Modeling and Simulation of a Biphasic Media Variable Stiffness Actuator

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***Abstract*—Nowadays, an increasing number of industrial processes are expected to have robots interacting safely with humans and the environment. Compliance control of robotic systems strongly addresses these scenarios. This article develops a variable stiffness actuator (VSA) whose position and stiffness can be controlled independently. The method for adapting the stiffness in the VSA includes a member configured to transmit motion that is connected to a fluidic circuit, into which a biphasic control fluid circulates. Actuator's stiffness is manipulated by varying pressure of control fluid into distribution lines. The control fluid used is composed of gas and liquid, which are separated from each other and in proportions with predefined ratio. An approach for the mathematical model is introduced and a model-based control method is implemented to track the desired position and stiffness. Results from force loaded and unloaded simulations and possible applications of the system are discussed.**

***Index Terms*—Biphasic media, variable stiffness actuator, stiffness control, position control**

I. INTRODUCTION

The problem of controlling stiffness in actuators used in machines appears when a robot is needed to perform motion tasks in the presence of humans, or when collisions with the environment are possible. For these scenarios, velocity and position should be sufficiently accurate while minimizing the risk of damaging humans, environment and itself [1-2]. This need for safety is extended to unintended interactions due to hardware failure, limitations on perception and cognition. A compliant system has improved open loop characteristics by reducing the reflected inertia on the contact side of the impact [3]. The applications that require controllable stiffness can be identified as "robot-human interaction" and "natural dynamics adjustment" [4]. The first one is focused on having a safer and more natural interaction between the human and the machine, some examples are industrial robots [5-7] and human rehabilitation devices [8], while the second one deals with the adjustment of natural dynamics of the mechanical system in order to have a desired natural motion to reduce the energy consumption: robotic prosthesis [9] and legged robots [10] are the most notable examples.

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Several types of Variable Stiffness Actuators (VSA) have been developed, the majority use mechanical springs and other elastic elements along with motors to obtain the desired position [2]. These actuators adapt the stiffness based on sensory feedbacks. The effectiveness of this approach is limited by the bandwidth of the system [11-12], which is mainly driven by the response time of the controller that is using information from the sensors.

Pneumatic actuators have been used to achieve variable stiffness in robotic devices. The approach presented in [13] considers the dynamic characteristics of a pneumatic cylinder as a series elastic actuator, by replacing a four-way servo valve with a couple of three-way valves, which allows the user to modulate the stiffness and the output force.

Variable Stiffness Device (VSD) units with nonlinear stiffness characteristics were developed to adjust the compliance of cable-driven manipulators in [14-15]. The total stiffness is given from cable stiffness and VSD by attaching in series a VSD along each cable of the manipulator. Robot's stiffness can thus be controlled by manipulating cable tensions.

The idea of biphasic media used for variable stiffness actuation has been studied in [1,5-8, 11-12]. The models were designed using pneumatic and hydraulic components. The gas fraction works as nonlinear elastic element, providing variation of stiffness of the system due to pressure gas changes, while the liquid one is assumed incompressible and used to provide pressure changes and motion to the output link, as shown in Figure 1. The major implementations of the system are industrial grippers for garment handling and surgical robotics concepts.

The following sections elaborately describe mathematical modeling of the actuator and development of a model-based control law to regulate position and stiffness, then simulation results are analyzed, and finally possible and feasible applications are discussed.

II. SYSTEM MODELING

The description of model is based on the method and architecture described in [1] and [11]. The control fluid used is composed of two different non-mixable fluids. In this system, force applied to the environment depends on pressure difference between the distribution lines. Sensors are used to measure pressures in the system.

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In event of collision, the pressure in one of the lines increases, leading to compression of gas in the chamber, which thresholds the actuation force applied to the driven link by the actuator and the inertial force applied to the driven link through the actuator by the preceding link.

Based on the notations of Fig. 1, the actuator is composed of a double acting hydraulic cylinder (Cyl1) connected to a hydraulic circuit, inside which a control fluid circulates. The hydraulic circuit includes a supply and distribution system (HyS). The circuit has two hydraulic distribution lines (HyC1 and HyC2) respectively connected to the cylinder's chambers (1 and 2) and the hydropneumatic accumulators (AC1 and AC2). The control fluid encompasses two different fluids that can be both compressible and are in proportions according to a predefined ratio; in this case, liquid (oil) and gas (nitrogen), having the compressible fluid isolated inside the accumulators. O-ring type seals (ORS) provide the translation of the piston without leakage. The hydraulic circuit must include means (pumps and valves) that vary pressure inside the chambers with aim of controlling stiffness of the actuator.

Considering a displacement of the piston compressing the fluid contained in chamber 1, in such a way that it behaves as the suction chamber, while chamber 2 acts as the discharge one. The system will follow one of the three following modes:

- The volume in AC2 remains constant, the fluid flowing respectively inside/outside of the suction and discharge chambers is the same, and the piston moves at constant velocity, while the stiffness of the system is constant.
- The volume in AC2 decreases, the fluid entering in discharge chamber is higher than the volume of the control fluid leaving suction chamber, and the stiffness of the system increases.
- The volume in AC2 increases, the fluid entering the discharge chamber is lower than the volume leaving the suction chamber, and the stiffness of the system decreases.

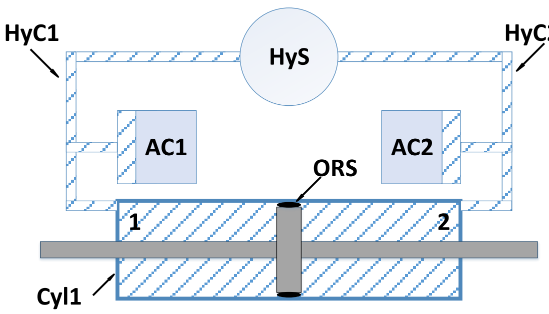


Figure 1. Biphasic media VSA schematic

To take advantage of the dynamic behavior of this architecture, the system was modeled based on the assumption of using hydropneumatic accumulators as nonlinear springs. The previous allows the user to accurately attain the desired position and stiffness of the system using a flow controller.

The biphasic VSA system can be analogically modeled as a damped harmonic oscillator as shown in (1), where m is the mass of the output link of the actuator, (piston, seals and mechanical attachments), x_p is the displacement of the piston, k_v is the coefficient of viscosity, F_p is the force generated by the difference of pressures P_1 and P_2 and F is the external force applied, which resembles the load on the actuator.

$$m\ddot{x}_p + k_v\dot{x}_p - F_p = F \tag{1}$$

The following assumptions must be considered for modeling this system:

- The gas is considered as ideal and there is no leakage from the hydropneumatic accumulators.
- There is no leakage between the cylinder chambers.
- The process is isothermal.

In the following equations the subscripts 1 and 2 denote chambers 1 and 2 of the hydraulic cylinder and their corresponding hydropneumatic accumulators.

$$F_p = (P_1 - P_2)S \tag{2}$$

$$P_i = \frac{C_i}{v_i} \tag{3}$$

where P_i is the pressure in the i^{th} chamber, S is the surface of the piston (same for both sides), C_i is the gas constant and v_i is the volume of compressible fluid in the i^{th} accumulator. The gas constant definitions are expressed in (4) and (5), where V_{ai} represents the added/subtracted volume of fluid volume in the system's chambers.

$$C_1 = P_1\left(v_1(0) - v_{a1} - Sx_p\right) \tag{4}$$

$$C_2 = P_2\left(v_2(0) - v_{a2} + Sx_p\right) \tag{5}$$

If the *gases* contained in both accumulators have the same pressure, the gas constant can be taken as equal for both sides ($C_1 = C_2 = C$), as shown in (6).

$$F_p = SC\left(\frac{1}{v_1} - \frac{1}{v_2}\right) \tag{6}$$

The *instantaneous* stiffness K of the actuator is the derivative of force generated by the difference of pressures in the cylinder with respect to displacement. In this case, the stiffness is a function of V_{ai} as shown in (7).

$$K = \frac{\partial F}{\partial x_p} = CS^2\left(\frac{1}{\left(v_1(0) - v_{a1} - Sx_p\right)^2} + \frac{1}{\left(v_2(0) - v_{a2} + Sx_p\right)^2}\right) \tag{7}$$

Based on the previous equations, the stiffness does not depend on m and k_v. If fluid is added to both chambers, the volume of gas reduces and the pressure in both chambers increases, increasing the stiffness. Similarly, if fluid is subtracted from both chambers, the volume of gas increases and the pressure in both chambers reduces, reducing the stiffness.

III. Position and Stiffness Control Strategy

The functional characteristics of the actuator depend on its design and the initial conditions. The measured quantities in the system are: (i) the position of the piston and (ii) the pressures in the chambers. Equations 8 and 9 describe the volumetric flow rate, where q_i is the represents the moving volume in the system's chambers ($\dot{v_{ai}} = q_i$). At any instant, the volume subtracted or added fluid cannot be larger than the total volume of gas since the liquid fraction is considered incompressible.

$$\dot{v_1} = -q_1 - S\dot{x_p} \tag{8}$$

$$\dot{v_2} = -q_2 + S\dot{x_p} \tag{9}$$

Considering u as a spring's restoring force equal to F_p and x_p^* as the desired position, the following was deducted:

$$u^* = -K_E(x_p - x_p^*) \tag{10}$$

Knowing that K_E is the characteristic gain or stiffness of the spring and $u*$ is the desired control signal. The error of position is stated next.

$$e = u - u^* \tag{11}$$

Differentiating (11) with respect to time leads to:

$$\dot{e} = \dot{F_p} - \dot{u^*} \tag{12}$$

From the combination of (2) and (12) it is possible to write:

$$\dot{e} = \frac{S}{C}\left[P_2^2\dot{v_2} - P_1^2\dot{v_1}\right] - \dot{u^*} \tag{13}$$

Substituting (8) and (9) into (13) leads to:

$$\dot{e} = \frac{S}{C}\left[P_1^2 q_1 - P_2^2 q_2\right] - w(t) \tag{14}$$

$$w(t) = \left[-\frac{S^2\dot{x_p}}{C}(P_2^2 + P_1^2) + \dot{u^*}\right] \tag{15}$$

$\dot{e}$ is described in equation (16), where γ is the tuning parameter to achieve minimal error.

$$\dot{e} = -\gamma e \tag{16}$$

From substituting (16) into (14), the equation that controls the position of the system is obtained (17).

$$\frac{C}{S}\left(-\gamma e + w(t)\right) = b_{x_p}(t) \tag{17}$$

$$b_{x_p}(t) = P_1^2 q_1 - P_2^2 q_2 \tag{18}$$

Based on equation (7), it is possible to confirm that the direct relation between the pressures and the stiffness of the systems allows controlling the stiffness. The pressure control equations were developed following the previous procedure used for position control. The error of pressure, given in equation (19), is defined as the difference between the current pressures of the system and related ones to desired stiffness (P_{ref}).

$$e_p = \left(\frac{P_1 + P_2}{2}\right) - P_{ref} \tag{19}$$

Differentiating (19) with respect to time and substituting (8) and (9) leads to:

$$\dot{e_p} = \left(\frac{1}{2C}\right)(P_1^2 q_1 + P_2^2 q_2) + w_p(t) \tag{20}$$

$$w_p(t) = \left(\frac{S\dot{x_p}}{2C}\right)(P_1^2 - P_2^2) - \dot{P_{ref}} \tag{21}$$

$\dot{e_p}$ is described in (20), where λ is the tuning parameter to achieve minimal error.

$$\dot{e_p} = -\lambda e_p \tag{22}$$

After substituting (20) into (22), the equation (20) is used to control the pressure.

$$b_p(t) = -2C\left(w_p(t) + (\lambda\, e_p)\right) \tag{23}$$

Solving (18) and (23) for q_1 and q_2 allows to control the position and stiffness of the system as shown in (24). Fig. 2 depicts the control block diagram implemented on the system.

$$\begin{bmatrix} q_1 \\ q_2 \end{bmatrix} = \begin{bmatrix} P_1^2 \\ -P_2^2 \end{bmatrix}\left(\frac{1}{P_1^4 + P_2^4}\right)\left(b_{x_p}(t)\right) + \begin{bmatrix} P_1^2 \\ P_2^2 \end{bmatrix}\left(\frac{1}{P_1^4 + P_2^4}\right)\left(b_p(t)\right) \tag{24}$$

The biphasic VSA can be considered to have antagonistic controlled stiffness [11], where two actuators with non-adaptable stiffness and nonlinear force-displacement characteristics are coupled antagonistically, working against each other; in this case, hydraulic proportional flow valves are used to the vary the pressures in both chambers.

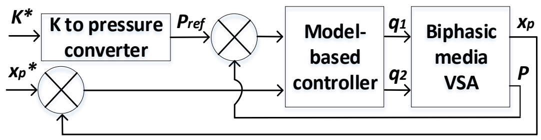


Figure 2. Control block diagram

IV. Simulations and Results

The following simulation was done to study the stiffness and position of the model using the parameters given in Table I. Two types of tests were performed: the first did not have a load force applied, while the second one had a force signal with a square wave shape with 75 N amplitude and baseline on 75 N at 0.5 Hz, as shown in Fig. 3. Position in the actuator is measured from left of the cylinder to end to right end.

TABLE I. SIMULATION PARAMETERS

Parameter	Value
m	1 kg
K_v	0.5 Ns/m
K_E	$8x10^3$ N/m
S	$3.776x10^{-4}$ m^2
C	300 Pa m^3
γ	$-1.3x10^2$
λ	$-1.2x10^4$
$x_p(0)$	0 m
x_pmax	0.15 m
x_pmin	0 m
$v_1(0) = v_2(0)$	$1.5x10^{-4}$ m^3

The desired position is given as a sinusoidal signal with 0.05 m amplitude and baseline on 0.075 m at 0.16 Hz, while the desired stiffness is a sinusoidal signal with 2.5x10^4 N/m amplitude and baseline on 5x10^4 N/m at 1.6 Hz. The results of the simulations prove that the model-based controller is able to properly track position and stiffness even if the desired signals to follow are nonlinear as illustrated in Fig. 4 and 5. It is worth to note that the controller was tuned to prioritize stiffness tracking over position tracking as displayed in Fig. 4.B and 5.B, whenever the load force differs from zero a displacement in the position of the actuator is generated. This displacement is directly related to the value of K_E.

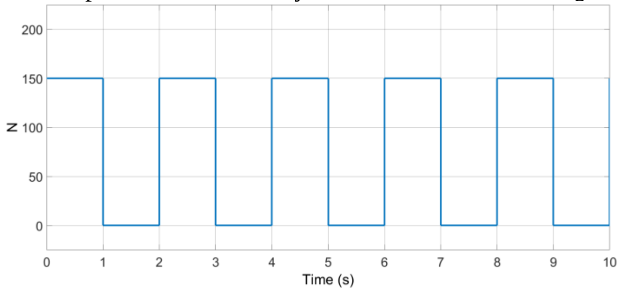
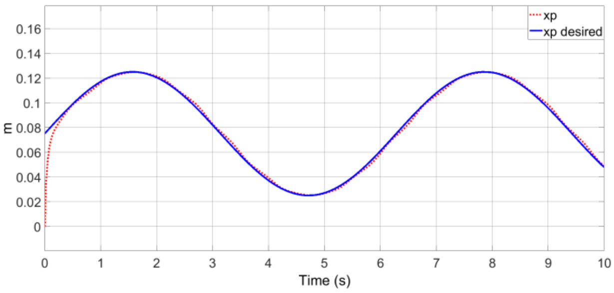


Figure 3. Load force signal

Fig. 6 displays the behavior of the volumes of gas inside the hydropneumatic accumulators. The volumes are rapidly decreased since the desired stiffness of the system is higher than the one provided by the original pressures of the gases, enabling to state that the volumes are constantly adjusted to follow the stiffness.



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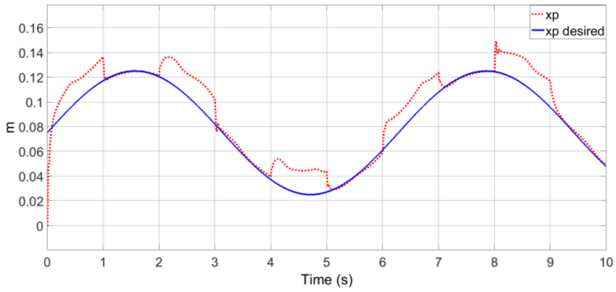
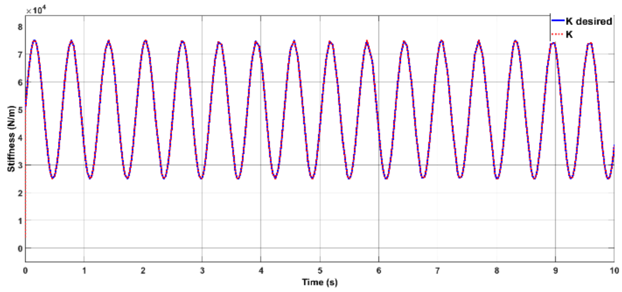


B

Figure 4. Position tracking in time. A: without load force. B: with load force applied.



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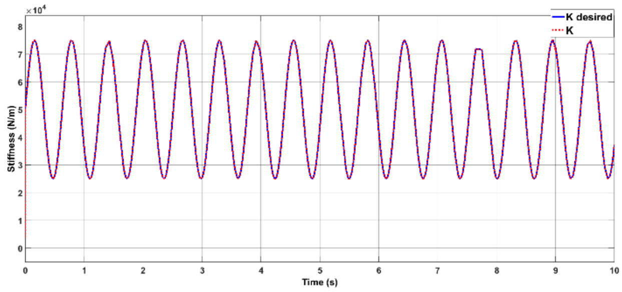
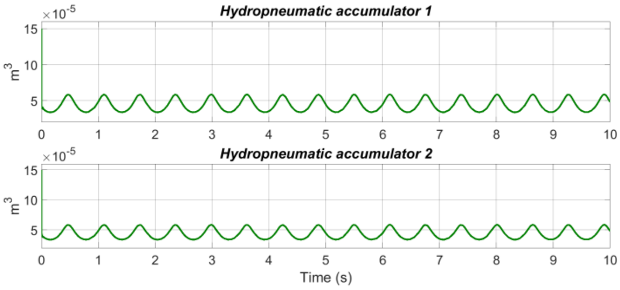


B

Figure 5. Stiffness tracking in time. A: without load force. B: load force applied.



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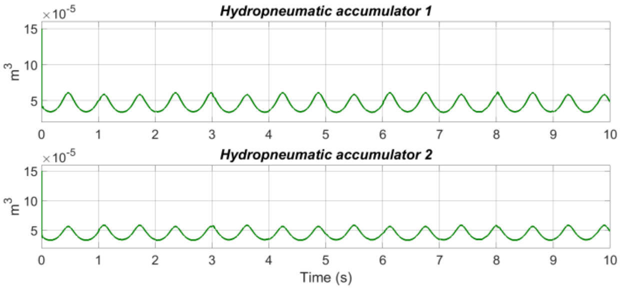

B

Figure 6. Variation in volumes inside the hydropneumatic accumulators in time. A: without load force. B: with load force applied.

V. Prototyping and Possible Applications

The system has been prototyped and is in testing phase. Fig. 7 displays the implemented hydraulic circuit, which is composed of a hydraulic cylinder (Cyl1), two hydropneumatic accumulators (AC1 and AC2), a hydraulic pump (HyS), two pressure gauges (PG1 and PG2) and four electro-proportional flow control valves (EPF1-4). To apply the previously described controller, a set of four EPF is used, two per chamber, allowing to manipulate q_1 and q_2. The position of the piston is measured via encoder and the stiffness is calculated in real-time using the pressure gauges. The design parameters in this prototype are given in Table 1. It is expected to generate more actuators with the same architecture but utilizing components in different scales to have a wider span of applications. The presented architecture can easily replace a standard hydraulic actuator, allowing it to be used in equipment using hydraulic cylinders where the adaptability in the stiffness provides practical advantage.

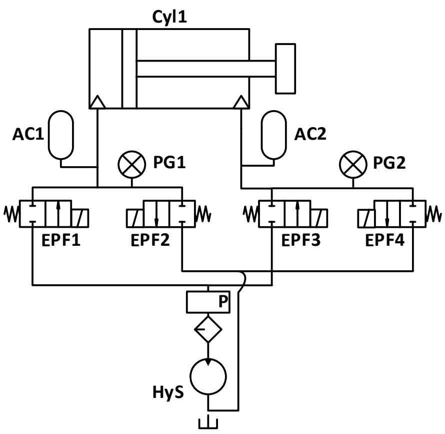

Figure 7. Hydraulic schematic of prototype

Some of the targeted applications are related to service and collaborative robots (cobot), providing the systems with the inherent safety that the VSA provides. In the case of service robots, the loading force is minimal (2-3 kg maximum), requiring a small hydraulic supply unit and low pressures contained in the accumulators. For the cobot, the loading force is related to the task to be performed, which requires higher pressure from the supply unit and inside the accumulators.

Utilization of this VSA in rehabilitation devices for upper limb has also been considered [16, 17]. A compliant machine would allow the patient to perform small controlled tasks with the arms, imitating pronation and supination motions.

VI. Conclusions

A VSA with inherent compliance can be designed using biphasic media as control fluids. The stiffness of the actuator is related to the pressures in the accumulators. The position and stiffness can be controlled using a logical pressure relation inside both hydraulic cylinder chambers. The hydraulic nature of the system gives it the possibility to have a high output force, while the nonlinearity of the gas contained in the accumulators provides a bigger range of stiffness than the VSAs using traditional elastic components.

The performed simulations of the system demonstrate that the model-based controller approach can accurately track desired position and stiffness, while an external force acts on it.

For future work, collision detection and reaction algorithms will be implemented in the system to contribute to ensuring safety to the human during physical interaction [18]. The detection will be carried out by an algorithm using the measurements of pressure gauges and encoder and comparing them to the desired values. The reaction algorithm is expected to be able to move the system into a safe state, reducing the stiffness and stopping the piston motion.

Conflict of Interest

The authors declare no conflict of interest.

Author Contributions

Jesus Lugo conducted the research, performed the simulations, and wrote this article; Giorgio Cannata developed the model-based control approach used to track position and stiffness; Matteo Zoppi modeled and manufactured the biphasic media variable stiffness actuator; Rezia Molfino supervised the project; all authors had approved the final version.

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