Proceedings of the 17th World Congress
The International Federation of Automatic Control
Seoul, Korea, July 6-11, 2008

Modelling and Identification for Control Design of Compliant Fluidic Actuators with Rotary Elastic Chambers: Hydraulic Case Study

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Abstract: This paper reports on the development of a comprehensive nonlinear mathematical model of a novel inherently compliant fluidic actuator with Rotary Elastic Chambers (REC–actuator). The actuator is intended for robotic devices working in direct physical contact with humans and can be operated by both, liquid as well as gaseous working fluid. In the hydraulic realization the REC-actuator is controlled by a custom developed bi-directional miniature pump. Applying physical modelling principles, a general model of the hydraulic REC-actuator is firstly developed. The actuator torque and volume characteristic, the hydraulic capacity, the volumetric displacement and the torque losses are defined as the main model components. Each of these components are unknown functions of the angular displacement and/or pressures in the actuator chambers which makes the modeling and identification procedure different as well as more complex than the one commonly performed for a "conventional" hydraulic actuator (a single vane motor in this case). For each of the mentioned characteristics a dedicated experiment is designed and carried out. Based on the collected data, analytical models of the characteristics are determined (including model structure and parameter estimation) and the final nonlinear model in the state space is obtained. The developed model suitable for control design is verified, showing good agreement between the simulation and experimental results.

1. INTRODUCTION

Compliant actuators with passive spring-like behaviour are becoming widely accepted solution for machines working in human dynamic environment, especially when physical human-machine interaction dictates that safety should have top priority (Bicchi *et al*., 2004). Among them, fluidic artificial muscle, mostly pneumatically actuated, is an inherently compliant actuator – so called soft actuator – characterized by high power/weight and power/volume ratios and thus well suitable for assistive robotics (Tsagarakis *et al*., 2003) and rehabilitation devices (Ferris *et al*., 2005). Vast majority of fluidic muscles are contractile devices meaning they provide linear motion only. To realize a revolute joint, an antagonistic set-up with mechanical transmission is used.

A rotary actuator with elastic chambers (REC – actuator) as a kind of fluidic muscle has been recently developed aiming at inherently compliant compact rotary joints intended for robots working in human environment, especially in assistive and rehabilitation tasks (Ivlev *et al*., 2006), (Kargov *et al*., 2007). Due to its design, the REC-actuator can be integrated in the revolute robotic joint directly, without any additional transmission elements, leading to more efficient and simpler robotic arm construction compared to those built by linear fluidic muscles. The REC-actuator can be operated either by oil or gas, owing to the material properties used for manufacturing the chambers. The hydraulic realization in comparison with the pneumatic one enables the design of fully modular encapsulated robotic joints with homogenous (only electrical) energy supply.

The REC-actuator as a plant has several specific properties, as complex torque and volume characteristics which are both dependent on pressure as well as angle, making modelling and control of the actuator not a common task. For the pneumatic realization of the REC-actuator where on-off valves are used to emulate pressure proportional valves, we applied the phenomenological modelling approach (Mihajlov *et al*., 2006). In this paper a detailed model of the hydraulic REC-actuator controlled by a bi-directional miniature pump is developed. Due to the above mentioned actuator specific properties, it was not possible to apply the procedure of modelling and identification of actuator pressure dynamics developed for "conventional" hydraulic actuators, as described for example in (Jelali *et al*., 2003). In the following, a procedure is proposed for modelling and identification of the hydraulic actuator with elastic chambers with almost no a-priori information on the actuator parameters, i.e. its characteristics.

2. SOFT HYDRAULIC REC-ACTUATOR

Working principle of the REC-actuator is analogous to common single vane fluid motor, which consists of two chambers, and a fixed and moving vane. The crucial difference is in the rotary elastic chambers that replace the rigid chambers of the conventional vane actuator. When pressurized, the chamber expands mainly in meridian direction and pressure force is transferred to torque by means of the moving vane. In hydraulic realization both chambers of the actuator are connected to a miniature bidirectional pump, forming a closed hydraulic circuit without oil tank (Fig. 1).

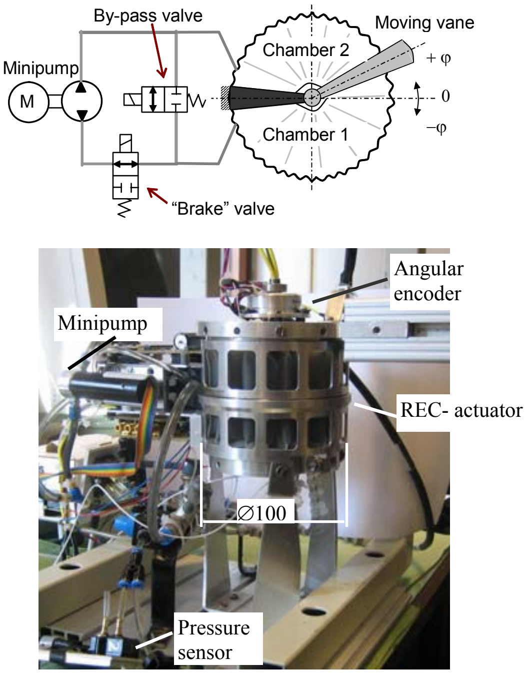

Fig. 1. Schematic diagram of the hydraulic REC-actuator and the experimental set-up.

The pump, developed by the Institute for Applied Computer Science in Forschungszentrum Karlsruhe, is of the fixed displacement external gear type and is driven by a brushless DC motor. The flow and pressure variation are realized by changing the motor speed while the change of motion direction can be achieved only by reversing the motor.

The only additional elements of the hydraulic circuit are solenoid on-off valves. The task of the valve connected in series with the pump is to stop the oil flow after the desired angular position has been reached – it is a kind of hydraulic brake. The valve connected parallel to the pump should provide instant pressure equalization in the chambers - it has bypass function in a situation of a sudden change of load. For control purposes a set of sensors are integrated in the actuator: absolute angular encoder, pressure sensors and a custom made torque sensor. For more information please see (Ivlev *et al.*, 2006)).

3. GENERAL SYSTEM MODELLING

General model of the hydraulic REC-actuator and a simplified model of the miniature pump are developed in this section. The torque and volume characteristics of the actuator play essential roles in this procedure and are firstly described.

3.1 Torque and volume characteristic

For the antagonistic set up of two elastic chambers, the total actuator torque is determined as difference of torques developed by each chamber, $M_{ACT} = M_1 - M_2$. To compute the actuator torque the analytical approximation of experimentally determined torque characteristic $M_1(p_1,\varphi)$ of one chamber can be used, see(Mihajlov *et al.*, 2006):

$$M_1 = f_1(p_1)\cdot\varphi^3 + f_2(p_1)\cdot\varphi^2 + f_3(p_1)\cdot\varphi + f_4(p_1) \qquad (1)$$

where the polynomial coefficients $f_i(p_1)$ are functions of pressure in the chamber. Assuming that the chambers are identical, the torque of the antagonist chamber is obtained by mapping the torque of the agonist symmetrically with respect to the ordinate, $M_2(p_2,\varphi) = M_1(p_1,-\varphi)$. In order to reduce errors due to analytical approximation, the torque of the hydraulic REC-actuator was determined in experiments with both actuator chambers. In the experiments, pressures in both chambers and the angular position were measured and the actuator torque was computed based on the measurements obtained by the force sensor mounted on the actuator lever. Note that for one value of input voltage u the pump produces approximately the same differential pressure in steady state (the difference of pressures in the actuator chambers) at all angular positions that belong to the considered angle area. For a given constant oil volume in the REC-actuator, a value of differential pressure $\Delta p = p_1 - p_2$ uniquely corresponds to one pair of pressure values in the chambers p_1 and p_2. Thus, it is possible to measure the torque characteristic of the hydraulic REC as a function of differential pressure and angular position, $M_{ACT}(\Delta p,\varphi)$. Figure 2 shows the measured torque characteristic when chamber 1 of the actuator is "active" (the oil is pumped from chamber 2 to chamber 1).

According to the third order polynomial approximation (1) of the torque developed by one elastic chamber, the total actuator torque M_{ACT} is to be modelled also by a third order polynomial. In order to get the first approximation however, the torque characteristic is approximated as:

$$M_{ACT}(\Delta p,\varphi) = (m_1\Delta p + m_2)\varphi + m_3\Delta p, \quad \Delta p = p_1 - p_2 \qquad (2)$$

where m_1, m_2, m_3 are constants. This torque approximation is also shown in Fig. 2. Note that the approximation error becomes significant at middle pressures (2-4 bar) for the positions $\varphi > |30°|$. The torque characteristic for the case that the chamber 2 is active is obtained by mapping (2) symmetrically with respect to the ordinate.

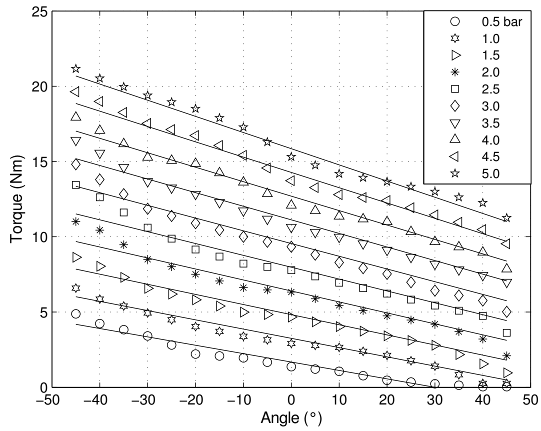

Fig. 2. Measured torque characteristic of the hydraulic REC-actuator (data points) and the polynomial approximation (line), the case when chamber 1 is active.

The volume characteristic was determined in a procedure with oil assuming that for the considered task the change of oil volume with small change in pressure is small enough to be neglected. The experiments were performed by measuring pressure for a constant amount of oil in the chamber for different angular positions realized by applying external torque. The overall characteristic $p(V,\varphi)$ was obtained performing the experiment for different values of oil volume in the chamber. Figure 3 shows the volume characteristic $V(p,\varphi)$ of one actuator chamber (chamber 1).

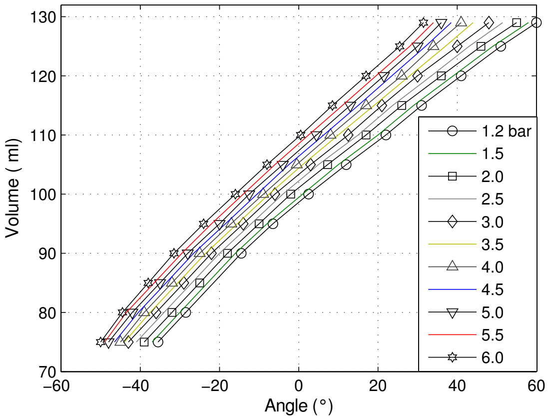

Fig. 3. Volume characteristic $V(p,\varphi)$ of chamber 1 of the REC-actuator.

The volume change at a constant pressure is a nonlinear function of angle that can be approximated with the second order polynomial. At φ=0° and absolute pressure of p=1.2 bar the chamber volume is 98 ml which is approximately one half of a volume of conventional vane actuator which corresponds in terms of dimensions to one rotary elastic chamber. For the same angle and at p=6.0 bar the volume is 110 ml, which means that the increase of volume due to increase in pressure is more than 10 %.

3.2 Dynamic model of the REC-actuator controlled by pump

Physical model of the hydraulic REC-actuator includes a model of pressure dynamics in the actuator chambers and a model of motion dynamics of the actuator moving parts.

The equation of the load motion arises by applying Newton's second law, as:

$$J\ddot{\varphi} = M_{ACT}(\Delta p,\varphi) - M_{loss}(\cdot) - M_{ext} \tag{3}$$

where J stands for total moment of inertia reduced to the axis of rotation, M_{loss} represents torque losses which will be defined in section 4, and M_{ext} includes gravitational forces as well as external disturbances.

The pressure dynamics in the actuator chambers of volume V is described using the flow continuity equation accounting for the fluid compressibility

$$\dot{V} + \frac{V}{E'}\dot{p} = \sum Q_{in} - \sum Q_{out} \tag{4}$$

where E' denotes the effective bulk modulus, Q_{in} and Q_{out} stands for incoming and outgoing volume flow rates, respectively. The flow continuity equation is defined assuming that (Watton, 1989) boundary deformation term ($\dot{V}$) describes volume change due to motion of moving parts of the actuator. The volume change due to pressure effects is to be included in the definition of effective bulk modulus and thus contained in the compressibility term only, $-(V/E')\dot{p}$.

Consequently, when applying (4) for modelling the pressure dynamics of the REC-actuator, the following important notes are to be made.

1) Although the volume of the rotary elastic chamber is a function of angular displacement and pressure in the chamber $V(p,\varphi)$, the term $\dot{V}$ in (4) takes into account volume change due to motion only. Thus

$$\frac{dV}{dt} \equiv \left(\frac{\partial V}{\partial \varphi}\right)\frac{d\varphi}{dt} \equiv C_v(\varphi)\dot{\varphi} \tag{5}$$

where $C_v(\varphi)$ describes change in volume with change in angle, at constant pressure. This characteristic stands for the volumetric displacement of the REC-actuator.

2) Overall compliance of the chambers (together with oil) is lumped in the compressibility term. The bulk modulus is defined as change in pressure divided by the fractional change in volume at constant temperature $E \equiv -V_0(\partial p/\partial V)|_{\theta}$. In the case of the hydraulic REC-actuator the main source of compliance is the torque dependence on angle as a consequence of the design of the elastic chambers. The air-pockets in the chambers (which are impossible to avoid) further influence the actuator compliance. In order to avoid accumulation of error when dividing total volume with the effective bulk modulus, the hydraulic capacity is introduced, as:

$$C_h \equiv -\left(\frac{\partial V}{\partial p}\right) = \frac{V(p,\varphi)}{E'(p,\varphi)} = \frac{V_0 + \Delta V(p,\varphi)}{E'(p,\varphi)} \tag{6}$$

3) The chambers are assumed to be identical. There is no oil leakage from the chambers.

Using (4) and taking into account the conclusions made above, the pressure dynamics for chambers 1 and 2 of the actuator can be written as

$$\dot{p}_1 = \frac{1}{C_{h1}(p_1,\varphi)}\left(Q - C_{v1}(\varphi)\dot{\varphi}\right), \tag{7}$$

$$\dot{p}_2 = \frac{1}{C_{h2}(p_2,\varphi)}\left(-Q - C_{v2}(\varphi)\dot{\varphi}\right), \tag{8}$$

where Q is the incoming volume flow rate.

The dynamic model of the pump can be obtained by using the same procedure as for the hydraulic motor. In this work however, a simplified empirical model of the miniature external gear pump is used. It consists of a linear first order differential equation for the pump dynamics and the pressure–flow characteristic, given respectively as:

$$\begin{aligned} T_p\dot{\omega}_p + \omega_p &= K_p u \\ Q_p &= f(\omega_p,\Delta p) \end{aligned} \tag{9}$$

where T_p is the time constant, K_p is the static gain, u the input voltage to the motor amplifier, and Q_p is the pump flow-pressure characteristic.

Having developed the models of the main system components, the general model of the hydraulic REC-actuator controlled by the bi-directional pump can be composed as shown in Fig. 4. The model consists of the pump subsystem (9), the pressure dynamics in the actuator chambers (7), (8) and the motion equation (3).

It is important to note that at this stage the model structure is still not completely known.

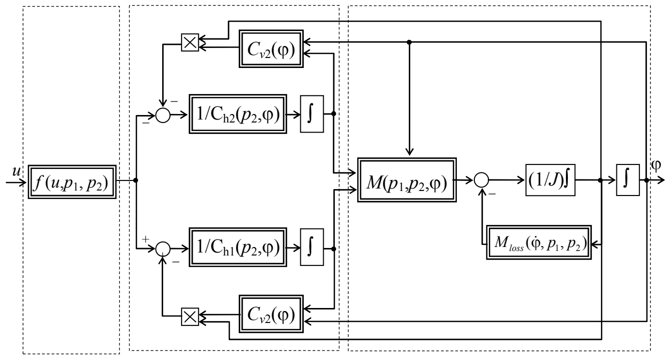

Fig.4. General model of the pump controlled hydraulic REC-actuator.

4. SYSTEM IDENTIFICATION

The REC-actuator and the miniature pump are newly developed components and thus all parameters of the model will be determined in the system identification experiments. The volume characteristic and the torque characteristic of REC-actuator were already explained in detail. The pump characteristic Q_p, the hydraulic capacity C_h, the volumetric displacement C_v and the torque losses M_{loss} are still to be considered and identified.

4.1 Pump flow-pressure characteristic

The flow characteristic of the pump $Q(u,\Delta p)$ was experimentally determined. To this end, the experimental set up was realized which enables pressure measurement on the intake and discharge side of the pump as well as the pump flow rate measurement (indirectly, by measuring the oil volume and time). The characteristic was measured for both pump directions. A constant voltage was applied to the motor amplifier and the effective flow rate was measured at different load pressures. The procedure was carried out for several voltages covering the whole range of the motor speed. Figure 5 shows the measurement results (data points). As seen, the pump characteristic is globally nonlinear - the flow rate decreases slower at higher voltages. For a constant input voltage the characteristic is however linear. Maximal flow rate at zero pressure difference is around 14 ml/s. The characteristic is symmetrical with respect to direction of rotation.

The measured flow characteristic is approximated by a polynomial:

$$Q(u,\Delta p) = (f_2\, u + f_3)\,\Delta p + f_1\, u + f_4; \quad f_1, f_2, f_3, f_4 = const. \tag{10}$$

This static nonlinearity is composed of the variable dead-zone and the variable saturation, whereby the differential pressure is a parameter for both nonlinearities. The value of the dead-zone is bigger at higher differential pressures while the maximal flow values decreases when the differential pressure increases. The time constant T_p which determines the pump dynamics could have not been determined exactly and its limits have been roughly estimated as $0.005 < T_p < 0.01$ s. For modelling purposes, the pump dynamics was neglected. As a consequence, the pump model (9) reduces to the characteristic $Q(u,\Delta p)$ described by (10).

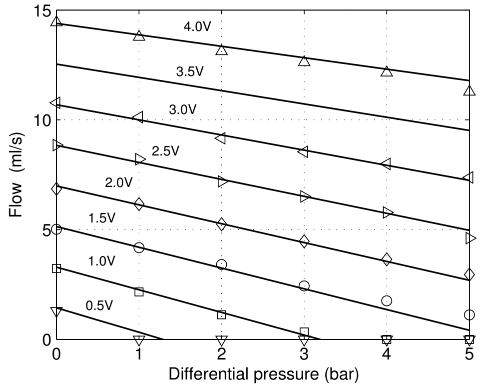

Fig. 5. Experimentally determined flow-pressure characteristic of the pump: the measured data (data points) and the polynomial approximation (solid line).

4.2 Hydraulic capacity

Hydraulic capacity stands for change in volume due to change in pressure. In the case when the actuator lever is clamped the pressure dynamics (7) becomes

$$\dot{p}_1 = (1/C_h(p_1))_{\varphi=const.} \cdot Q \tag{11}$$

As no a-priori knowledge of the hydraulic capacity was available (concerning the dependence with respect to pressure at least, for the clamped joint) it was not possible to perform common identification procedure based on (11). Instead, the dedicated experiment is performed. To this end, rewrite first (6) using finite differences for the case the lever is clamped:

$$C_h\big|_{\varphi=const} = -(\Delta V / \Delta p)\big|_{\varphi=const} \tag{12}$$

To determine C_h, the actuator lever is clamped at position φ=const. The oil was released from the chamber in steps of 1 ml and pressure was recorded. By using (12) the hydraulic capacity at one position is computed. The experiment is carried out for positions φ ∈[-40°, 40°] in steps of 10°. The measurement results are shown in Fig. 6.

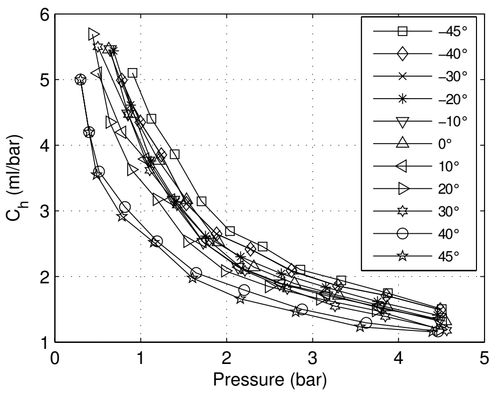

Fig. 6. Hydraulic capacity characteristic of one actuator chamber.

Based on the measurement results it is concluded that the hydraulic capacity decreases with pressure increasing meaning that to attain the same difference in pressure more oil volume is needed at smaller pressure values. This dependence is essentially nonlinear and has exponential character. The hydraulic capacity at same pressure is bigger when the chamber is contracted and decreases with increase of the expansion angle.

Analytical model of the experimentally determined C_h characteristic is obtained by fitting the data to a parametric model. In the first step fitting the data when pressure only is an independent variable is to be performed. The power series is found to be the most appropriate model. In the second step, the influence of angular position is modelled by scaling the power series in an exponential manner.

$$C_{h1} = e^{-\alpha}(h_1 p_1^{h_2} + h_3); \; \alpha = a_1\varphi + a_2; \quad (13)$$

where $h_1, h_2, h_3, a_1, a_2 = const$, p is pressure in the chamber and φ is angular position in degrees. For fitting the data, the Matlab function (*nlinfit*) which implements nonlinear least squares regression was used. The model (13) can be applied for the antagonistic chamber also - it is enough to map α symmetrically with respect to the ordinate.

4.3 Volumetric displacement

Volumetric displacement of a revolute hydraulic actuator is defined as the volume of fluid displaced per radian of rotation. Recall here that the boundary deformation term $\dot{V}$ in the flow continuity equation describes the volume change due to motion of the moving parts only. Thus, the displacement characteristic of the REC-actuator is defined as change in volume due to change in angular displacement, neglecting thereby the influence of pressure. A straightforward way to determine this characteristic is to use the polynomial approximation of the volume characteristic (see Sec 3.1). In doing so, the actual volumetric displacement C_{vp} was firstly determined:

$$C_{vp}(\varphi, p) = \frac{\partial(v_{1i}\varphi^2 + v_{2i}\varphi + v_{3i})\big|_{pi=const}}{\partial\varphi} = v_{1i}'\varphi + v_{2i}'\big|_{pi=const} \quad (14)$$

where v_{1i}, v_{2i}, v_{3i} are polynomial coefficients at pressures p_i =1, 2, … 6 bar. Note that the pressure dependence is still present in C_{vp}. As the boundary deformation does not take into account pressure dependence, real displacement characteristic consisting of several linear functions is reduced to one function at middle pressure. The displacement model (14) can be used for the antagonistic chamber also. It is enough to observe that for the volume characteristic of the chamber 2 holds that $V_2(\varphi)=V_1(-\varphi)$. Note that the displacement characteristic of the REC-actuator, even when independent of pressure, is a linear function of angular position unlike volumetric displacement of the common revolute actuator which is a constant.

4.4 Torque losses

The rotary elastic chambers capsulate oil volumes and thus no seals are needed. This eliminates the main source of static friction as well as the stick-slip effect as common problems of conventional hydraulic actuators. Current realization of the REC-actuator however has other sources of torque losses. The pressure forces in the actuator act not in meridian direction, but also in radial directions. Due to this and the fact that the chambers rotate, there is friction between the parts in relative motion: the fasteners (that connect the chambers with the axis) and the bushing and also between the chambers and the actuator housing.

In order to determine the torque losses due to friction, a common approach to investigate friction as a function of velocity is used. In experiments performed at constant velocities the torque losses can be determined using the equation of motion (3) as

$$M_{loss}\big|_{\dot{\varphi}=const} = M_{ACT}(\Delta p, \varphi) \quad (15)$$

For the considered system, a condition for motion at approximately constant velocity (in open loop) is the constant flow rate of the pump. To compute the input voltages that correspond to constant flow rates, the pump flow characteristic (10) has been inverted for the flow rates Q_d = 1,2,…14 ml/s. When applying this voltage function, after a transient period a steady state velocity is reached, but only for a certain time interval. For measured pressure in the chambers and angle in this time interval, the torque losses being equal to the actuator torque can be obtained by using the torque look-up table or computed using the analytical approximation (2). The latter solution was used and the average values of torque losses computed during the time intervals of constant velocities are shown as data points in Fig. 7. Assuming that the actuator is symmetrical, the experiments were performed only for positive velocities and the determined values of torque losses are then used for negative velocities.

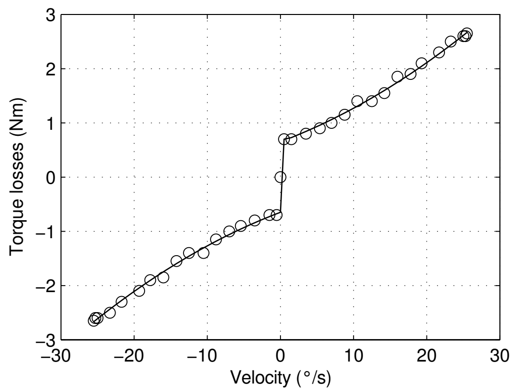

Fig. 7 Experimentally determined torque losses versus the actuator velocity.

The experimentally determined torque losses are approximated as:

$$M_{losses} = \begin{cases} l_1 \cdot \dot{\varphi}^2 + l_2 \cdot \dot{\varphi} + l_3, & \dot{\varphi} \geq a \\ l_4 \cdot \dot{\varphi}, & |\dot{\varphi}| < a \\ -l_1 \cdot \dot{\varphi}^2 + l_2 \cdot \dot{\varphi} - l_3, & \dot{\varphi} \leq -a \end{cases} \quad (16)$$

By increasing the load and keeping the same velocity the torque losses could be determined for different pressure conditions. It is reasonable to expect bigger values of torque losses at higher pressures.

By using the general model of the REC-actuator (3),(7), (8),(9) and the analytical descriptions of the model parameters (10), (13), (14) (16), by defining a state vector as $\mathbf{x} = [\varphi \;\; \dot{\varphi} \;\; p_1 \;\; p_2]^T$ one arrives at the following set of nonlinear state space equations for the dynamics of the hydraulic REC-actuator (final model):

For $u>0$

$$
\begin{aligned}
\dot{x}_1 &= x_2 \\
\dot{x}_2 &= \frac{1}{J}\left(-m_1x_1x_4 + m_1x_1x_3 - m_2x_1 + m_3x_4 - m_3x_3 - M_{loss}\right) \\
\dot{x}_3 &= \frac{1}{h_1x_3^{h_2} + h_3}\left(-f_3(x_4 - x_3) - f_4 - (v_1x_1 + v_2)x_2\right) \\
&\quad - \frac{1}{h_1x_3^{h_2} + h_3}\left(f_1 + f_2(x_4 - x_3)\right)u \\
\dot{x}_4 &= \frac{1}{h_1x_4^{h_2} + h_3}\left(f_3(x_4 + x_4) + f_4 - (v_1x_1 - v_2)x_2\right) \\
&\quad + \frac{1}{h_1x_4^{h_2} + h_3}\left(f_1 + f_2(x_4 - x_3)\right)u
\end{aligned}
\tag{17}
$$

For the case that the input signal is $u<0$, the model is obtained in an analogue way.

5. MODEL VALIDATION

The dynamic model of the hydraulic REC-actuator controlled by the pump was built in the Matlab/Simulink. A series of experiments were conducted to compare the simulation results with the real system dynamics. All experiments were performed in open loop, for the cases that the actuator operates without the influence of the gravitational force (Fig. 8a), and under the influence of the gravitational force (Fig. 8b). The model performance was tested for the whole angle operating area under the nominal load ($J = 0.085$ kg·m^2). The model performance for a square type input signal is presented in these figures. After changing the direction of input signal, the modelled pressure in the high pressure chamber start to deviate from the measured value. The pressure oscillations when changing the direction of input signal cannot be observed in the model. Still, the results present reasonable agreement between the measured and simulated data.

6. CONCLUSIONS

A comprehensive mathematical model of a novel inherently compliant hydraulic REC-actuator has been developed and system identification has been subsequently carried out. In case of the here considered soft actuator, due to the elastic chambers, it has not been possible to apply the common procedure of modelling and identification of pressure dynamics in the actuator chambers. Instead, the model of pressure dynamics has been developed combining first principles and suitably designed experiments for each of the model components, whereby some of them are functions of pressure and/or angle, unlike the "conventional" case where the parameters are constants. Furthermore, since no a priori knowledge on the system components was available, all subsystems were modelled and identified. The general model of hydraulic soft actuators was developed and the final model of the REC-actuator has been obtained in a form suitable for the control design. Hereby applying of the nonlinear model-based control strategies appropriate for "conventional" fluid actuators is intended. The considered inherently compliant REC-actuator is going to be used in the rehabilitation robotic devices.

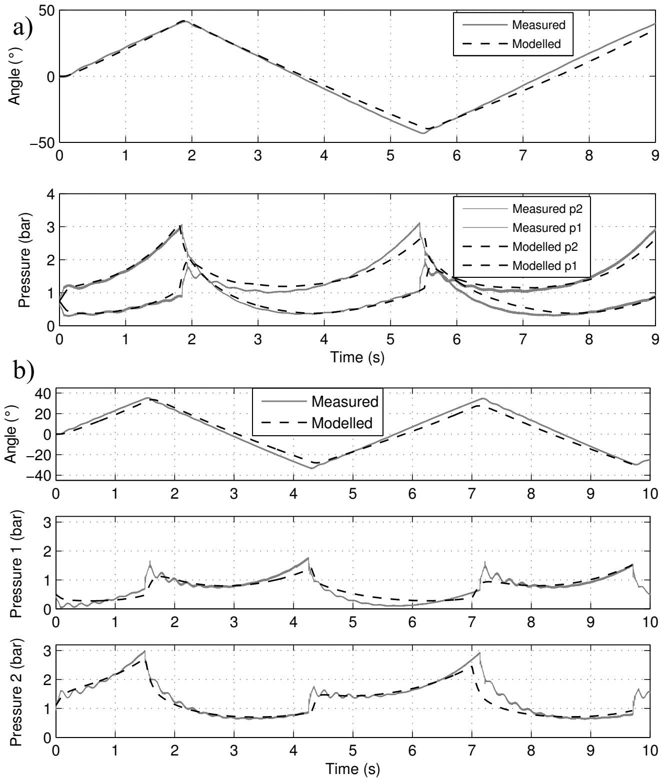

Fig. 8 Comparison between measured and modelled results for a square wave input signal $|u(t)|=0.9|u_{max}|$, the actuator in horizontal plane (a) and in vertical plane (b).

ACKNOWLEDGEMENT

This work was partially supported by the AiF - the German Federation of Industrial Research Associations "Otto von Guericke", (Grant ZN113) and by the BMBF- the German Federal Ministry of Education and Research (Grant 16SV2290 and Grant 01EZ0769). The work done by Mr. Tadas Sperauskas in performing the numerous experiments is gladly acknowledged.

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