Joint Actuation Based on Highly Dynamic Torque Transmission Elements – Concept and Control Approaches

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Abstract—Electric motors clearly constitute the most common drive principle in robotics and mechatronics. Smart materials, however, offer considerably higher power-to-mass ratios than electric motors. If mechanical energy instead of electrical energy can be distributed through a system, highly dynamic and efficient torque transmission elements based on smart materials, e.g. piezoceramics, can be used to transmit torque from an input to an output element. Just like electric motors, they can thus provide position, velocity, and force-torque control of the output element. This paper introduces machine components, called adaptronic couplers, which can transmit variable torques highly dynamically from an input element to an output element employing static or dynamic friction. In the long run, systems (e.g. robots) based on these machine components are envisaged to compete with systems based on classic drive principles especially electric motors w.r.t. dynamics and power-to-mass-ratio. Apart from the concept itself, this paper addresses different control approaches and discusses their influence on energy consumption and wear. Moreover, various experimental results proving the basic concept are presented.

I. INTRODUCTION

In various mechatronics applications – especially in robotics – joints are commonly actuated by electric servo motors. Servo mechanisms with appropriate sensors enable flexible control of the joint, e.g. position, velocity, or torque control. However, in several applications, distributing mechanical energy rather than electrical energy through the system may be feasible as well. As an example, serial manipulators are briefly considered. The achievable dynamic performance of serial manipulators is limited by the structural mass to be accelerated. Electric servo motors and gears constitute a significant fraction of this mass. Instead of distributing electrical energy through the kinematic chain, mechanical energy may be distributed and transmitted to the joints rendering energy conversion elements - such as electric motors - unnecessary. This approach requires a machine element which is capable of transmitting variable torques from an input shaft to the joints in a highly dynamic manner thus enabling torque, velocity, and position control of the joint with a performance that is comparable to electric servo motors. Moreover, the torque-to-mass ratio of this machine component has to exceed that of electric motors to allow reducing the structural mass at all. Existing machine components such as torque-controllable clutches, however, offer neither the necessary flexibility, nor a favorable torque-to-mass ratio, nor the required dynamics. Revolute adaptronic couplers (RACs) introduced in this paper may overcome these drawbacks. Employing appropriate sensors, RACs can provide torque, velocity, and position control of driven elements such as joints or output shafts and hence offer a flexibility comparable to electric servo motors.

The basic idea of RACs is to control (static and/or dynamic) friction to transmit variable torques between input shafts and output elements. The friction contact may be based on fluid friction or solid friction. Regarding fluid friction, electrorheological and magnetorheological actuators may be used to transmit variable torques dynamically. However, both electrorheological and magnetorheological actuators bear disadvantages compared to piezoelectric actuators. Clutches based on magnetorheological fluids have been investigated intensively, e.g. [1], [2]. Unfortunately, these solutions show undesirably high residual torques due to remaining viscous friction as well as comparatively low torque-to-mass-ratios due to the components for the generation of the magnetic field. Regarding electrorheological fluids, the transmissible shear stresses are even lower than those of magnetorheological fluids. In contrast to the previously described solutions, our approach employs piezoelectric actuators combined with suitable solid friction couples. In contrast to electrorheological and magnetorheological solutions, wear and durability issues have to be investigated more closely since torque transmission is based on solid friction. However, attractive torque-to-mass ratios and high dynamics of piezoelectric actuators render this approach worth to pursue.

RACs may be employed to drive various mechanisms involving revolute joints. Apart from torque transmission and revolute joints, special adaptronic couplers may also be used to transmit forces and drive prismatic joints. These so-called prismatic adaptronic couplers (PACs) have been proposed in [3]. PACs and PAs may also be employed in serial, parallel, or hybrid kinematic machines that have less motors than active joints. Particularly interesting are serial, parallel or hybrid kinematic machines that are driven by a single motor integrated in the robot base. This idea of robots which are driven by only one motor has already been pursued by Karbasi et al. [4]. They used application-specific spring wrap clutches in each joint to transmit variable torques from an input shaft. Due to the poor dynamic properties of spring wrap clutches the overall
performance of the system is clearly inferior to common drive principles. Actuators employing smart materials, particularly piezoceramics but also magnetorheological and electrorheological fluids can achieve clearly better dynamic properties and higher power-to-mass ratios than spring wrap clutches or other conventional machine components that can be used to transmit variable torques.

Due to the clearly nonlinear characteristics and hysteresis of the employed piezoelectric stack actuators [5] and the friction contact, linear control theory yields suboptimal performance results with RACs. As the full force range of the actuator is used, the large signal behavior of the actuator has to be investigated [6] if nonlinearities and hysteresis are to be compensated. Regarding control approaches which employ fast switching between minimum dynamic friction and static friction, high operating frequencies are necessary and hence dielectric losses in the actuator have to be considered. Several thermo-electro-mechanical models of piezoelectric actuators under operating conditions that are close to those of RACs have been proposed, e.g. [7]. Senousy et al. [8] recently showed that appropriate heat-sinking is necessary to reduce self-heating of PZT actuators when driving the actuator with high frequencies and high voltages, i.e., operating conditions that resemble those of pulse width modulation (PWM) or pulse frequency modulation (PFM) controlled RACs. Apart from the actuator the friction contact may show significant nonlinearities. Therefore, the system performance may be improved by considering models of the friction contact [9], [10] in the control approach. The proposed RACs show a residual torque which results from a residual normal force due to actuator prestress. The residual torque may be reduced by superimposing high-frequency signals on the controller output [11].

The remainder of the paper is organized as follows: In Section II the basic idea of RACs is introduced and the functional principle of the current prototype is presented. Moreover, key issues of the concept are discussed. Section III briefly addresses the employed control approaches and relates them to energy consumption. In Section IV various experimental results proving the feasibility of this drive principle are presented. Furthermore, critical design parameters are identified. Section V presents potential areas of future work and concludes the paper.

II. CONCEPT

This section introduces the concept of RACs. The first subsection addresses the general functional principle of RACs and presents a future example application in robotics to motivate the approach. In the second subsection the current prototype is described. The last subsection focuses on wear, heat losses, and energy efficiency.

A. General Principle and Application in Robotics

The basic idea of RACs is to use dynamic or static friction deliberately to transmit torques between shafts or an input shaft and an output element [3]. Depending on the employed sensor configuration and controllers, torques, velocities, or joint/shaft angles may be used as set points. One potential field of application are serial kinematic machines. Instead of employing a geared electric motor in each joint, RACs could be used to transmit mechanical drive energy to each joint directly. Fig. 1 depicts an abstract view of a serial kinematic structure based on RACs. In contrast to common serial robots, the proposed structure features a single motor that is integrated in the robot base. The energy of the motor is transmitted through the entire kinematic chain using drive shafts, traction mechanism drives, etc. Each joint requires one RAC unit consisting of up to three RACs. Two RACs in each RAC unit are employed to move the joint in either direction of rotation while the third optional actuator may be used to clamp two links to avoid any unnecessary dynamic friction when the joint is not moving.

Each joint has to be able to rotate clockwise and counterclockwise. Therefore, the direction of rotation of the drive shaft has to be inverted for one RAC in each joint. To supply multiple joints independently by a single drive train, the rotation speed of the drive train has to be higher than the maximum rotation speed of the joints. Therefore, reduction gears have to be employed to reduce the speeds of the input shafts of the RACs. Thus, to each joint, a fraction of the drive energy is transmitted. Since the drive energy is provided mechanically, no energy conversion elements, e.g. electric motors, are necessary in the links. As piezoelectric stack actuators show very high power-to-mass-ratios and no high gear ratios are necessary, in the future the mass of each link may be reduced compared to classic drive principles involving geared electric motors – thus improving the dynamic properties of the system.

B. Current Prototype

To evaluate the viability of the concept, a prototype has been developed that contains one piezoelectric stack actuator which establishes a friction contact to an input shaft thus enabling rotation of an output element around the input shaft. As only one actuator and no reversing gear is employed, the transmitted torque can only cause a rotation of the output element against gravity. The maximum acceleration of motions in the opposite direction is hence limited by gravity. Nevertheless, controlled motions in this direction can be executed.

Fig. 2 shows a CAD model of the prototype. The housing ((1) and (2)) of the prototype is parted radially to simplify the assembly of the RAC. The input shaft (3) running through the housing is axially and radially directed by two flanged bushes (4). The left end of the input shaft is connected to a servo motor by a bellows coupling; on the right end an encoder is mounted to measure the input shaft angle. The
actuator (6) applies a normal force to the central sliding bush (5) which causes a friction force and hence an output torque. The actuator is guided in the radially oriented bushing (9) by two washers ((7) and (8)) at its ends which also protect the actuator from lateral forces. The bushing is screwed into the upper half (1) of the housing and fixed by a lock nut (10). The radial prestress of the centered sliding bush and the actuator can be adjusted by a threaded cap (11). The force generated by the piezoelectric actuator is measured by a very stiff force transducer (12) mounted in the load path between the cap and the upper washer. The pulley (13) transmits the rotation of the output element to a precision encoder (not shown).

As already stated above, torque transmission is based on the friction force generated at the bush-shaft contact. The prestress applied by the threaded cap and the expansion of the actuator result in a displacement of the lower washer and a deformation of the upper part of the housing which accommodates the central sliding bush. In Fig. 3 the displacements of the lower washer and the deformations of the upper part of the housing are shown for two loading conditions. Fig. 3a) shows the displacements due to the initial prestress applied by the threaded cap; Fig. 3b) shows the displacement/deformation occurring with maximum actuator stroke. The ball shaped washer results in a circular deformation of the bush receptacle and hence the sliding bush.

Fig. 4 shows a picture of the prototype that has been used for the experiments presented in Section IV. Apart from the RAC itself, the picture shows the servo motor that drives the input shaft and the precision encoder which measures the angle of the output element. Due to the rather simple and rugged design, the torque-to-mass ratio of the depicted adaptronic coupler (not the entire experimental setup) is not yet comparable to that of a state-of-the-art electric servo motor.

The current prototype employs a Piezomechanik high voltage PZT stack actuator Pst 1000/16/80 (stack diameter \( d_{st} = 16.1\, \text{mm} \); stack length \( l_{st} = 72\, \text{mm} \)) with a blocking force of approx. \( F_{\text{max}} = 12\, \text{kN} \) and a maximum stroke of \( \Delta l_{\text{max}} = 80\, \mu\text{m} \). A high voltage amplifier Piezomechanik LE 1000/035 with a voltage range of \( 0 - 1000\, \text{V} \) and an maximum current of \( I_{\text{av}} = 100\, \text{mA} \) supplies the actuator. The input shaft is driven by a servo motor Maxon EC 60 with a planetary gearhead (nominal torque \( \tau_{\text{nom}} = 25\, \text{Nm} \); max. angular velocity \( \dot{\alpha}_{\text{in,max}} = 540^\circ/\text{s} \)). The mechanical energy of the input shaft is transmitted by a PTFE-coated cylindrical sliding bush (Federal Mogul GLYCODUR F; diameter \( d_{bu} = 25\, \text{mm} \)) with a low coefficient of dynamic friction ranging between \( 0.03 < \mu_d < 0.25 \). The high wear and low friction coefficient of the PTFE running-in layer [12] are clearly suboptimal for this application. An internal force sensor (GS sensors XFC-200 10kN) whose stiffness is approx. 13 times higher than the maximum stiffness of the piezoelectric actuator and the remaining load path is used to measure the applied normal force. Forces applied to the environment are measured with an external 6D force-torque sensor manufactured by JR3. Closed loop control and data logging are performed with a sampling frequency of \( f_s = 5\, \text{kHz} \).

C. Key Issues – Wear, Heat Losses, and Energy Efficiency

The transmission of energy via a friction contact inevitably entails heat losses and wear of the friction couple which reduce energy efficiency and service life of the device. However, the wear rate \( \dot{w} \) of the friction couple may be reduced by the selection of appropriate materials and optimized control approaches. Currently, a conventional friction couple is employed, i.e. a tin bronze sliding bush coated with PTFE and a steel shaft. Except for the PTFE running-in layer, the wear rate \( \dot{w} = \frac{\dot{F}_{\text{fr}}}{dx} \) (\( dz \) denotes the reduction in thickness; \( x \) is the sliding distance) of the sliding bush is comparatively low. However, the friction coefficient \( \mu = \frac{F_{\text{fr}}}{F_n} \) (\( F_{\text{fr}} \) denotes the friction force; \( F_n \) the applied normal force) is very low as well – as already stated above. Additionally, \( \mu \) may vary by an order of magnitude depending on parameters like temperature, velocity, etc. To allow for an efficient energy transmission, a friction couple with both a low wear rate \( \dot{w} \) and a high friction coefficient \( \mu \) is desirable. As the magnitude of the friction force \( F_{\text{fr}} \) directly relates to the achievable output torque \( \tau_{\text{fr}} = F_{\text{fr}} r_{sh} \) (\( r_{sh} \) denotes the radius of the input shaft), a high \( F_{\text{fr}} \) is generally desirable. This aim may be achieved by either applying high normal forces \( F_n \) which generally involves powerful and heavy actuators or selecting friction couples with a high \( \mu \). The transmitted output torque is given by Eq. (1)

\[
\tau_{\text{fr}} = \mu F_n r_{sh}
\]

\( \mu \) and \( r_{sh} \) are critical design parameters that directly influence the achievable output torque \( \tau_{\text{fr}} \). The combination of a high friction coefficient \( \mu \) and a low wear rate \( \dot{w} \) is desirable for applications like clutches, brakes, and actuators that employ dynamic or static friction, e.g. inchworm motors. Generally, low wear is accompanied by low friction as is high wear by high friction [13]. However, tribological principles and advanced manufacturing processes may be employed to create friction couples that offer high friction coefficients and low wear at the same time. For example, Chatterjee et al. [14] have developed a friction interface with pseudoelastic nickel titanium showing a high friction coefficient and low wear for the use in inchworm motors. Regarding conventional tribological coatings, \( Al_2O_3 \) offers a viable trade-off between a low wear rate and a high
friction coefficient [15].

Even when the wear rate \( \dot{w} \) and the necessary actuator power can be reduced by a suitable friction couple, the problem of heat losses in the friction contact remains. Combining constructional techniques and control approaches, however, heat losses in the friction contact may be drastically reduced by avoiding dynamic friction and instead switching between minimum dynamic friction and static friction. Thus, the dissipated energy may be reduced significantly depending on the rate of transition between minimum dynamic friction and static friction. Since piezoelectric actuators offer very low response times, this switching control may be a viable approach. However, fast switching entails considerable heat losses in the piezoelectric actuator and additionally the heat losses due to deformations of the bush and housing have to be considered.

### III. CONTROL APPROACHES

As already indicated above, the achievable flexibility and performance of RACs are mainly determined by the employed control approaches. Depending on the control approach, RACs may be used as torque-controllable clutches, continuously variable transmissions, or swivel units (for revolute joints). Hence, these applications may be considered as different operation modes, which can be switched by simply changing the active controller.

Although RACs show several sources of nonlinearity, e.g. the voltage-torque characteristic of the actuator and the characteristics of the friction contact, linear control theory and simple models may be applied successfully to demonstrate the viability of this concept.

#### A. Position Control

The most simple control approach successfully applied for the adaptronic coupler is single-loop PID control with a simple feedforward term considering system dynamics of the actuator. The experimental results presented in Section IV have been obtained with a cascaded PID controller though – consisting of a position controller and a velocity controller. The velocity controller of the servo motor employs a feedforward term neglecting the dependency on temperature and the necessary actuator power.

The required torque \( \tau_{\text{ffwd}} \) used as a feedforward term is given by Eq. (2).

\[
\tau_{\text{ffwd}} = J \ddot{\alpha}_{\text{out}} + mg r_{\text{COM}} \cos \alpha_{\text{out}} - \tau_{\text{ba}}(\dot{\theta}) \quad (2)
\]

\( J \) denotes the moment of inertia of the output element around the shaft axis; \( m \) denotes the mass of the output element; \( r_{\text{COM}} \) is the radius (measured from the center of the input shaft) at which the center of mass (COM) of the output element is located; \( g \) denotes the gravitational acceleration; \( \alpha_{\text{out}} \) denotes the angle of the output element and \( \ddot{\alpha}_{\text{out}} \) denotes its angular acceleration. The residual friction of the three bush effects that surround the input shaft is roughly approximated by \( \tau_{\text{ba}}(\dot{\theta}) \) where \( \dot{\theta} \) indicates the temperature dependency.

The output torque \( \tau_{\text{ffwd}} = 1 \) is generated by the friction force \( F_{fr} \) resulting from the normal force \( F_n \) applied to the sliding bush by the actuator. Based on a suitable model of the actuator, e.g. given by Eqs. (3) and (4), a feedforward voltage could be calculated.

\[
F_{n}^{t+\Delta t} = \Gamma \left( \left[ V_{\text{act}}^t, V_{\text{act}}^{t+\Delta t}, ..., V_{\text{act}}^{t-M\Delta t} \right], p, g, s_{\text{env}} \right) \quad (3)
\]

\[
V_{\text{act}}^t = \tilde{\Gamma}^{-1} \left( F_{\text{des}}^n, \left[ V_{\text{act}}^{t-\Delta t}, ..., V_{\text{act}}^{t-M\Delta t} \right], p, g, s_{\text{env}} \right) \quad (4)
\]

\( \Gamma \) and \( \tilde{\Gamma}^{-1} \) denote non-linear functions. \( V_{\text{act}}^t \) denotes the applied actuator voltage at time \( t \); \( \Delta t \) is the sampling time, and \( M \) denotes the memory length. \( F_{\text{des}}^n \) is desired normal force. The vector \( p \) contains the piezoelectric 'constants', i.e., material parameters that actually depend on temperature, loading, etc.; \( v \) comprises the necessary geometric parameters; \( s_{\text{env}} \) denotes the stiffness of the sliding bush/housing.

Since the experimentally determined voltage-torque characteristic of the system shows considerable nonlinearity and hysteresis (cf. Fig. 11) as well as a dependency on the temperature of the friction contact and the actuator, the feedforward term is currently based on the inverse of the experimentally determined voltage-torque characteristic \( \Psi^{-1} \) instead of Eq. (4), i.e.,

\[
V_{\text{act}, \text{ffwd}} = \Psi_{V_{\text{act}, \text{fr}}}^{-1}\left( \tau_{\text{ffwd}} \right) \quad (5)
\]

The voltage-torque characteristic used for the feedforward term neglecting the dependency on temperature and the hysteresis can be approximated by a third order function.

This \( V_{\text{act}, \text{ffwd}} \)-continuous control approach relies on dynamic friction between the sliding bush and the input shaft. As already stated in Subsection II-C, dynamic friction is undesirable regarding both energy dissipation and wear of the friction couple. Due to the inherent mechanical low-pass characteristic of the system, PWM/PFM control approaches may be applied to switch between the states of minimum dynamic friction, i.e., the residual dynamic friction due to the prestress of the actuator, and static friction. Since the accelerations achievable with piezoelectric stack actuators and appropriate drive electronics are generally very high, the transition times between the states are short.

Regardless whether PWM or PFM is employed, a state-dependent voltage has to be calculated that assures the friction contact to enter the state of static friction while at the same time minimizing actuator energy dissipation and the deformation energy dissipated in the sliding bush and its receptacle.

The influence of the employed control approach will be briefly discussed for a serial kinematic machine based on RACs. Regarding the energy conversion efficiency of a serial kinematic machine with a single motor and adaptronic couplers in each of its \( N \) joints, the total input energy is comprised of the energy supplied to the motor \( \tau \rho E_{\text{mo}} \) and the
energy to power the adaptronic couplers $^{in}E_{adc}^i$ (cf. Eq. (6)).

$$^{in}E_{tot} = {^{in}E_{mo}} + \sum_{i=1}^N {^{in}E_{adc}^i}$$

(6)

Since the adaptronic couplers are sole torque transmission elements, their output energy does not contribute to the net energy output of the joint. Moreover, mechanical input energy of the shaft is dissipated in the friction contacts when dynamic friction occurs $^{diss}E_{fr}^i$. The efficiency of the motor and the gears is considered by $\eta_{mo}$ and $\eta_{tr}$ respectively. Thus, the total dissipated energy is roughly given by Eq. (7).

$$^{diss}E_{tot} = {^{in}E_{mo}}(1-\eta_{mo}\eta_{tr}) + \sum_{i=1}^N \left( {^{in}E_{adc}^i + ^{diss}E_{fr}^i} \right)$$

(7)

The fraction of the energy provided by the motor which is dissipated in each RAC due to dynamic friction is given by Eq. (8).

$$^{diss}_E_{fr}^i = \int \tau_{fr}(t)|\dot{\alpha}_{out}(t) - \dot{\alpha}_{in}(t)|dt$$

(8)

Again, $\dot{\alpha}_{out}(t)$ and $\dot{\alpha}_{in}(t)$ denote the angular velocity of the output element and the input shaft respectively. Concerning the energy supplied to each of the adaptronic couplers, two major terms can be identified.

$$^{in}E_{adc}^i = {^{env}E_{adc}^i} + ^{diss}E_{adc}^i$$

(9)

$^{env}E_{adc}^i$ denotes the fraction of input energy which is actually passed to the environment and $^{diss}E_{adc}^i$ denotes the fraction which is dissipated into heat in the actuator due to dielectric losses.

This brief consideration reveals the critical design parameters of the system. Focusing on the energy that is dissipated in the RACs, three terms are crucial: the energy dissipated by dynamic friction $^{diss}E_{fr}^i$, the energy that eventually leads to a deformation of the sliding bush $^{env}E_{adc}^i$, and the dielectric losses of the actuator $^{diss}E_{adc}^i$. These terms directly lead to the following optimization options: (i) $^{diss}E_{fr}^i$ may be reduced by avoiding dynamic friction, e.g., using a PWM/PFM control approach; (ii) $^{diss}E_{adc}^i$ may be decreased by avoiding high frequencies (or steps resp.) in the actuator voltage; and (iii) $^{env}E_{adc}^i$ may be decreased by high friction coefficients as the required normal force is reduced. Particularly (i) and (ii) lead to conflicting design goals, viz., avoiding dynamic friction by PWM/PFM control approaches increases dielectric losses which increase with the frequency of the operating voltage. Therefore, trade-offs between these goals have to be closely investigated.

Regarding the proposed $V_{act}$-continuous control approach, the torque is transmitted by means of dynamic friction unless the velocity of the output element reaches the angular velocity of the input shaft, i.e., when the desired angular velocity is equal or greater than the angular velocity of the input shaft. However, transmitting the torque using dynamic friction leads to considerable wear of the friction couple and undesirable energy dissipation $E_{fr}$. If the system could instantaneously switch between the state of minimum dynamic friction and the state of static friction, ideally no energy would be dissipated at the friction contact. Since piezoelectric actuators offer very high dynamics, fast switching between these states proves to be a viable approach to reduce the intervals of non-minimum dynamic friction and thus the dissipated energy $E_{fr}$ and the wear $w$ of the friction couple. While $E_{fr}$ is reduced by the PWM/PFM control approach, $^{diss}E_{adc}$ is increased due to the increased dielectric losses in the actuator resulting from the high operating frequencies.

The PWM/PFM control approach switches between a state of minimum dynamic friction and a state of static friction by applying an actuator voltage of $V_{act,off} = V_{min}$ in the interval $t_{off}$ and a voltage $V_{act,on}$ during $t_{on}$. The PWM cycle time $t_{PWM}$ is given by $t_{PWM} = t_{on} + t_{off}$. $V_{act,on}$ is selected such that the friction contact enters the state of static friction. Therefore, $F_{act}^{des}$ in Eq. (4) has to be selected accordingly, i.e.,

$$F_{act}^{des} = \frac{\tau_{fr}}{\mu_s \tau_{sh}}$$

(10)

$\mu_s$ denotes the coefficient of static friction. $\gamma > 1$ assures that the system enters the state of static friction.

Regarding a single RAC, the energy dissipated by dynamic friction during one PWM control cycle may be estimated by Eq. (11).

$$E_{fr} = \int_0^{t_{s_1}} F_n(t)\mu_s \tau_{sh} \left| (\dot{\alpha}_{in}(t) - \dot{\alpha}_{out}(t)) \right| dt$$

$$+ \int_{t_{s_2}}^{t_{PWM}} F_n(t)\mu_s \tau_{sh} \left| (\dot{\alpha}_{in}(t) - \dot{\alpha}_{out}(t)) \right| dt$$

(11)

where $t_{s_1}$ and $t_{s_2}$ denote the time instants when the system enters/leaves the state of static friction. $|\dot{\alpha}_{in} - \dot{\alpha}_{out}|$ is the absolute value of the relative angular velocity between the input shaft and the output element. Clearly, PWM/PFM control and low response times can significantly reduce the energy dissipated by dynamic friction.

B. Force Control

The force control approach employed in the experiments presented in Section IV uses a single-loop PID controller and a voltage feedforward term based on the voltage-torque characteristics of the system (cf. Eq. (5) and Fig. 11)). Fig. 5 shows the RAC and a simplified view of the functional blocks of the control system.

IV. EXPERIMENTAL RESULTS

This section presents the experimental results obtained with the prototype described in Section II. The preliminary experimental results demonstrate the viability of the concept and identify the weak points to be addressed.
A. Position Control

The first experiments show the performance of the RAC in position control mode. Fig. 6 shows the step response of the system for an angle step of $\Delta \alpha_{\text{out}} = 5^\circ$. The slope of the step response equals the angular velocity of the input shaft since the controller output voltage causes an actuator force that establishes a static friction contact.

The trajectory-following behavior of the system is depicted in Fig. 8. Both trajectories are jerk-limited 7-segment trajectories [16]. The trajectory on the left has been executed with the $V_{\text{act}}$-continuous control mode and the trajectory on the right using the PWM controller and a PWM cycle time of $t_{\text{PWM}} = 10\text{ms}$. As can be seen, no qualitative differences in the following behavior can be observed. This observation is confirmed by similar mean control errors.

To allow for low PWM cycle times, the actuator and the remaining system have to offer sufficient dynamics characteristics. Figs. 9 and 10 show the resulting normalized normal forces $F_n$ measured by the internal force transducer (cf. (12) in Fig. 2) when an upward and a downward voltage step are applied to the actuator. The upward response shows a delay of approx. $1.5\text{ms}$; $90\%$ of the actuator force are reached after approx. $t_{\text{up}} = 4\text{ms}$. The downward step does not lead to a noticeable delay. The PWM approach relies on establishing a static friction contact. Depending on the dynamic state of the output element, the required normal forces require high actuator voltages. Due to the delayed response of the system the minimum PWM cycle time of the system is currently limited.

B. Force Control

To evaluate the force control performance, the output element driven into contact with an external force/torque sensor fixed on the mounting plate (cf. Fig. 4) and different force profiles are applied to the force controller. The input shaft velocity is constant again. Fig. 12 shows the force control performance of the system. The maximum control error during the constant force interval is less than $0.4\text{N}$. In contrast to the position control performance, the force control performance is already very promising.

Fig. 11. The voltage-torque characteristic of the RAC shows hysteresis which is mainly due to the hysteresis of the piezo stack actuator. Additionally, a significant nonlinearity can be observed especially in the range of $V_{\text{act}} = 0\text{V} - 400\text{V}$. The nonlinearity may be attributed to the nonlinearity of the piezoelectric actuator as well as the nonlinear spring constant of the bush receptacle and the sliding bush. The residual torque at $V_{\text{act}} = 0\text{V}$ is approximately $3\%$ of the maximum output torque of $\tau_{\text{fr}} = 4\text{Nm}$ at $V_{\text{act}} = 800\text{V}$. This torque can be attributed to the remaining friction of the sliding bushes, especially the prestressed sliding bush. Compared to magnetorheological or electrorheological solutions, the residual torque is quite low.
A. Conclusions

The introduced machine components, called RACs, may be employed in various systems to drive revolute joints. Apart from the basic concept itself, this paper has presented the design of and results obtained with a prototype which uses standard friction bearings for torque transmission. While the force control performance of the prototype is already very promising, electric servo motors clearly surpass the current prototype of the RAC w.r.t. position control performance. Significant improvements may be achieved regardless of the employed control approach. Regarding the design of and results obtained with a prototype which uses standard friction bearings for torque transmission. While the force control performance of the prototype is already very promising, electric servo motors clearly surpass the current prototype of the RAC w.r.t. position control performance. Significant improvements may be achieved regardless of the employed control approach. Regarding the position control, significant improvements may be achieved by considering the periodic deviations of the transmitted torque which are due to the geometric imperfections of the input shaft and the sliding bush. As the position control performance of the PWM/PFM approach (PWM cycle time \( t_{\text{PWM}} = 10 \text{ ms} \)) is comparable to the \( V_{\text{act}} \)-continuous control approach, a key problem of RACs, i.e., heat losses at the friction contact, may be reduced significantly by employing PWM/PFM control. Since the control performance improves but the friction losses in the actuator increase with decreasing PWM cycle times (resp. increasing PFM frequencies) resulting in actuator self-heating, the total heat losses of the system have to be considered when targeting energy-efficient control. The second key problem of RACs is the wear of the friction couple. While the employed sliding bushes provide desirably low wear, they also possess a very low friction coefficient which is disadvantageous for our application as a high normal force is required to obtain a given output torque.

B. Outlook

Since adaptropic couplers show a highly nonlinear behavior due to the piezoelectric actuator and the friction contact, the applied linear control approaches and the Coulomb friction model are clearly suboptimal. Developing suitable system models and applying model-based nonlinear control approaches will further improve control performance. The employed sliding bushes show low wear but also a low friction coefficient thus requiring high normal forces. The selection of appropriate coatings, e.g., \( Al_2O_3 \) [15], is supposed to improve wear behavior and increase the coefficient of friction. Another promising friction interface employs nickel titanium based materials as proposed by Chatterjee [14]. Apart from the materials of the friction couple, the geometry of the friction surfaces and the normal force distribution over the friction surfaces has to be considered to optimize the wear profile. To show the feasibility of higher torque-to-mass ratios, a new multi-joint prototype is currently being developed based on the design principles of structural and function integration.

VI. ACKNOWLEDGMENTS

We would like to thank QNX Software Systems for providing free software licenses.

REFERENCES