Flatness Based Control of a Rotary Vane Actuator

Markus Bröcker†, Frank Heidtmann★

†TR W Automotive, Düsseldorf Technical Center, Hansaallee 190, D-40547 Düsseldorf, +49-(0)211-584 648
★University Duisburg-Essen, Institute of Mechatronics and System Dynamics, Chair of Dynamics and Control, D-47048 Duisburg, +49-(0)203-379 1587
e-mail: Markus.Broecker@trw.com, heidtmann@uni-duisburg.de

Abstract — For robots that are used to operate in wide ranges or to carry heavy loads, hydraulically driven actuators have to deal with elastic deformations of the links and with desired trajectories for the tool center point of such a robot. Rotary vane actuators can be integrated alternatively to the so far usual differential cylinders at the arms of large manipulators. In this paper a nonlinear dynamic model for two one behind the other chained rotary vane actuators and a flatness based controller are derived for such a driven robot arm. The flatness based approach as a nonlinear option turns out to be a good controller alternative to deal with complex tasks of spatial movements – especially with rotary vane angle tracking tasks. The experimental results demonstrate the importance of a suitable nonlinear system modeling for the controller design and good tracking performance for the flatness based control of each rotary vane actuator.

Keywords — modeling, flatness based control, rotary vane actuator

I. INTRODUCTION

For robots that are used to operate in wide ranges or to carry heavy loads, hydraulically driven actuators have to deal with elastic deformations of the links and with desired trajectories for the tool center point of such a robot. For this kind of manipulators (e.g. a concrete pump), hydrostatic differential cylinders are often applied for actuation due to their simple construction and their high power density. State of the art is that such differential cylinders realize a translational movement which is converted by a transmission kinematics element into a rotational movement of the manipulator arm. For flexible manipulators fundamental investigations took place in [1]. A vibration control concept was studied for a differential cylinder of a flexible robot in [2]. Improvements for the cylinder piston tracking of differential cylinders were achieved by nonlinear control strategies, e.g. the flatness based control in [3] and the disturbance attenuation control in [4], [5].

Rotary vane actuators can be integrated alternatively to the so far usual differential cylinders at the arms of large manipulators. By these new constructions of manipulator arms, it is possible to eliminate the disadvantageous transmission kinematics: Rotary vane actuators as rotational drives provide rotational movements directly because they are constructed as a joint and actuator in one. This aim makes high demands against the drive, so that a precise concept and a high-quality trajectory tracking of the rotary vane angle is indispensable. However, the use of rotary vane actuators is associated with high internal oil leakage and/or high friction. Therefore, the nonlinear modeling of rotary vane actuators as well as their nonlinear control here contain the exact modeling of leakage and friction and represent one approach to deal with the high-quality trajectory tracking of the rotary vane angle. The modeling of a rotary vane actuator takes place similarly to synchronizing cylinders, since both are symmetric drives, i.e. the same dynamic behavior can be achieved with right and left movements. First beginnings to model investigations of a rotary vane actuator and to linear control of a rigid robot, which is driven by such an actuator, are described in [6].

This paper presents for the first time a nonlinear dynamic model for two one behind the other chained rotary vane actuators with coupled arm and payload

Fig. 1. Robot: Two one behind the other chained rotary vane actuators with coupled arm and payload
rotary vane actuator and particular characteristics for either of them.

II. MODELING OF HYDRAULIC ROTARY VANE ACTUATORS

For the flatness based control as a model based control concept a physical model and, adapted from this one, a mathematical model of the robot have to be derived. Therefore, both actuators, the coupled robot arm as well as parts of the hydraulic system (figure 2) are taken into consideration.

![Hydraulic circuit diagram](image)

Fig. 2. Hydraulic circuit diagram of the test bed. 1.0: Actuator 1, without vane seals, vertical rotation axis; 1.1, 2.1: Sandwich plate with connection threads; 1.2, 2.2: Relief valve; 1.3, 2.3: Counterpressure valve with check valve; 0.1: 4 ball valves; 0.2: Check valve; 1.4, 2.5: 4/3 Proportional directional valve; 1.5, 2.6: Pressure reducing valve; 2.0: Actuator 2 with vane seals, horizontal rotation axis; 2.4: Counterpressure valve with check valve; 0.1: 4 ball valves; 0.2: Check valve

For model simplifications, the following assumptions are made:

- No temperature influences, especially not on friction and leakage.
- The hydraulic unit is an ideal pressure source, the supply pressures \( p_{0.1}, p_{0.2} \), the return line pressure \( p_T \) and the bulk modulus \( E_b \) are constant, the line dynamic is negligible.
- For both rotary actuators, it is imperative that \( p_T \leq p_A \leq p_0 \) and \( p_T \leq p_B \leq p_0 \). This condition has always been fulfilled.
- The friction torques and the internal oil leakages of the rotary vane actuators can be described by time invariant characteristic curves. Other leakages and influences of friction are negligible.
- The proportional directional valves can be considered as type 0 systems. Relief valves, counterbalance valves and counter pressure valves need not to be taken into account.

At first, just one of the rotary vane actuators will be considered. Based on the points above and the hydraulic circuit diagram (figure 2) the plant model in figure 3 has been built. It shows the physical model of the actuator with its internal oil leakages \( Q_{L_1 A} + Q_{L_1 B} = Q_{L_1} \), the friction torque \( M_F \) and the outer actuator load torque \( M_L \). Furthermore the medial radius \( m_r \), the pressure effective area \( A_v \) of one vane side and the rotation angle \( \psi \) are drawn in. The plant model also shows the associated valves of the rotary actuator: 4/3 Proportional directional valve, check valve and pressure reducing valve.

Using the states

\[
\begin{align*}
    x_1 &= \psi \\
    x_2 &= \dot{\psi} \\
    \Delta p &= p_A - p_B
\end{align*}
\]

a nonlinear mathematical model for this system is set up as in [7]:

\[
\begin{align*}
    \dot{x}_1 &= x_2 \\
    \dot{x}_2 &= \frac{1}{\nu_{ave}} (A_v L_r m \Delta p - M_F(x_2) - M_L) \\
    \dot{\Delta p} &= \frac{-E_{Fl}}{V} (2A_v L_r m x_2 + 2Q_{L_1}(\Delta p)) \\
    &+ \frac{\sqrt{2E_{Fl}B_v K_v U}}{V} \sqrt{p_0 - p_T - \Delta p}, U \geq 0 \\
    &- \frac{\sqrt{2E_{Fl}B_v K_v U}}{V} \sqrt{p_0 - p_T + \Delta p}, U < 0 \\
    y &= x_1
\end{align*}
\]

For this the normalized valve control voltage \( U = u/u_{max} (-1 \leq U \leq 1) \) is the input of the system, the rotation angle \( \psi \) is the output. The overall mass moment of inertia

\[
J_{ove} = J_{act} + J_{oil} + p_{Fl}(V_{LA} + V_{LB}) r_m^2
\]

includes the inertia of the rotating actuator parts and the inertia of the oil inside the actuator and both operating line
oil volumes \( V_{LA}, V_{LB} \). The average volume \( V \) is calculated according to \( V = \frac{1}{2}(V_{LA} + V_{LB} + \Delta V_{La m}R) \), \( \Delta V \) is a flow coefficient (c.f. [8]) and \( K_v \) is the valve amplification factor.

With a signal of the moment \( M_L \), it is possible to derive a \( D \)-flatness-based control from the model above as in [7]. Here, without such a signal, it is necessary to extend the model with regard to the robot parts coupled with the respective actuator. Apart from the hydraulic components, the system is thereto regarded as a multi-body system (MBS). Accordingly figure 4 shows a sketch of the whole robot. In addition to both rotary actuators, the centers of mass of the robot arm (mass \( m_{arm} \)), the payload (mass \( m_{hal} \)) and the holder profile (mass \( m_{hal} \)), which links the robot arm to the second rotary actuator, are drawn in. Furthermore the corresponding overall center of mass with the mass \( m_{hal} \) is shown.

![Fig. 4. Actuator coordinate systems, centers of mass of holder profile, arm and payload](image)

The equations of motion of this system are derived using Lagrange’s equations of the second kind:

\[
\frac{d}{dt} \left( \frac{\partial L}{\partial \dot{q}_k} \right) - \frac{\partial L}{\partial q_k} = Q_k \quad , k = 1, \ldots, f . \tag{4}
\]

This is true for a system of \( m \) particles with \( f \) degrees of freedom where the \( f \) generalized coordinates \( q_k \) clearly describe the positions of the \( m \) particles. The Lagrangian \( L = T - V \) is defined as the difference between the system’s kinetic and potential energy, so it contains the potential of the conservative forces. Against that, the generalized forces \( Q_k \) need not necessarily be derived from a potential, they can be calculated after \( Q_k = \sum_{j=1}^{m} F_{ij} \frac{\partial \Gamma_j}{\partial \dot{q}_k} \). Here, \( F_{ij} \) is the effective, resulting force acting on the \( j \)-th particle positioned at \( r \) in space. In the present case, \( F_{ij} \) is non-conservative only.

Concerning the robot, its two rotation angles \( \psi_1 \) and \( \psi_2 \) can be used as the generalized coordinates of the system. By means of these angles and through the reduction of the robot’s mass moments of inertia with regard to the two actuator axis, the robot’s kinetic energy can be calculated according to

\[
T = T_1 + T_2 = \frac{1}{2}J_1(\psi_2)\dot{\psi}_1^2 + \frac{1}{2}J_2\dot{\psi}_2^2 . \tag{5}
\]

It’s potential energy is

\[
V = m_{hal}gco\cos \left( \psi_2 + \frac{\pi - R}{2} \right) . \tag{6}
\]

The generalized forces here correspond to the torques

\[
\begin{align*}
Q_1 &= A_V L_{m} \Delta p_1 - M_{F,1} \left( \dot{\psi}_1 \right) \quad \text{and} \\
Q_2 &= A_V L_{m} \Delta p_2 - M_{F,2} \left( \dot{\psi}_2 \right) .
\end{align*} \tag{7}
\]

After all, the angular accelerations \( \ddot{\psi}_1 \) and \( \ddot{\psi}_2 \) can be calculated by inserting eq. (5), eq. (6) and eq. (7) in eq. (4). The obtained equations, together with similar equations to eq. (2) (pressure build-up \( \Delta p \)), form the state space model of the robot:

\[
\begin{align*}
x_1 &= \psi_1, x_2 = \dot{\psi}_1, x_3 = \Delta p_1, x_4 = \psi_2, x_5 = \dot{\psi}_2, x_6 = \Delta p_2 \\
\dot{x}_1 &= x_2 \\
\dot{x}_2 &= \frac{1}{J_1(\psi_2)} \left( x_2 x_5 \frac{dJ_1(\psi_2)}{dx_4} - A_V L_{m} x_3 + M_{F,1}(x_2) \right) \\
\dot{x}_3 &= \frac{1}{J_1(\psi_2)} \left( 2A_V L_{m} x_3 + 2Q_{1,1,1}(x_3) \right) \\
&+ \sqrt{E_{\psi_{hal}}K_{\psi_{hal}}} \left( \sqrt{P_{01} - P_{T} - x_3}, U_1 \geq 0 \right) \\
\dot{x}_4 &= x_5 \\
\dot{x}_5 &= \frac{1}{J_1(\psi_2)} \left( x_5^2 \frac{dJ_1(\psi_2)}{dx_4} + m_{hal} g \cos \left( \psi_2 + \frac{\pi - R}{2} \right) \right) \\
&+ A_V L_{m} x_6 - M_{F,2}(x_5) \\
\dot{x}_6 &= \frac{1}{J_2(\psi_1)} \left( 2A_V L_{m} x_5 + 2Q_{1,2,2}(x_5) \right) \\
&+ \sqrt{E_{\psi_{hal}}K_{\psi_{hal}}} \left( \sqrt{P_{02} - P_{T} - x_6}, U_2 \geq 0 \right) \\
y_1 &= x_1 \\
y_2 &= x_4 \\
\end{align*}
\]

with the angle dependent mass moment of inertia

\[
J_1(\psi_2) = \left( I_r^{c_{\psi_{hal}}} + I_{r_{\psi_{hal}}} + I_{r_{\psi_{hal}}} \right) \cos^2 \left( \psi_2 + \frac{\pi - R}{2} \right) \\
+ \left( I_r^{c_{\psi_{hal}}} + I_{r_{\psi_{hal}}} + I_{r_{\psi_{hal}}} \right) \sin^2 \left( \frac{\pi - \psi_2}{2} \right) \\
+ \left( I_{r_{\psi_{hal}}} + \sin \left( \psi_2 + \frac{\pi - R}{2} \right) r_{c_{\psi_{hal}}} \right)^2 m_{hal} \\
+ \left( I_{r_{\psi_{hal}}} + \sin \left( \psi_2 + \frac{\pi - R}{2} \right) r_{c_{\psi_{hal}}} \right)^2 m_{hal} \\
+ J_{act,1} + J_{act,2} + m_{oil} \left( \frac{\pi - \psi_2}{2} \right) . \tag{9}
\]

The principal mass moments of inertia \( I_{r_{\psi_{hal}}} \), \( I_{r_{\psi_{hal}}} \), \( I_{r_{\psi_{hal}}} \) and \( I_{r_{\psi_{hal}}} \), \( I_{r_{\psi_{hal}}} \), \( I_{r_{\psi_{hal}}} \) of holder profile, robot arm and of payload relate to the respective center of mass at a (body-fixed) coordinate system orientation as shown in figure 4 for actuator 2. The parallel axis theorem determines the rows three to five of eq. (9). \( J_{act,1} \) corresponds to eq. (3) and is specified for actuator 1 here, \( J_{act,2} \) are the reduced mass moments of inertia regarding actuator axis 1 of actuator mounting (connects both actuators, see figure 4) and of actuator 2. Finally, the moment of inertia of the oil mass \( m_{oil} \) (same for both actuators) of actuator 2 is added.

The characteristic curves of the friction torques \( M_{F,1}(\psi_1) \), \( M_{F,2}(\psi_2) \) and the internal oil leakages...
$Q_{\Lambda 1}(\Delta p_1)$, $Q_{\Lambda 12}(\Delta p_2)$ are experimentally determined. The curve progressions of the friction curves characterize them as so called Stribeck-Curves. Such curves can be described by

$$M_F(\dot{\psi}) = M_{OH} + f_V \dot{\psi} + \text{sign}(\dot{\psi})(MC + M_H \exp \left(\frac{-|\dot{\psi}|}{c_{11}}\right)),$$  \hspace{1cm} (10)

Against that, the leakage curves are nearly linear. But to be more precisely, especially in the beginning and in the end of the curves, they are approximated by polynomials of sixth degree resp. of third degree.

III. MATHEMATICAL FOUNDATIONS OF FLATNESS BASED CONTROL

Differential algebra was introduced by [9] and found its way into control theory in the 1980s [10]. Classic algebra has been introduced to handle (algebraic) equations of variables with numerical coefficients. Analogously, differential algebra was introduced to handle differential equations of variables with coefficients which may be meromorphic functions in a complex region $\mathbb{C}^m$. Hence, the close connection between algebra and differential algebra is as follows: Algebraic equations may be considered as differential equations in which (time) derivatives of variables do not occur, i.e. corresponding coefficients are equal to zero. In order to introduce the definitions of flatness and flatness based control, several basic algebraic differential algebraic definitions are specified in the following.

Let $K/k$ be a (differential) field extension and $L \subset K$ such that all elements of $L$ are $k$-(differential)-algebraically independent. The maximum number of such elements in $L$ is called the (differential) transcendence degree of $K/k$ (differential transcendence degree of $K/k$) and $L$ is called a transcendence basis of $K/k$ [11].

Let $k$ be a differential field. Then, according to [12] a system is a finitely generated differential field extension $K/k$.

Let $u$ be the input of a dynamic system. Then, the dynamics of a system $L/k(u)$ is a finitely generated differential field extension $L/k(u)$ [12].

Hence, diff. $\text{trg}L/k(u) = 0$ for all (finitely generated) dynamics holds (c.f. [12]). Outputs of the corresponding system are finite many elements of the dynamics $y_1, \ldots, y_p \in L$. Thus, there exists a set $\{x_1, \ldots, x_n\} \subset L$, which is a (non-differential) transcendence basis of the dynamics $L/K(u)$. The elements of this transcendence basis of the dynamics are the states of a system.

Two systems $D/k$ and $\bar{D}/k$ are called equivalent or equivalent by endogenous feedback iff any element of $D$ (or $\bar{D}$ resp.) is algebraic over $\bar{D}$ (or $D$ resp.). Two dynamics $D/k(u)$ and $\bar{D}/k(u)$ are said to be equivalent iff the corresponding systems, $D/k$ and $\bar{D}/k$, are so [13].

A system $D/k$ is called (differentially) flat iff it is equivalent to a purely differential transcendental system $L/k$. A differential transcendence basis $\mathbf{y} = [y_1 \ldots y_p]^T$ of $L/k$ with the property $L = \langle \mathbf{y} \rangle$ is called the linearizing or flat output of the system $D/k$ [13].

Roughly speaking, the definition means that all states and inputs of a nonlinear system can be computed in dependence on the linearizing or flat output $\mathbf{y}_{lin} = h(x, u, \ldots, u^{(m)})$ without any integration. The state vector $x$ and the system input vector $u$ can be expressed by the (local) conditions [14]

$$x = f_1(y_1, y_2, \ldots, y^{(3)}) \hspace{1cm} (11)$$

$$\mathbf{u} = f_2(y_1, y_2, \ldots, y^{(3+1)}) \hspace{1cm} (11)$$

and

$$\dim \mathbf{u} = \dim \mathbf{y} .$$  \hspace{1cm} (12)

The expression for $\mathbf{u}$ in eq. (11) is then defined as the flatness based control law and the flat output vector $\mathbf{y}$ and its time derivatives have to be substituted by the reference trajectory $y := y_{rt}$ to achieve tracking control.

IV. FLATNESS BASED CONTROLLER DESIGN

With the output $\mathbf{y} = (y_1, y_2)^T = (\psi_1, \psi_2)^T$ of the robot model (eq. (8)), representations according to eq. (11) are possible:

$$\begin{align*}
x_1 &= y_1 \\
x_2 &= \dot{y}_1 \\
x_3 &= J_1(y_2)\dot{y}_1 + T_1 + 2M_{F_1}y_1 + M_{F_1}(y_1) \\
x_4 &= y_2 \\
x_5 &= \dot{y}_2 \\
x_6 &= J_2\dot{y}_2 - \frac{d_1(y_2)}{2} \dot{y}_2 - m_{hal}g_{hal} \sin(y_2 + \frac{\pi}{2}) + M_{F_2}(y_2)
\end{align*}$$

satisfies the first row, and

$$\begin{align*}
U_1 &= \frac{u_1}{u_1^2}, \\
U_2 &= \frac{u_2}{u_2^2}, \hspace{1cm} \text{with} \hspace{1cm} (14)
\end{align*}$$

$$\begin{align*}
U_1^N &= J_1(y_2)V \ddot{y}_1 + \frac{d_2(y_2)}{2}V \dddot{y}_1 + 2M_{F_1}(y_1) + 2A_2^2L^2r^2E_s \dddot{y}_1 + 2A_2LrE_s \dddot{y}_1 \\
&+ 2A_2LrE_s \dddot{y}_1 + 2A_2LrE_s \dddot{y}_1 + 2A_2LrE_s \dddot{y}_1 + 2A_2LrE_s \dddot{y}_1 + 2A_2LrE_s \dddot{y}_1 \hspace{1cm} (13)
\end{align*}$$

$$\begin{align*}
U_2^N &= J_2V \dddot{y}_2 - \frac{d_1(y_2)}{2}V \dddot{y}_2 + \frac{d_2(y_2)}{2}V \dddot{y}_2 + 2M_{F_2}(y_2) + 2A_2L^2r^2E_s \dddot{y}_1 \\
&- \frac{m_{hal}g_{hal}}{A_2Lr} \cos(y_2 + \frac{\pi}{2})V \dddot{y}_2 + 2A_2LrE_s \dddot{y}_1 + 2A_2LrE_s \dddot{y}_1 + 2A_2LrE_s \dddot{y}_1 \hspace{1cm} \text{with} \hspace{1cm} (14)
\end{align*}$$

$$\begin{align*}
U_1^D &= \sqrt{2A_2LrE_s \dddot{y}_1} \sqrt{\langle J_1(y_2)\dot{y}_1 + T_1 + 2M_{F_1}(y_1) \rangle} \\
U_2^D &= \sqrt{2A_2LrE_s \dddot{y}_1} \sqrt{\langle J_2\dot{y}_2 - \frac{d_1(y_2)}{2} \dot{y}_2 - m_{hal}g_{hal} \sin(y_2 + \frac{\pi}{2}) + M_{F_2}(y_2) \rangle} \hspace{1cm} \text{with} \hspace{1cm} (14)
\end{align*}$$

$$\begin{align*}
U_1^N &= \frac{u_1}{u_1^2}, \\
U_2^N &= \frac{u_2}{u_2^2}, \hspace{1cm} \text{with} \hspace{1cm} (14)
\end{align*}$$

$$\begin{align*}
U_1^D &= \sqrt{2A_2LrE_s \dddot{y}_1} \sqrt{\langle J_1(y_2)\dot{y}_1 + T_1 + 2M_{F_1}(y_1) \rangle} \\
U_2^D &= \sqrt{2A_2LrE_s \dddot{y}_1} \sqrt{\langle J_2\dot{y}_2 - \frac{d_1(y_2)}{2} \dot{y}_2 - m_{hal}g_{hal} \sin(y_2 + \frac{\pi}{2}) + M_{F_2}(y_2) \rangle}. \hspace{1cm} \text{with} \hspace{1cm} (14)
\end{align*}$$
(\(−\), \(U \geq 0\); \(+\), \(U < 0\)) the second one of eq. (11). Moreover, the dimensions of the system input and output correspond as stipulated in eq. (12): \(\text{dim } u = \text{dim } (U_1, U_2)^T = \text{dim } (\psi_1, \psi_2)^T = \text{dim } y\). Consequently, eq. (8) describes a flat system, that owns the flat output \(y_{\text{lin}} = (\psi_1, \psi_2)^T\). Further, the expression for \(u = (U_1, U_2)^T\) in eq. (14) is the corresponding flatness based control law. In order to compensate model imprecision and incorrect initial values, the flatness based feedforward control (output voltage \(u_{\text{ff}}\)) will be supplemented by a proportional controller (amplification factor \(K_{\text{fl}}\), output voltage \(u_{\text{add}}\)). Figure 5 shows the complete controller design.

\[
\begin{align*}
\text{off-line calculation: } & y_{\text{rt}}, \dot{y}_{\text{rt}}, \ldots, y_{\beta+1}, \text{rt} \\
\text{flatness-part } & K_{\text{fl}} \\
\text{nonlinear feedforward control } & u_{\text{ff}} \\
\text{flatness-part } & K_{\text{fl}} \\
\text{plant } & u \\
\text{controller output voltage, feedforward voltage and additive voltage of actuator 1} \\
\text{controller output voltage, feedforward voltage and additive voltage of actuator 2} \\
\text{pressures and differential pressure of actuator 1}
\end{align*}
\]

**V. EXPERIMENTAL RESULTS**

Two sine curves are used as reference trajectories to test the developed multivariable control algorithm on the robot (figure 6). The corresponding control errors are shown in figure 7. Moreover, figures 8 and 9 show the fragmentation of the respective controller output voltage \(U\) into its two parts \(U_{\text{ff}}\) and \(U_{\text{add}}\) (c.f. figure 5). The part of \(U_{\text{ff}}\) in \(U\) gives an idea of the plant model accuracy. Simply said: The more \(U_{\text{ff}}\) equals \(U\), the more accurate the plant model is. So figure 9 shows, that the part of \(U_{\text{ff},2}\) in \(U_2\) rises, when actuator 2 lifts (c.f. figures 4 and 6) the robot arm (e.g. \(t \approx 22 \text{ - } 29\text{ s}\)). A possible cause for that is the neglected counterbalance valve (see figure 2), which mainly goes into action – decisive are the system pressures – when the robot arm is lowered: The model accuracy decreases by what the proportional controller increases in influence. In comparison with that, figure 8 shows, that the part...
of $U_{\text{h1}}$ in $U_1$ is approximately constant: Because of the corresponding horizontal movement, the influence of the counterbalance valve can be nearly identical for right- and left-hand movements so that there is no significant change in the composition of $U_1$. The voltage leaps of $U_{\text{h}}$ in figure 8 (e.g. $t \approx 17\, \text{s}$, $t \approx 33\, \text{s}$) and small ones in figure 9 (e.g. $t \approx 12.5\, \text{s}$, $t \approx 37.5\, \text{s}$) are caused by the change of sign of the friction torque according to the respective friction curve every time the moving direction changes.

Furthermore, the curve progressions in figure 10 show a different behavior for the right/left-hand movement of actuator 1, e.g. unequal high pressure maxima for $t \approx 8\, \text{s}$ and $t \approx 25\, \text{s}$. This dissimilarity can be recognized in the corresponding voltage and error curves too and it also appears if only actuator 1 moves. Apparently the real system behaves not as “symmetrically” as supposed, maybe as a result of asymmetrical valve behavior (proportional directional valve or one of the neglected valves). Such a right/left-hand movement difference in figure 11 is caused by the force of gravity (actuator 2 moves vertically).

VI. CONCLUSION

The flatness based controller of the rotary vane actuators offers simple adjustability by just one coefficient for each actuator (c.f. figure 5). Further it only needs the rotation angles as measurement signals and therefore just two angleometers as sensors. For the angular measurements incremental angle encoders are used, so there is no problem with measurement noise at all. Particularly, with regard to the discrete signals of the encoders, it is very favorably that no time derivatives of them are necessary for the flatness based control of the rotary vane actuators.

With a signal of the (disturbance) torque $M_L$ (c.f. figure 3), that represents the outer actuator load, a decentral control of the respective rotary vane actuator can be established. I.e.: Each joint (actuator) only together with connected valves, pressure source and return line forms a separate plant (c.f. eq. (2)). The outcome of this approach is a relativ simple plant model (compare eq. (2) to eq. (8)) and high controller robustness with regard to changes in the outer actuator load, e.g. caused by changes in the payload. In comparison to a differential cylinder (load force can be easy measured e.g. by means of strain gauges applied to its piston rod) the outer actuator loads of the used rotary vane actuators are difficult to measure. So, continuous works can deal with the major problem of disturbance determining for an adequate system class.

The experimentally determined characteristic curves (friction curves, leakage curves) make a precise system modeling possible. But the curves refer to a reference state of the system (e.g. a fixed oil temperature and system abrasion). Away from that state, they lose on validity. Furthermore, the actuator friction also depends on the system load because of the joint function of the rotary vane actuator. In the case of a dynamic system load, this effect cannot be exactly described by a time-invariant friction curve. Consequently, an actuator control without the explicit use of such characteristic curves could be subject of further works.

At last, with regard to the used testbed, the modeling of the neglected hydraulic valves and their coupling to the directional valves may be investigated in order to determine their influences and to increase model accuracy.

REFERENCES


Fig. 11. Pressures and differential pressure of actuator 2