Design of a Modular Hydraulically Driven Variable Geometry Truss Structure and its Nonlinear Controller Architecture for Highly Dexterous Motion

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Abstract

In the paper, a design of novel 3-DOF octahedron-shaped modules for hydraulically actuated variable geometry truss manipulators and its nonlinear control architecture will be introduced. The main features of the design are the optimized multiple collocated spherical joints and a structure-integrated supply of the drive fluid for the hydraulic actuators. Based upon the presented structure, a family of highly maneuverable light-weight hyper-redundant manipulators can be derived.

Furthermore a model-based nonlinear control architecture is introduced for the hydraulically driven manipulator to ensure high control performance all over its workspace, in spite of the pronounced nonlinearities of the hydraulic drives and the structures kinematics. The advantages of the presented control approach are demonstrated by a comparison with well-known control designs for hydraulic drives using a virtual model of the manipulator.

KEYWORDS: variable-geometry-truss, hyper-redundant robots, hydraulic drives, nonlinear control architecture, feedback linearization

1. Introduction

Conventional manipulators for manufacturing processes are typically used in strongly structured environments, performing prescribed tasks and guaranteeing a required precision. In standard serial robotic manipulators the number of actuated joints is usually limited to the required minimum for the demanded task in order to ensure low moved masses. Consequently this restricts the reachable workspace in the presence of obstacles. Tasks for which an obstacle-free working environment cannot be asserted have therefore been largely exempt from thorough automation. The high cost percentage of the manual labor triggers today's significant efforts towards the exploration of new strategies to make those tasks accessible to automation.

To enhance maneuverability and flexibility of motions, additional axes of motion can be introduced, leading to kinematically redundant manipulators in which the number of actuated joints exceeds the degrees of freedom of the end-effector. They possess a larger dexterous workspace due to the possibility to avoid obstacles in the workspace, to compensate for joint limitations and singular configurations of the manipulator.

The drives in a redundant manipulator have to be able to realize a high payload-toweight and power-to-weight ratio in order to allow the cascading of the necessary number of structural modules to achieve kinematic redundancy. The feasibility of hydraulic actuation for redundant manipulators has been proven by a few successful redundant robot designs, most notably the Schilling Titan 4 /1/ and SARCOS Dexterous Arm /2/, both with seven DOF's and Zhao's ten DOF robot arm /3/. These and the most of all other approaches are in serial kinematic design which results, in order to ensure a demanded stiffness, in a relative low payload-to-weight ratio.

In contrast the stiff but light-weight hybrid serial-parallel variable geometry truss (VGT) structures provide an excellent payload-to-weight ratio when combined with hydraulic actuation. Until now, the full potential of hyper-redundant hybrid serial-parallel VGT structures has not been exploited, due to the complex dynamical behavior of the hydraulic drives, the nonlinear kinematics and the necessity for a suitable control design taking these effects into account. Due to their local nature, standard linear control designs for hydraulic drives, like PID- and state space controllers in combination with velocity-feed-forward controllers, cannot fully exploit the potential of the resulting structure to provide high accuracy and highly dynamic control of the manipulator all over the workspace.

2. Research objective

Our research was focused on the design of a hyper-redundant manipulator which achieves its high potential from: 1. A specific module kinematic with one translational and two rotatory degrees of freedom (DOF), which allows easy scalability, light-weight design and high common part degree; 2. The usage of hydraulic drives to provide high force-to-weight ratio to keep the moved masses low and to achieve high dynamic; 3. The development of a control design which can deal with the hydraulic and kinematical nonlinearities to access the full potential of the manipulator.

2.1. Module design

The class of hyper-redundant robots can be divided into three main groups /4/: highly segmented serial robots, hybrid serial-parallel robots and continuous robots. Because of their inherent underactuation and the resulting problems in modeling and control, the group of continuous robots was disconsidered from our study. Among the remaining segmented serial and hybrid serial-parallel structure, the most promising was the VGT in octahedron design, see **figure 1**. The main advantages of this type of structure are: theoretically zero bending moments on actuators and links due to the use of spherical joints; minimum number of actuators for the desired DOFs; large dexterous workspace of the composed manipulator; high common part degree. The designed modules primarily consist of three elements, see figure 1: the active links (a), the passive links (b), and the multi-collocated spherical joints (c).



Figure 1: Kinematic structure of a single module.

2.2. Hydraulic actuation and power supply

From the variety of existing hydraulic linear drives, the single rod double acting cylinder is the most feasible for the actuation of the octahedron VGT, see **figure 5**. In order to achieve a high movability, as well as a large workspace of the kinematic structure, the active link has to realize a high maximum-to-minimum length ratio k, which excludes hydraulic cylinders with a built in position sensor or end-of-stroke-damping. Beside the

high length ratio k the actuator has to possess a large piston rod area $A_B = \pi/4 \cdot (d_p^2 - d_{pr}^2)$ to provide high forces under compression $F_B = A_B \cdot p_0$. In order to choose the cylinder for the kinematic structure, an optimum between the length ratio k and the maximum force F_B , with respect to the constraints of maximum system pressure p_0 and maximum piston diameter d_p , has to be found. Special attention has to be paid to the danger of buckling of the hydraulic cylinder. Due to the mainly horizontal, free hanging and simply-supported cylinder, the danger of buckling is increased because of bending under the influence of the weight of the cylinder and the other assembly parts (valves, adapter plate, etc.). A safety factor of minimum 5 is recommended. By using zero lapped 4/3 proportional-way-valves, a good controllability of the actuators and a low energy consumption during holding operations can be achieved.

To realize the drive fluid supply to the actuators, a variety of solutions using hydraulic hoses and pipes is imaginable; three approaches are depicted in **figure 2**. The design shown in figure 2 (right), in which the passive links of the structure are designed as hydraulic pipes to transport the fluid proved the most promising. Since there are six passive links in each module, it is possible to use three links for power and three links for tank supply in order to reduce pressure losses/drops. This variant is mainly selected due to the advantage, that there are no separate and unrestrainedly moving hydraulic hoses within the structure. On each link a branching is inserted from which the power supply port of the respective actuator is connected via a conventional hydraulic hose, see **figure 3** (right). Also the ends of the passive links are connected with hydraulic hoses because these hoses have to cross each other in each joint point. The resulting system possesses a high hydraulic stiffness because of its rigid hydraulic pipes and minimal hose length (reduced up to 80 percent in comparison to the other variants).



Figure 2: Power supply routing

Since the passive links of the structure are used as hydraulic pipes, the joints have to provide enough space between the endings of the passive links to connect them with hydraulic hoses. In /5/ we compared different types of collocated spherical joints which are usable for the octahedron VGT. As a basis we chose the SJM (Spherical Joint Mechanism) introduced by Bosscher /6/, and optimized it for mechanical load, collision freeness, and to provide enough space between the passive links in order to connect them with hydraulic hoses, see figure 3 (left). When the structure is moving, the passive links are twisting towards each other, which would result in torsion of the hoses. To avoid this torsion, which is not recommended for standard industrial hoses, therefore one of the connections to the passive link has to be designed as a swivel coupling.



Figure 3: Spherical joint with bearing for <u>active</u> and <u>passive</u> links (left) and hose routing (right).

2.3. Resulting manipulators and properties

The designed modules can be cascaded to form a manipulator as shown in **figure 4**. For one module the maximal attainable relative height difference $\Delta h^* = \Delta h / l_0$ over both module angles ± 30° is 0.33 and the payload-to-module-weight ratio *m* reaches values of up to 22 /5/. If more than two modules are combined, the resulting manipulator will have redundant DOFs. The payload-to-mass ratio r_{pm} , which is an important benchmark value to evaluate the handling capability of manipulators, reaches values of up to 10 for the three module VGT shown in figure 4. Standard industrial manipulators possess an r_{pm} of 0.05, while modern light-weight manipulators can reach an r_{pm} of up to 1. The high r_{pm} of the presented manipulator ensures higher dynamic motions and lower energy consumption in comparison to standard industrial robots.



Figure 4: Cascaded manipulator

3. Modeling of the hydraulic drives

The presented hydraulic drives of the VGT consist mainly of four units: constant pressure sources, hydraulic pipes and hoses, proportional-valves and hydraulic cylinder, see figure 5. The constant pressure source, the pipes and hoses we supposed to be ideal, therefore the modeling effort was concentrated on the valves and cylinders.



Figure 5: Nomenclature for modeling of the hydraulic drive.

3.1. Proportional valve

The dynamic behavior of the proportional valve can be described as a second-order dynamic system. The control spool position y_v is measured in the valve. Its behavior can be described by the differential equation (1).

$$\frac{d}{dt}\begin{bmatrix} y_v\\ \dot{y}_v \end{bmatrix} = \begin{bmatrix} \dot{y}_v\\ \omega_0^2 K_v u_E - 2D_v \omega_0 K_v \dot{y}_v - \omega_0^2 y_v \end{bmatrix}$$
(1)

The valve gain is the quotient of maximum valve position and maximum control voltage $K_v = y_{v,max} / u_{E,max}$. The angular resonance frequency ω_0 and damping D_v can be derived from the datasheet of the valve. With $\omega_0 = 350 \text{ Hz}$, the typical corner frequency of the valve is three times higher than the resonance frequency of the cylinder. The static valve behavior, which leads to the volume flow Q_A in operating port A and Q_B in operating port B, is a function of the spool valve position y_v and the pressure drop over the control edges which are depending on the system pressure p_0 , the tank pressure p_T , and the pressure on the control ports p_A and p_B . By defining $sgn(x) = \{x \forall x > 0, 0 \forall x \le 0\}$ the volume flowing into or out of the control ports A and B can be calculated using Equation (2) with nominal flow Q_{nom} and the nominal pressure drop $\Delta p /7/$.

$$\begin{bmatrix} Q_{A} \\ Q_{B} \end{bmatrix} = \frac{Q_{nom}}{\sqrt{\Delta p}} \cdot \begin{bmatrix} sgn(y_{v}) \cdot \sqrt{(p_{0} - p_{A})} - sgn(-y_{v}) \cdot \sqrt{(p_{A} - p_{T})} \\ sgn(-y_{v}) \cdot \sqrt{(p_{0} - p_{B})} - sgn(y_{v}) \cdot \sqrt{(p_{B} - p_{T})} \end{bmatrix}$$
(2)

3.2. Hydraulic cylinder

To allow the control of the actuator the piston rod position y_y and velocity \dot{y}_y , as well as the pressure values p_A and p_B on the control ports A and B have to be known. Under disregard of external leakage, the actuator behavior is depending upon the volume flows into the cylinder chambers Q_A and Q_B and the external force F_{ext} . The state vector to describe the behavior of the cylinder is therefore $\begin{bmatrix} y_y & \dot{y}_y & p_A & p_B \end{bmatrix}^T$, its derivative is written in (3) with the viscous damping constant d, the effective mass of the piston rod and attached elements that move along with it m, the hydraulic stiffness K_{oil} and the inner leakage factor K_{Lip} .

$$\frac{d}{dt}\begin{bmatrix} y_{y} \\ \dot{y}_{y} \\ p_{A} \\ p_{B} \end{bmatrix} = \begin{bmatrix} \dot{y}_{y} \\ \frac{1}{m} \cdot \left[p_{A} \cdot A_{A} - p_{B} \cdot A_{B} - d \cdot \dot{y}_{y} - F_{ext} \right] \\ \frac{K_{oil}}{A_{A} y_{y}} \cdot \left[Q_{A} - \dot{y}_{y} \cdot A_{A} + K_{Lip} \cdot (p_{B} - p_{A}) \right] \\ \frac{K_{oil}}{A_{B} \left(y_{y,max} - y_{y} \right)} \cdot \left[Q_{B} + \dot{y}_{y} \cdot A_{B} - K_{Lip} \cdot (p_{B} - p_{A}) \right] \end{bmatrix}$$
(3)

4. Nonlinear control architecture

Due to the inherently nonlinear behavior of both the electrohydraulic actuating system and the manipulator dynamics, linear controllers provide only very limited control performance. Therefore, for the control of the VGT manipulator we present a cascaded nonlinear control architecture as sketched in **figure 6**. The inner loop features a controller that linearizes the nonlinear force generation and applies a linear control law to the linearized force plant. The outer motion control loop features a feedforward linearization of the manipulator dynamics by computed torque along with speed and position controllers to suppress disturbances.





4.1. Feedback linearization of the hydraulic drive

Since the piston rod position y_y and its velocity \dot{y}_y change their value slowly in comparison to the hydraulic states, they can be regarded as time-invariant parameters for the control of the hydraulic subsystem. This allows the decoupled control of hydraulic and mechanic states. With this separation, only uncomplex linearization

terms result, which then allows a straightforward design of the controller, as compared to the one in /8/.

The dynamic model of the 4/3 proportional valve, see Equation (1), was neglected for the feedback linearization, and will be taken into account in the force controller design process later in this section. The general equations (2) of the resulting volume flows Q_A and Q_B can be simplified direction-dependent for positive and negative spool valve positions $y_v = K_v \cdot u_E$, so that the dynamic of the hydraulic drive can be written as an affine in control system:

$$\dot{\boldsymbol{p}} = \boldsymbol{f}(\boldsymbol{p}) + \boldsymbol{g}(\boldsymbol{p}) \, \boldsymbol{u}_{\boldsymbol{E}} \tag{4}$$

with state vector $\boldsymbol{p} = \begin{bmatrix} p_A & p_B \end{bmatrix}^T$, control voltage u_E and the nonlinear vector fields $\boldsymbol{f}(\boldsymbol{p})$ and $\boldsymbol{g}(\boldsymbol{p})$. With substitution $V = K_v K_{oil} Q_{nom} / \sqrt{\Delta p}$, $\boldsymbol{f}(\boldsymbol{p})$ and $\boldsymbol{g}(\boldsymbol{p})$ are given by:

$$\boldsymbol{f}(\boldsymbol{p}) = \begin{bmatrix} -\frac{K_{oil}}{A_A y_y} \left(K_{Lip} \left(p_A - p_B \right) + A_A \dot{y}_y \right) \\ \frac{K_{oil}}{A_B \left(y_{y,max} - y_y \right)} \left(K_{Lip} \left(p_A - p_B \right) + A_B \dot{y}_y \right) \end{bmatrix}$$
(5)

$$\boldsymbol{g}(\boldsymbol{p}) = \begin{cases} \boldsymbol{g}_{+}(\boldsymbol{p}) & \forall \quad \boldsymbol{u}_{E} > \boldsymbol{0} \\ \boldsymbol{g}_{-}(\boldsymbol{p}) & \forall \quad \boldsymbol{u}_{E} < \boldsymbol{0} \end{cases}, \text{ with}$$
(6)

$$\boldsymbol{g}_{+}(\boldsymbol{p}) = \begin{bmatrix} \frac{V}{A_{A}y_{y}} \sqrt{p_{0} - p_{A}} \\ -\frac{V}{A_{B}(y_{y,max} - y_{y})} \sqrt{p_{B} - p_{T}} \end{bmatrix}, \quad \boldsymbol{g}_{-}(\boldsymbol{p}) = \begin{bmatrix} \frac{V}{A_{A}y_{y}} \sqrt{p_{A} - p_{T}} \\ -\frac{V}{A_{B}(y_{y,max} - y_{y})} \sqrt{p_{0} - p_{B}} \end{bmatrix}$$

The output F_a of the hydraulic system is the actuator force generated by pressure differences in the cylinder chambers (7). Its derivative is given by (8).

$$F_a = A_A p_A - A_B p_B \tag{7}$$

$$\dot{F}_{a} = A_{A}\dot{p}_{A} - A_{B}\dot{p}_{B} = \alpha(\boldsymbol{p}) + \beta(\boldsymbol{p}) \cdot \boldsymbol{u}_{E}$$
(8)

The exact input/output linearization of the nonlinear system (8) by the input transformation (9) with the terms $\alpha(\mathbf{p})$ and $\beta(\mathbf{p})$ forms a linearizing feedback with linear input \mathbf{v} .

$$u_{E} = \frac{1}{\beta(\boldsymbol{p})} (\boldsymbol{v} - \alpha(\boldsymbol{p}))$$
(9)

The resulting linearized actuator dynamic is thereby $\dot{F}_a = v$. The remaining inner dynamics of the system (4)-(6) is locally asymptotically stable for $p_0 > p_T$ and $y_y = \int 0$, $y_{y,max} [$. The linearizing terms $\alpha(\mathbf{p})$ and $\beta(\mathbf{p})$ are:

$$\alpha(\boldsymbol{p}) = -\frac{K_{oil}}{Y_{y}} \left(K_{Lip} \left(\boldsymbol{p}_{A} - \boldsymbol{p}_{B} \right) + A_{A} \dot{\boldsymbol{y}}_{y} \right) - \frac{K_{oil}}{Y_{y,max} - Y_{y}} \left(K_{Lip} \left(\boldsymbol{p}_{A} - \boldsymbol{p}_{B} \right) + A_{B} \dot{\boldsymbol{y}}_{y} \right), \tag{10}$$

$$\beta(\boldsymbol{p}) = \begin{cases} \beta_{+}(\boldsymbol{p}) & \forall \quad u_{E} > 0\\ \beta_{-}(\boldsymbol{p}) & \forall \quad u_{E} < 0 \end{cases}$$
(11)

with the direction-dependent values for $\beta(\mathbf{p})$:

$$\beta(\boldsymbol{p}) = \begin{cases} \beta_{+}(\boldsymbol{p}) & \forall \quad u_{E} > 0\\ \beta_{-}(\boldsymbol{p}) & \forall \quad u_{E} < 0 \end{cases}$$
(12)

$$\beta_{+}(\boldsymbol{p}) = \frac{V}{Y_{y}} \sqrt{p_{0} - p_{A}} + \frac{V}{Y_{y,max} - Y_{y}} \sqrt{p_{B} - p_{T}}, \qquad (13)$$

$$\beta_{-}(\boldsymbol{p}) = \frac{V}{Y_{y}}\sqrt{\boldsymbol{p}_{A}-\boldsymbol{p}_{T}} + \frac{V}{Y_{y,max}-Y_{y}}\sqrt{\boldsymbol{p}_{0}-\boldsymbol{p}_{B}}.$$

In the linearization loop, the sign of the term $(\mathbf{v} - \alpha(\mathbf{p}))$ is used to switch between the values of $\beta(\mathbf{p})$. Since $\beta_+(\mathbf{p})$ and $\beta_-(\mathbf{p})$ are both positive terms the switching results only in scaling of the controller output u_E , but not in a change of its sign. Therefore there is no "chattering" of the valve position.

After the exact input/output linearization the resulting linearized plant can be simplified into an integrator and a first order approximation of the valve dynamics with a time constant of $T_e = 2D_v/\omega_0$. For the force control loop a linear proportional feedback controller with the gain k_F was calculated using the amplitude optimum method:

$$k_F = \frac{1}{2T_e} \tag{14}$$

Since the feedback linearization relies on an approximation of the plant the resulting behavior is not exactly linearized, as can be seen in **figure 7**. The performance of the direction dependent force controllers is still comparable, though.



Figure 7: Normalized step responses of the force controlled actuator in a clamped configuration (Controller with $u_E > 0$ (left), with $u_E < 0$ (right)).

4.2. Motion control structure

Typical motion control algorithms for robotic manipulators lead to control in either joint or operational space. The implementation of an operational space control scheme necessitates a precise analytical description of the manipulator dynamics. Since the dynamic relations in the described VGT cannot be feasibly executed in real-time, the motion controller for the VGT structure is implemented as a set of independent joint controllers, controlling the manipulator in joint space, see **figure 9**.

After application of the linearizing force controller the resulting force control loop exhibits quasi-linear first order lag behavior with a time constant of $2T_e$, which allows the design of a classical positioning system in cascaded structure, with a velocity and a position controller. Two control loops where applied to the rod velocity plant, the inner with a proportional and the outer loop with an integral controller. Both were designed using the amplitude optimum criterion, as it allows good reference tracking, along with a desirable disturbance rejection behavior. To evaluate the performance of the velocity control loop, see figure 8 (left), as a result of step inputs of the reference and disturbance signals. It can be seen, that the controlled system shows a well-damped behavior and the rod velocity meets the desired values of 0.2, 0.4, and 0.6 m/s after only 5 ms. Moreover, force disturbances fed into the system as a step input for $t \ge 0.02s$ are quickly eliminated.



Figure 8: Behavior of the controlled (left) and open velocity loop (right) as response to step inputs for the reference and disturbance signals.

For the remaining position control loop a proportional controller was selected, since it is sufficient to provide exact steady state reference tracking for constant inputs even in the presence of output disturbances, due to the plant integrator in the open control loop. The designed controller structure enables the system to follow stepwise position as well as velocity commands for a single joint even in the presence of force disturbances.

Due to the control in joint space the desired end-effector trajectories $\mathbf{x}(t)$ in Cartesian space have to be transformed into the respective joint space trajectories $\mathbf{q}(t) = IKP(\mathbf{x}(t))$ which serve as reference signal for the position controller. To improve tracking performance, the resulting joint space velocities $\dot{\mathbf{q}}$ are calculated from the Cartesian end-effector velocities $\dot{\mathbf{x}}$ via the differential kinematic of the manipulator $\dot{\mathbf{x}} = \mathbf{J}(\mathbf{q})\dot{\mathbf{q}}$. The joint velocities are fed into the velocity controller, so that it becomes active even before a position error accumulates.

Due to the nonlinear coupling between joint motions through the manipulator structure, every motion of single actuators will cause disturbance forces within the manipulator structure. These disturbance forces cause errors in the affected joints which their controllers would pick up and eliminate slowly. The occurring errors in these axes can be largely prevented, by using the knowledge about the dynamic couplings between the motions to approximately compensate disturbance forces. The dynamic behavior of the manipulator is described by the following relation:

$$\boldsymbol{M}_{\boldsymbol{X}} \ddot{\boldsymbol{X}} + \boldsymbol{C}_{\boldsymbol{X}} (\boldsymbol{X}, \dot{\boldsymbol{X}}) \dot{\boldsymbol{X}} + \boldsymbol{g}_{\boldsymbol{X}} (\boldsymbol{X}) = \boldsymbol{F}_{\boldsymbol{X}}$$
(15)

The Cartesian forces necessary for the desired manipulator movement in operational space are projected into joint space via the transposed Jacobian matrix, described by

the manipulator static force relation $\mathbf{F}_x = \mathbf{J}^T \mathbf{F}_q$. Consequently the necessary joint force vector to produce a desired motion in operational space is calculated and fed to the force controller as a reference signal:

$$\boldsymbol{F}_{fwd} = \boldsymbol{J}^{-T} \left(\boldsymbol{M}_{x} \ddot{\boldsymbol{x}} + \boldsymbol{C}_{x} (\boldsymbol{x}, \dot{\boldsymbol{x}}) \dot{\boldsymbol{x}} + \boldsymbol{g}_{x} (\boldsymbol{x}) \right)$$
(16)

With the feedforward values for the joint forces, the desired joint variables q_{fwd} and the joint velocities \dot{q}_{fwd} all necessary feedforward values for each controller in the cascade structure are known. This constitutes the computed torque feedforward controller structure, depicted in figure 9, which was implemented for the motion control of the manipulator. Since all information is fed about the desired trajectory is used in the feedforward signals, thus the feedback controller only needs to suppress unmodelled dynamics and disturbances and can be laid out accordingly.



Figure 9: Motion control structure for the used set of independent joint controllers with feedforward compensation through a computed torque approach.

4.3. Evaluation of the control design

For the evaluation of the control designs, a virtual prototype of a two module VGT, as seen in **figure 10**, was built in MATLAB^(R) to model the hydraulic system and the mechanical structure with all its nonlinearities. Using this model the proposed control structure was compared with well-known control designs for hydraulic systems, like adaptive P-controller with velocity feedforward or adaptive reduced state controllers with hydraulic-model-based feedforward signals. In order to compare these control designs we have chosen a circle-trajectory for the tool center point (TCP) which goes through an area of high sensitivity in the manipulators workspace, which means that an

error of the cylinder lengths results in a high position error of the TCP. This sensitivity is calculated with the 2-norm for each entry of the manipulators inverse Jacobian matrix J^{-1} and is, as can be seen in **figure 11**, the highest in the border of the workspace.



Figure 10: Applied concentrated parameter model for computed torque calculation with masses and inertia of active (a) and passive links (b), joint masses (c) and TCP load mass (d).



Figure 11: Analysis of sensitivity *s* over the relative Cartesian displacements $x^* = x / I_0$ and $y^* = y / I_0$ with I_0 as the basis side length of the octahedron.

Using standard controllers for hydraulic systems, we achieved the best results with an adaptive reduced state controller (position and pressure feedback) with hydraulic-model-based feedforward signal of jointspace velocity and acceleration. One disadvantage of this controller design is that, in contrast to the presented nonlinear architecture, all controller gains are variable and just pay respect to the nonlinearities of the hydraulic system. When compared with the end value of the integral of time absolute error (ITAE) of the position in workspace we can notice, that under application

of the presented nonlinear control architecture the control accuracy can be improved approximately 15 times for a two module VGT, see **figure 12**.





5. Summary

In this paper we presented the mechanical design of a modular hydraulically driven octahedron shaped VGT and an advanced nonlinear control architecture to control the derived manipulators.

Due to the use of hydraulic drives in the presented stiff and light-weight structure a high payload-to-mass ratio can be achieved, which allows highly dynamic motions and a low steady state energy consumption of the system. The composition of more than two modules of the presented design already results in a highly maneuverable, kinematically redundant VGT manipulator, which is capable of dexterous motion. The setup with identical modules and the high common part degree results in low-cost scalability of the manipulator to adapt to specific operation tasks.

The static and dynamic characteristics of the manipulator with the proposed control approach were evaluated in a simulation study using a virtual prototype. The performance benefit obtained by the presented nonlinear control architecture is demonstrated in comparison with well-known linear control designs for hydraulic drives. While the linear controller, which is used as a reference, has to be chosen conservatively enough to tolerate all nonlinearities of the plant, the proposed nonlinear controllers are taking the nonlinear hydraulic and mechanical plant characteristics into

account to ensure optimal control quality all over the workspace of the manipulator in order to ensure access to the full dynamical potential and accurate manipulation tasks.

6. Literature

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7. Equation signs

- *x*, *q* Coordinates in Cartesian and joint space, respectively
- F_{fwd} , \dot{q}_{fwd} , q_{fwd} Feedforward values for motion control
- M_x , C_x , g_x , J Manipulator parameter in Cartesian space

A_A, A_B	Areas of cylinder rod
d	Viscous damping constant of cylinder
F _A , F _B , F _{ext}	Forces: side A and B, external load force
<i>K</i> _F	Controller gain of force controller
K _{Lip}	Inner oil leakage factor
K _v	Valve gain relating uE and yy
т	Effective piston rod mass
р ₀ , р _т , р _А , р _В	Pressures: system, tank, port A and B
Q _{nom} , Q _A , Q _B	Volume flows: nominal valve flow, port A and B
T _e	Time constant of first order approximation of valve dynamics
V	Linearized input of the feedback linearization scheme
y _v	Valve position
y_{y}	Cylinder displacement
Δρ	Nominal pressure drop in valve
α, β	Linearization terms
K _{oil}	Stiffness of hydraulic oil
ω_{o}, D_{v}	Dynamic properties of valve: corner frequency and damping