be calculated by means of the cylinder rod area, A, and the sprung mass. V_0 can be calculated due to fluid volume in accumulator (by rod area and rod displacement). Spring force generated is the cylinder rod area times the system pressure (Equation 3).

$$F_s(x) = A \cdot P_0 \cdot \left(\frac{V_0}{V_0 + A \cdot x} \right)^{\kappa} \tag{3}$$

FRICTION FORCE

A dynamic friction model [1] was used in friction modelling. Typically friction is modelled with static, coulomb and viscous friction. These friction models produce a direct relation between velocity and friction force but do not take account the hysterics at low velocities. Hydraulic cylinder seals work as a spring in the Stribeck region and the dynamic friction model can simulate seal friction behaviour more accurately.

$$\frac{dz}{dt} = v - \frac{|v|}{g(v)} z \tag{4}$$

$$g(v) = \frac{1}{\sigma_0} \left(F_C + (F_S - F_C) \cdot e^{-(v/v_S)^2} \right) \tag{5}$$

$$F = \sigma_0 z + \sigma_1(v) \frac{dz}{dt} + F_V v \tag{6}$$

The dynamic friction model is characterized by six parameters: The seal stiffness σ_0, the seal damping coefficient σ_1, the viscous friction level σ_2, the Coulomb friction level F_C, the stiction force level F_S and the Stribeck velocity v_s which have to be fitted with measurements. Measurements were done over a large velocity range and parameters were fitted to each velocity measurement. Then average parameters were evaluated to fit all these measurements as well as possible. Figure 2 represents the simulated and measured force (spring force and friction force) when the cylinder has forced sine shaped motion (amplitude 20mm and frequency 2Hz).

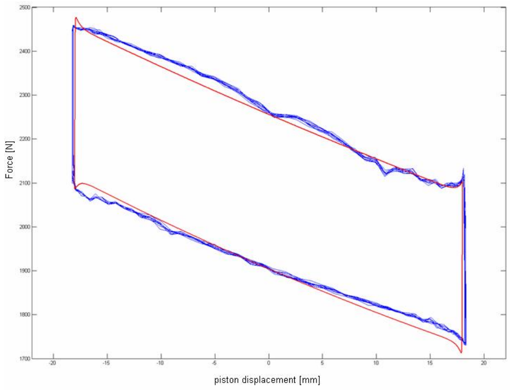

Figure 2. Simulated(red) and measured(blue) spring and friction force.

DAMPING FORCE

The damping force generated by the element is achieved by the fluid flow through the damping valve. The flow, Q, caused by the cylinder movement can be calculated with the cylinder rod area and the rod velocity. Assuming turbulent flow, the flow can be defined with the pressure loss achieved in an orifice with cross-section area of A_d. Equation 7 can be written [2]. In equation 7, ρ denotes the fluid density and the discharge coefficient, C_d, is usually about 0.7 in hydropneumatic suspension systems.

$$Q = C_d \cdot A_d \cdot \sqrt{\frac{2 \cdot \Delta P}{\rho}} \tag{7}$$

Solving pressure loss ΔP from equation 7 and taking into account the circular shaped cross-section of the orifices, equation 8 for pressure loss can be written.

$$\Delta P = \frac{8 \cdot \rho}{C_d^2 \cdot d_h^4 \cdot \pi^2} Q^2 \tag{8}$$

The hydraulic diameter d_h is a function of the valve rotor angle θ. When defining the hydraulic diameter as a function of θ, two assumptions are made: The shape of the flow channel cross-section remains circular with every value of θ and, the flow and leaks are always turbulent. There are three values of θ that correspond with their hydraulic diameters to chosen orifice diameters.

Figure 2-a illustrates the measured pressure loss in valve when turning the rotor and keeping the flow constant. Because decrease in hydraulic diameter increases pressure loss, it is convenient to choose the hydraulic diameter to vary as a second (or higher, see fig. 2 -a) order function of θ with a negative highest order coefficient. The complete function includes three functions each representing the diameter behaviour in the region of the chosen orifice. The function is therefore discontinuous and has its

discontinuity points with values of θ that will correspond to the valve closed condition between the orifices. The value of hydraulic diameter at discontinuity points represents the leaks in the valve. In the modelled valve, there are three orifices placed at 45 degrees to each other. This means that θ has values 0…90°. At angle θ = 0°, the hydraulic diameter has the value of the chosen orifice diameter. As θ increases, the hydraulic diameter decreases until the fully closed condition is attained. In the modelled valve the fully closed conditions are at θ = 22.5^{o} and θ = 67.5^{o}. To model this behaviour, equation 9 is developed. In equation 6 terms a and b are mechanical construction dependent constants.

$$d_h = \begin{cases} (a \cdot d_1 + b)\theta^4 + d_1, \theta \text{ p } 22.5 \\ (a \cdot d_2 + b)\cdot(\theta - 45)^4 + d_2, 22.5 \le \theta \text{ p } 67.5 \\ (a \cdot d_3 + b)\cdot(\theta - 90)^4 + d_3, \theta \ge 67.5 \end{cases} \tag{9}$$

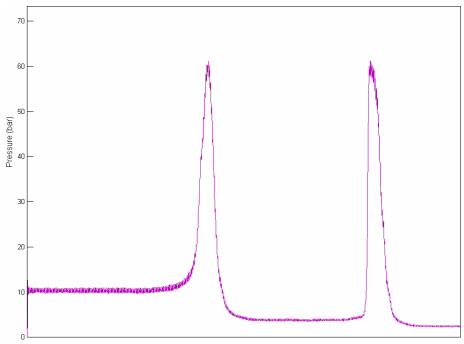

Figure 2 –a. Measured P(θ)-behaviour.

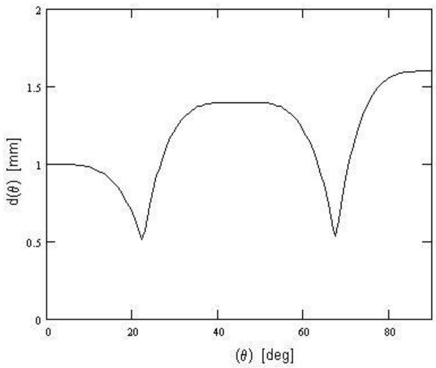

Figure 2-b. Hydraulic diameter.

Equation 9 produces the curve illustrated in figure 2 - b. By connecting equation 8 and equation 9, the function ΔP(Q,θ) is obtained. This is the pressure loss function that relates the flow and rotor angle together and gives the pressure loss generated in the valve. This is illustrated in figure 3. The damping force is obtained from pressure loss in the damping valve by multiplication with the rod cross-section area.

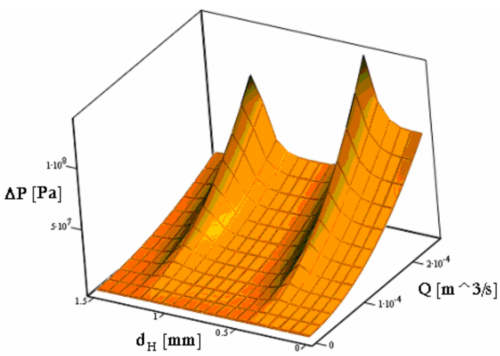

Figure 3. Pressure loss chart

THE VALVE ACTUATOR

The valve is driven by means of a DC – motor. With the construction discussed in this paper, the duration of damping state change can be measured in seconds. Modelling the actuator which is used to drive the valve rotor gives the possibility to study the behaviour of the system during the damping state selection. The motor torque, T, is related to the armature current, I, by the constant factor, K_t, and the back emf, e, is related to the rotational velocity dθ/dt by equations 10 and 11.

$$T = K_t i \tag{10}$$

$$e = K_e \frac{d\theta}{dt} \tag{11}$$

The motor is modelled with differential equations based on Newton’s and Kirchoff’s laws (equations 12 and 13).

$$J \frac{d^2\theta}{dt^2} = T - b \frac{d\theta}{dt} \tag{12}$$

$$L \frac{di}{dt} = -Ri + V - e \tag{13}$$

In equations 9 and 10, J is rotor and load inertia, b is the damping ratio, L is the electric inductance and R is the electric resistance.

COMBINED MODEL

The element model (see fig. 3) has all force components summed. The model takes the diameters of the chosen orifices, motor drive voltage and piston displacement as inputs. The spring force has always the same sign. The damping force has always the sign opposite to piston velocity. The sign of the damping force can be taken account with the switch – block in MATLAB/Simulink.

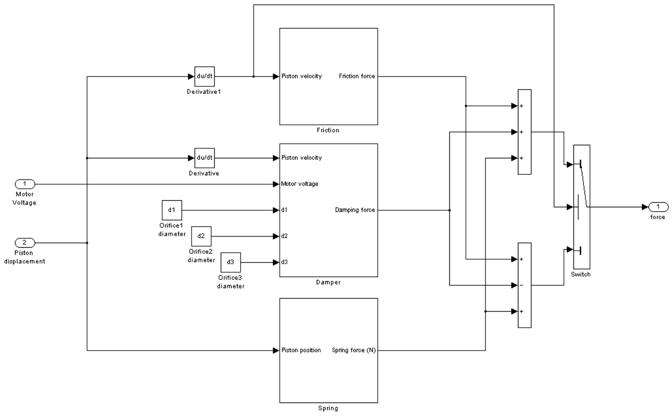

Figure 3. Element model.

SUMMARY

In this study a sub – model for full vehicle simulation model was created. The model was parameterized and verified. The model created uses an adiabatic ideal gas compression model for spring force, a selective hydraulic damping valve model for the damping force and a dynamic friction model for the friction force. The selective damping valve actuator was also modelled.

The selective damping valve construction was assumed to be such that rotating the valve rotor will not change the shape of the cross-section of the orifice. Leak flow was assumed to be turbulent. These assumptions are not quite exact, but close enough. Parameters describing the leak flows and valve construction were introduced.

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