PRECISE HYDROSTATIC CYLINDER DRIVE WITH INCREASED PRESSURE LEVEL FOR INDUSTRIAL APPLICATIONS

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ABSTRACT

Primary controlled hydrostatic drives, mainly based on speed-variable pumps, are increasingly used in various industrial applications, efficiency and controllability being the main advantages. Still, it is an open question, if the simple substitution of a proportional valve by a speed-variable pump leads in general to a qualitatively comparable system behaviour, or if additional measures are necessary.

Especially, for the 4-quadrant control of single rod cylinders, the use of a servo-driven double-pump is advantageous in terms of efficiency. Nevertheless, the sizing of the pump-unit with respect to the cylinder geometry is a non-trivial task. In the following it will be shown that it is essential for a servo-quality performance, to firstly ensure a flow balance matching the cylinder geometry and secondly to achieve an increased pressure level in order to take over load forces without exceeding pressure limits. To this aim, a control of the sum pressure is known to be appropriate [1]. Introducing another controllable motor-pump-unit allows for coping with both aspects at a time with a purely primary controlled architecture. Within the following a generic systematics of corresponding setups is shown and the example of a cylinder drive for a large machine table proves the servo quality of the solution, making it suitable even for precision machining.

Keywords: Primary control, efficiency, single rod cylinder, 4-quadrant-operation, hydraulic circuit, sum pressure, load stiffness, machine tool

1. INTRODUCTION

Hydraulic cylinders are in general considered as a competitive, efficient and robust component for linear motion in a wide range of actuator applications. Exemplary numbers of the global market size in 2023 show, that cylinders contribute with approximately 18 billion US-$ [2] to nearly 1/3rd of the total world actuator market (~ 55 billion US-$, [3]). Even with these rough, unverified numbers it can be assumed that hydraulic actuation is of considerable importance for present and future applications. However, there are some valid reasons for improvements in hydraulic actuation, most of them resulting from the general ways to generate, distribute and control hydraulic power. If the role of hydrostatics is restricted to power transmission only, while control and power distribution is based on switched power electronics and AC-(servo-)motors, efficiency can be increased significantly.

This leads to the well-known concept of a hydrostatic gear (left in **Figure 1**), where cylinder and speed-variable pump are directly connected in a closed circuit, thus in principle enabling for a full 4-quadrant operability. With no valves involved, a single pump can only operate a double-rod cylinder, while a single-rod cylinder requires a second pump to obtain a matching flow balance (right in **fig. 1**). Ideally, both gears will show no losses, the movement being defined by the pump flow(s) while all internal load conditions result from external forces. Given the dynamics of a servo-motor this arrangement may obviously be a considerable step towards improved efficiency and controllability.

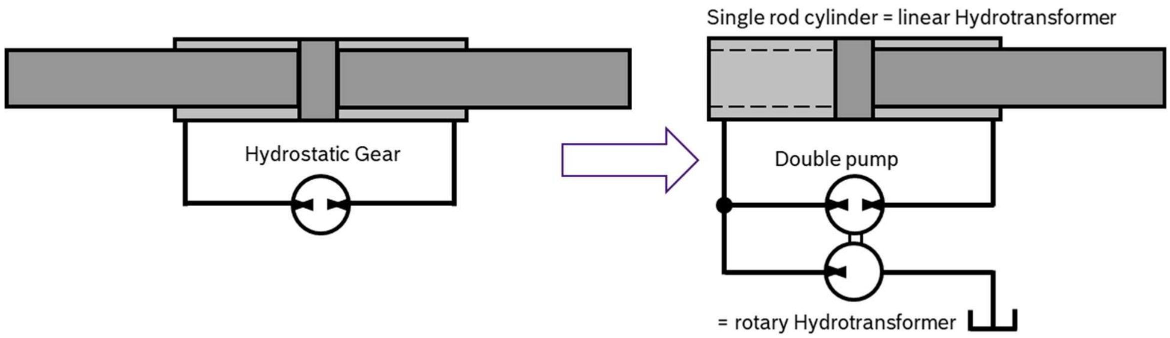

Figure 1: Basic concept of a hydrostatic gear with double- and single-rod-cylinder

Single-rod cylinders are commonly preferred to double rod-cylinders due to many good reasons (except controllability), so while the double-pump approach seems to be a consistent idea to obtain continuous 4-quadrant-ability, there are 2 issues of this kind of gear that deserve attention.

- Firstly, if pump flows do not meet the area ratio of the cylinder, the pressure level in the circuit will change during and because of a movement.[1] It also will because of leakage. The effects on the operability turn out to be considerable.
- Secondly, while pressure on rod side is higher than on piston side, which is valid for a wide range of load conditions, the power flow of the pumps will be in opposite direction, making an efficient power exchange between both units advisable, e.g. by mechanical coupling.

Both issues shall be treated in the following under various aspects.

2. TOPOLOGY OF ELECTRO-HYDROSTATIC LINEAR GEARS

2.1. Basic Concept, basic traits and basic prerequisites

Some relevant traits of hydrostatic gears show clearly in a model-based comparison with a valve-controlled cylinder drive. The scope of models may be minimal but should consider pump leakage.

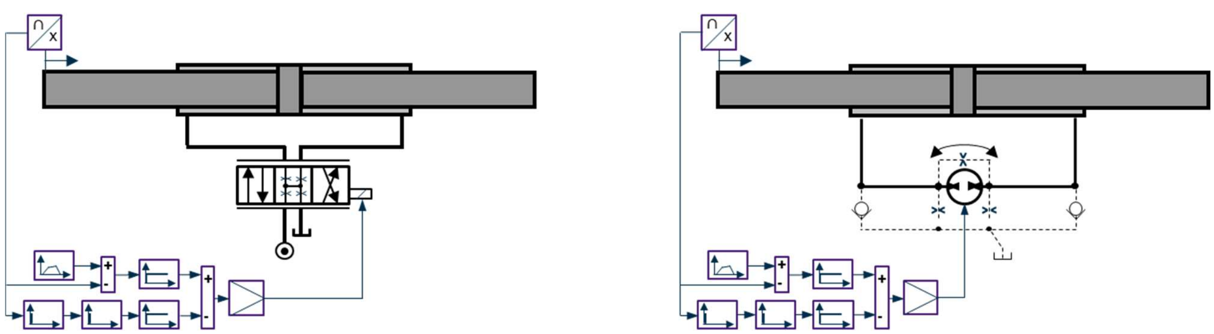

Figure 2: Schemes of simulated drive models, pump with leakage and suction valves.

Because of external leakage, pressures in the pump driven axis will tend to decrease to reservoir level in stand-still and without load. Any motion of the pump or of the cylinder will lead to a pressure increase in one and a (undesirable) pressure drop below reservoir level in the other cylinder chamber. Check valves (see **fig. 2**) are commonly used to ensure filling of the chamber and protecting components. Unfortunately, in doing so, they introduce discontinuous state changes showing in poor, unreproducible controlled performance, which shall be illustrated by **Fig. 3**. In initial state, shown in the left scheme in **fig. 3**, with pump in standstill and no load applied, chamber pressures are down to reservoir level. The check valves are closed and can therefore be neglected in the scheme. An external

[1] Example: if the additional pump covers more than the share of the rod area, pressures will increase during extraction and decrease during retraction, no matter what the load conditions are.

load displaces the piston and one chamber pressure decreases, opening the respective check-valve and connecting the chamber to reservoir, as shown in the right scheme in **fig. 3**. In this state, the axis behaves like a plunger cylinder and structurally has lost its full 4-quadrant-ability.

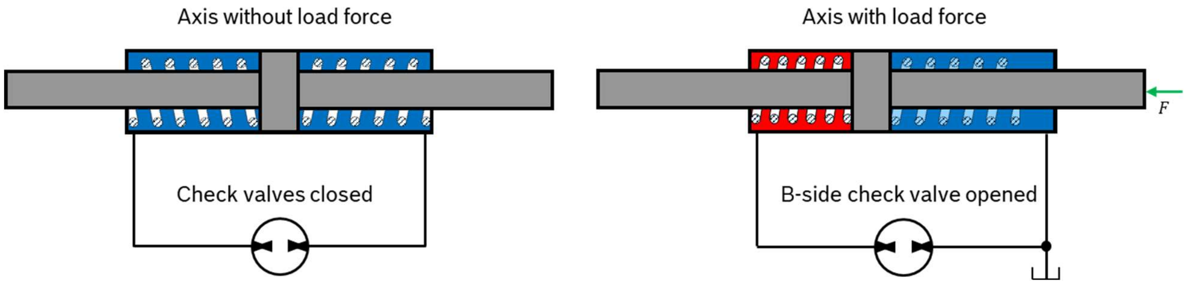

Figure 3: State of axis and check-valves with and without load force.
Schematic springs shown for illustration purposes only.

While cylinder chambers are a closed space, the fluid inside acts like a spring attached to the piston, represented by symbolic springs **fig. 3**. This may clarify that the opened check-valve keeps pressure constant and disengages one "spring" completely, thus decreasing load stiffness and dynamics of the gear considerably. Oscillations at this operation point cause valve chattering and discontinuous state-changes[2], showing in a simulation with the following main model and simulation parameters:

- Cylinder dimensions: 50/36-36/600 (in mm), moved mass 300 kg.
- Valve characteristics: Nominal flow 20 l/min @ 70 bar, 2% neg. overlap, supply 210 bar.
- Pump size 11.5 ccm, leakage coefficient 0.0015 l/min/bar, tank pressure 2.5 bar.
- Target values for position interpolation: travel distance 400 mm, velocity 400 mm/s.
- Load is applied near extracted position, resulting in about 150 bar of pressure difference.

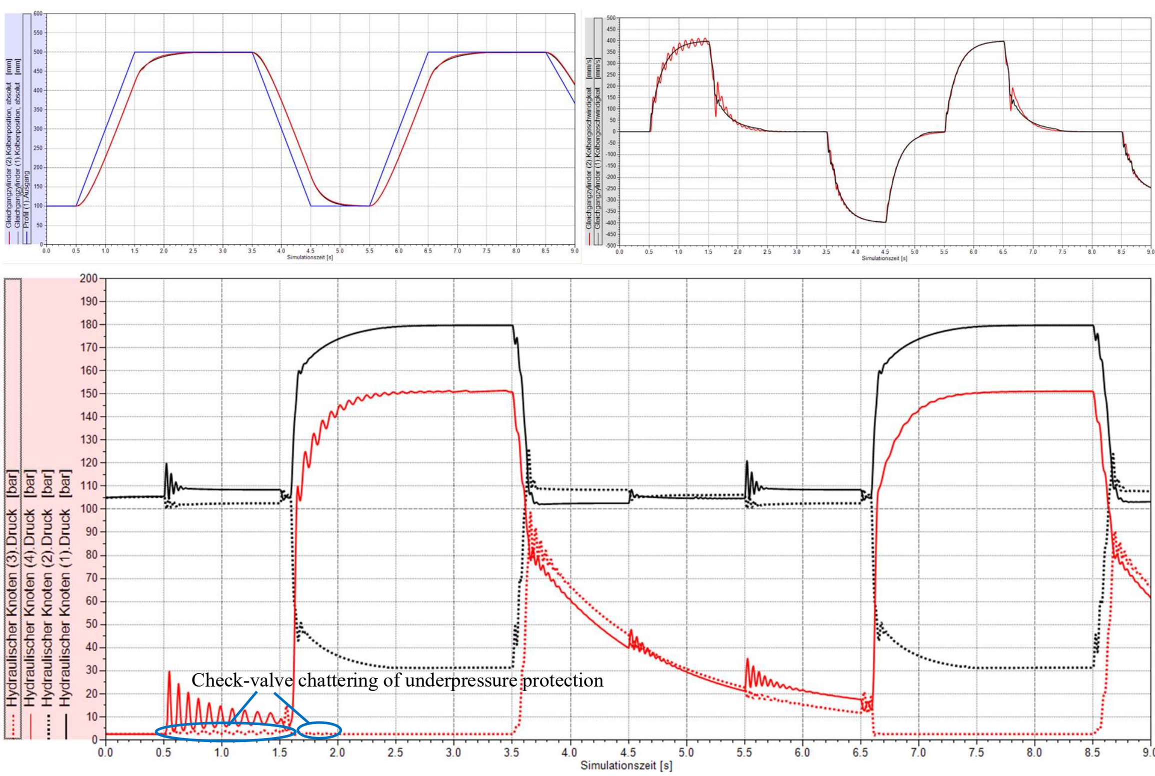

Figure 4: Simulation results for valve- and pump-controlled axes in an exemplary press process.

[2] The lift-off of the unloaded spring shown in **Fig.6** will not happen in the shown way, as the cylinder will always be oil-filled from the reservoir. Nevertheless, it helps to illustrate, that the gear changes its state in a discontinuous way.

Fig. 4 shows the simulation results in 3 consecutive plots: position (top left), velocities (top right), and pressures (bottom). Values of valve-controlled axis are black, pump-controlled axis in red, set-values are blue. Pressures of the retracting chambers are dotted. The following findings are important:
Position (top left):

- Both types of axes are controllable and reach the demanded positions, following the set-value-path with a contouring error corresponding to the desired loop gain of the position controller.

Velocity (top right):

- Progression is strongly different in the two cycles.
- If velocity were relevant for axis performance, the behaviour of the pump-driven axis would most probably be inacceptable in the first cycle, as acceleration is periodically changing sign.

Pressures (bottom):

- With valve-control chamber pressures are in average half of supply pressure (105 bar), while pressures of the pump-controlled axis are defined by the highest load pressure, resulting in a level of around 85 bar after decrease of the load force which then decreases because of leakage.
- Both types of axes show considerable oscillations when accelerating or when load is changed.
- Oscillations of the pump-controlled axis are significantly higher. One reason are the small hydraulic losses, which increase efficiency, but decrease dissipation of oscillation energy.
- Eigenfrequency with valve-control is 24 to 25 Hz, corresponding to an effective load stiffness of roughly 7.4 kN/mm, which is alike for pump control with pressures at around 85 bar.
- Eigenfrequency of the pump-controlled axis decreases with the pressure level to 18 Hz at 20 bar, indicating only 3.8 kN/mm stiffness due to dependency of bulk modulus on pressure. Additionally, it is significantly varying between the two cycles.

In sum, the valve-controlled axis shows the desired full 4-quadrant-ability, resulting from the constantly increased pressure level. The drive state is always steady in the nominal force range. The load stiffness is up to the best possible level and reproducible in every cycle.
In the pump-driven axis the actuator is structurally not able to provide energy to both chambers at the same time, so that pressures cannot be changed without changing load conditions or movement. With constant load and motion, pressures will decrease due to leakage, until at least one chamber pressure is down to reservoir level, potentially leading to undefined, unsteady state changes, where the axis switches from 2-quadrant- to 4-quadrant-ability and back.
In this configuration the axis is not on a level that could be achieved with a valve control in terms of controlled accuracy, not to mention an electro-mechanic axis. The loop-gain of the position controller, which is at its limit in this simulation, is a factor 10 to 20 lower than what could be expected using e.g., a pre-tensioned spindle. Even more, state- and force-control are very sensitive to harsh changes in drive stiffness.[3] The conclusion is, that for maxing out the full abilities of a hydrostatic gear it must be pre-tensioned by increasing pressure level. Means to achieve this and their systematic implementation, using primary servo-control only, shall be presented in chapter 2.3.

2.2. Consideration of Single-Rod Cylinder and Double Pump as Hydrostatic Transformers

In **fig. 1** the (single-rod) cylinder of the introduced hydrostatic gear is introduced as a hydraulic transformer. This interpretation may be based on the following considerations:
If there are no other forces (friction, load or inertia) are acting on the cylinder except pressure forces, then their sum must be zero, so it is $p_A \cdot A_A = p_B \cdot A_B$. Therefore, it counts for the chamber pressures:

$$\Rightarrow p_B = \alpha \cdot p_A \quad with\ \alpha = {}^{A_A}/_{A_B} \quad and\ F = 0 \tag{1}$$

[3] There are of course means to avoid the chattering state under external force, e.g. by actively opening decompression valves, but this is only suitable as long as the operating conditions always lead to load conditions that definitely allow for 2-quadrant-operation. Furthermore, precise mechanical elements like spindle nuts are commonly also pretensioned.

If the cylinder moves with the velocity v, the following displacement flows occur at the flanks of the piston: $Q_A = v \cdot A_A$ and $Q_B = v \cdot A_B$. The relation of the flows is therefore also given by the area ratio:

$$Q_A = \alpha \cdot Q_B \quad with\ \alpha = {}^{A_A}/_{A_B} \tag{2}$$

In the given ideally load-free state the power balance over the piston is:

$$p_A \cdot Q_A - p_B \cdot Q_B = 0 \tag{3}$$

As pressures and flows are both transformed by the area-ratio α between port A and B, and ingoing power at A equals outgoing power at B, the single-rod cylinder indeed *is* a hydraulic transformer.

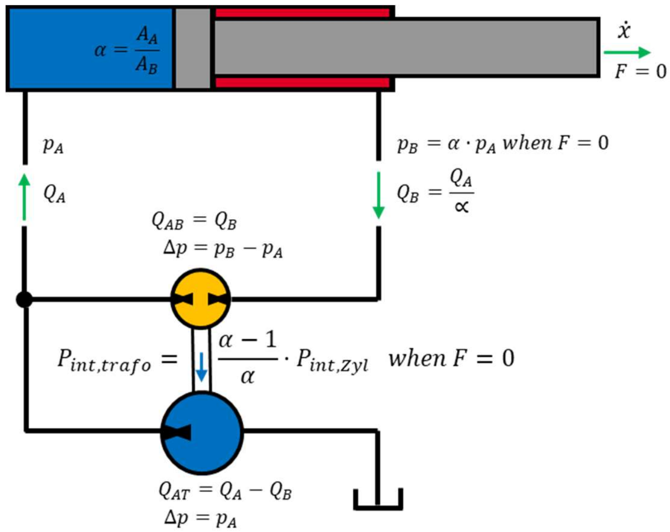

Figure 5: Power balance of the hydrostatic gear with single rod cylinder: internal power split with no external load applied.

Concerning the double pump configuration in **fig. 5** the power balance can be obtained similarly. In steady state, pump flows (marked with suffixes "AT" and "AB") must match the cylinder flows:

$$Q_{AB} \stackrel{\text{def}}{=} Q_B \tag{4a}$$
$$Q_{AB} + Q_{AT} \stackrel{\text{def}}{=} Q_A \tag{4b}$$
$$\Rightarrow Q_{AT} = Q_A - Q_B = Q_A \cdot \frac{\alpha - 1}{\alpha} \tag{4c}$$

Two mechanically coupled pumps have identic speed, so the relation of pump sizes must be:

$$\beta \stackrel{\text{def}}{=} \frac{V_{AT}}{V_{AB}} = \frac{Q_{AT}}{Q_{AB}} \stackrel{\text{def}}{=} \alpha - 1 \tag{5}$$

Corresponding to the considerations on the cylinder, the load balance of the double-pump-unit is:

$$\Delta p_{AB} = p_B - p_A \tag{6a}$$
$$\Delta p_{AT} = p_A - p_T \tag{6b}$$

With reservoir pressure p_T and external torque M being zero, the power balance reads:

$$-\Delta p_{AT} \cdot Q_{AT} + \Delta p_{AB} \cdot Q_{AB} = 0 \tag{7}$$

Comparing equ. 3 and equ. 7 it shows that *a double-pump also acts as a hydraulic transformer and the combination with a matching single-rod cylinder results in a balanced functional hydrostatic gear.* The internal power split of the double-pump is in fixed relation to the internal cylinder power.

2.3. Basic Topologies of the primary controlled Single-Rod Cylinder

Based on the previous considerations the design of the hydraulic circuit involving double-pump and single-rod cylinder shall be generalized in the following. Basically, a hydrostatic gear always requires the control of flow between three ports: cylinder chamber A and B and reservoir T, which may accordingly involve 3 displacement-controlled flows: A-T, A-B, and B-T. **Fig. 6** gives an overview.

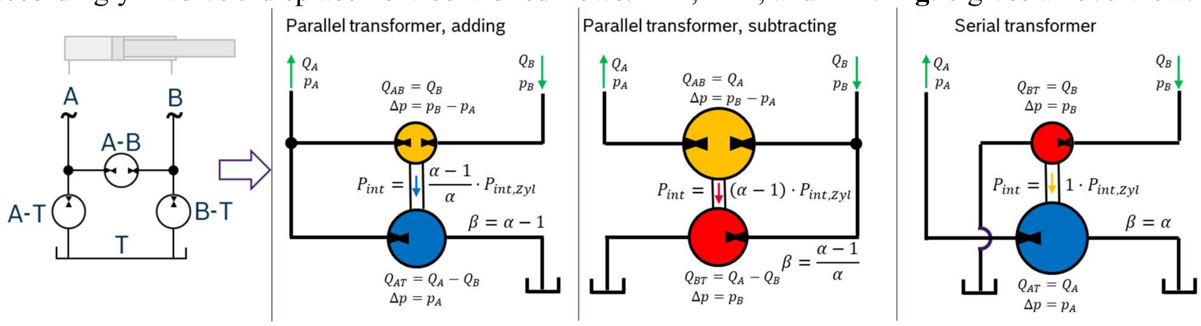

Figure 6: Systematics of suitable double-pump configurations, derived from generic function scheme.

Three suitable double-pump configurations (see **fig. 6**.) can be derived from the three basic functions. The internal power balances of these configurations and the relation β of pump sizes matching the area ratio α of the cylinder are also included in **fig. 6**. Still, these consistent hydrostatic gear configurations for single-rod cylinders require exactly balanced displacement units and cannot provide a well-defined pressure level in the gear. This problem will be addressed in the next chapter.

3. REALIZATION OF CONTINUOUS 4-QUADRANT-OPERABILITY

3.1. The internal pressure level as a separate Control Task for Cylinder Drives

The main task of a hydraulic cylinder is to realize motion and forces. Efficacious in this sense is only the sum of pressure forces of the two chambers, the value of a particular pressure being unimportant:

$$F_{p,eff} = p_A \cdot A_A - p_B \cdot A_B \tag{8}$$

In chapter 2.1 on the contrary it was shown, that for the operability of the gear an increased pressure level is crucial. The inevitable leakage of real components means an ongoing loss of pre-tension, which, however, can be compensated by an actively controlled flow from the reservoir to the gear. It is clearly target of the presented work, to not use any valve- or throttle-control, but to stay in the concept of direct pump control. To refill the gear and keep up pressure level will therefore require:

- at least one displacement unit connected to reservoir.
- a second degree of freedom to separately control the refilling flow to chamber A and B.

The following shall show, how this additional effort can reasonably be "double used".

3.2. Flow Matching and Pre-Loading – a Multi-variable Control Task

All combinations of double-pumps require a certain relation between the area ratio of the cylinder and the ratio of pump sizes. With real components an adequate match of geometries is unrealistic. Standardised diameters such as 100/70 (α = 1.960784…), won´t be matched by fixed displacement double-pumps. The problem can be solved by adding a third pump, driven by a separate motor, so introducing a second source of energy and a second degree of freedom, e.g. to compensate for leakage. **Fig. 7** shows an exemplary arrangement, including all relevant flows and speeds necessary for covering the displacement flows of the cylinder and also gives the power balances of the hydrostatic gear with additional motor-pump unit. Indeed, the share of power taken over by the additional unit matches with the share of B-side flow this unit covers. The additional motor pump unit may therefore be very small in comparison to the main unit. It shall be named "flow matching unit" (FMU) in the following. Coupling the two electric drives with a DC-bus adds an electric power split, so that the complete arrangement has again no external power demand if load and losses of the gear are zero.

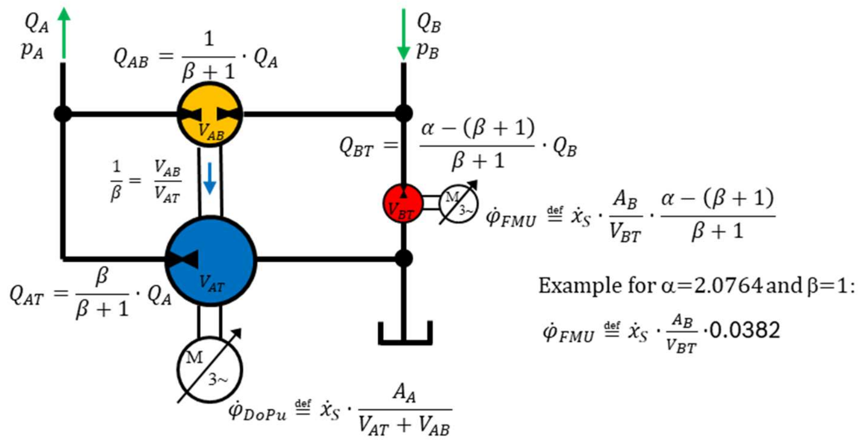

Figure 7: Adding parallel transformer with additional BT-pump as "flow matching" unit (FMU).

Fig. 8 gives an overview of possible configurations of FMU´s corresponding to the three basic types of transformers, marking their basic function as supplement or complement to the double-pump.

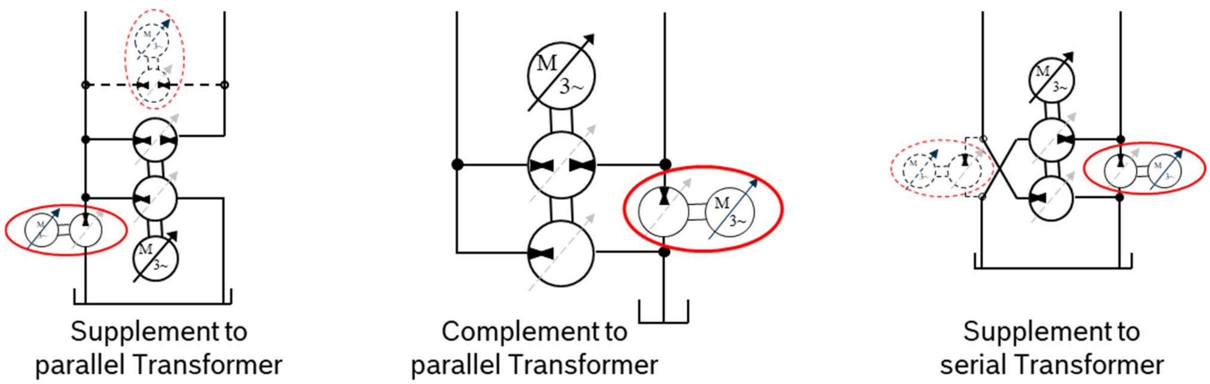

Figure 8: Combinations of "flow matching" units with different types of transformers.

3.3. Sum Pressure Control

It was shown previously that an increased pressure level is crucial for controllability and performance and that load changes should simultaneous increase and decrease pressures in the both chambers.

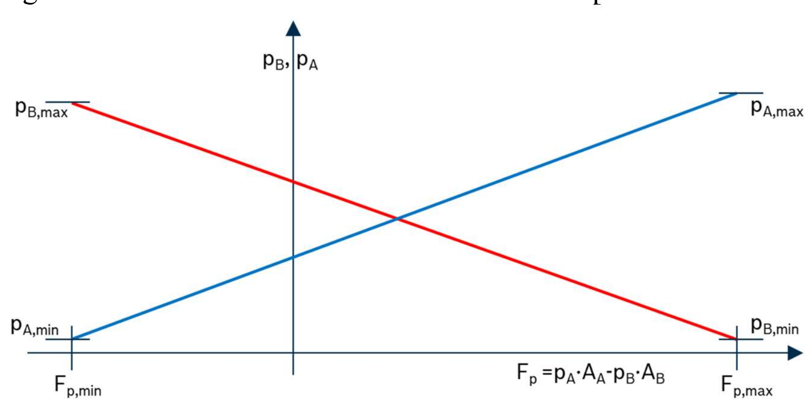

Figure 9: Chamber pressures as a function of force and pressure limits.

Maximum pressure in one and minimum pressure in the other chamber should always be reached simultaneously, thus realizing maximum pushing resp. pulling force, while pressures change contrarywise and linear with load force. This characteristic is shown in **fig. 9** and can be written as:

$$F_{press,max} = p_{A,max} \cdot A_A - p_{B,min} \cdot A_B \tag{9a}$$
$$F_{pull,max} = p_{A,min} \cdot A_A - p_{B,max} \cdot A_B \tag{9b}$$

With symmetric pressure limits $p_{A,max}=p_{B,max}=p_{max}$ and $p_{A,min}=p_{B,min}=p_{min}$ this can be achieved by:

$$p_A + p_B = p_{max} + p_{min} \tag{10}$$

Introducing a "sum pressure" p_{Sum} this can be realized by the following controller:

$$p_{Sum,Set} = p_{max} + p_{min} \tag{11a}$$
$$p_{Sum,act} = p_A + p_B \tag{11b}$$
$$Q_T = \mathrm{f}\left(p_{Sum,Set} - p_{Sum,act}, t\right) \tag{11c}$$

The controller outputs a demanded flow from the reservoir into the gear. The distribution of this flow to the two chambers must be determined separately, favourably in a way that prevents an influence of the sum pressure controller on the axis motion. The controller could be e.g. a PI-controller.

3.4. Outline for a generic model-based Multi-variable Control

The setups for hydrostatic gears considered previously show a variety that is demanding for the design of (control-)software. **Fig. 10** shows a modular software concept minimizing the adaptation efforts. The proposal for the functional architecture is based on mechatronic considerations and reflects the physical interrelationships of cylinder and drives by implementing a control-layer, the inverse [flow → linear motion] transformation layer and the inverse [rotatory motion → flow] transformation layer, representing control, cylinder and motor-pump-units. Modifications in the axis-configuration concern mainly pump arrangements, so restricting software variations to the corresponding functional layer.

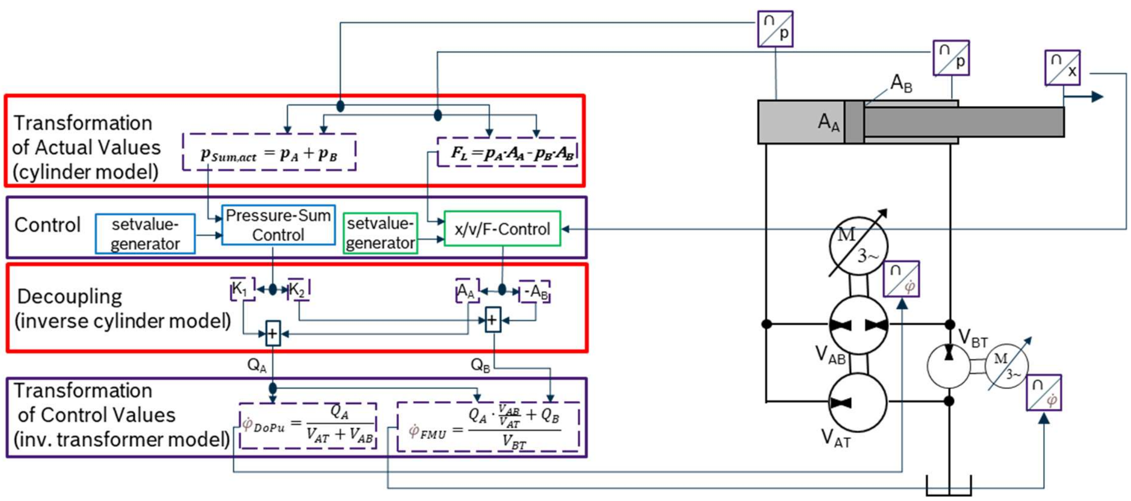

Figure 10: Functional Layers of a Model-Based Multi-variable Axis Control

4. LARGE-SCALE GRINDING MACHINE – AN APPLICATION EXAMPLE

Grinding is of crucial importance for the manufacturing of machine parts prone to sliding and rolling movement as well as for sealing surfaces. Machining of ship engines, machine tool guides etc. may give an idea of the size of grinding machines needed to this aim. **Fig. 11** shows a machine, which is an example of the "WayGrind"-series offered by german Waldrich Coburg GmbH.

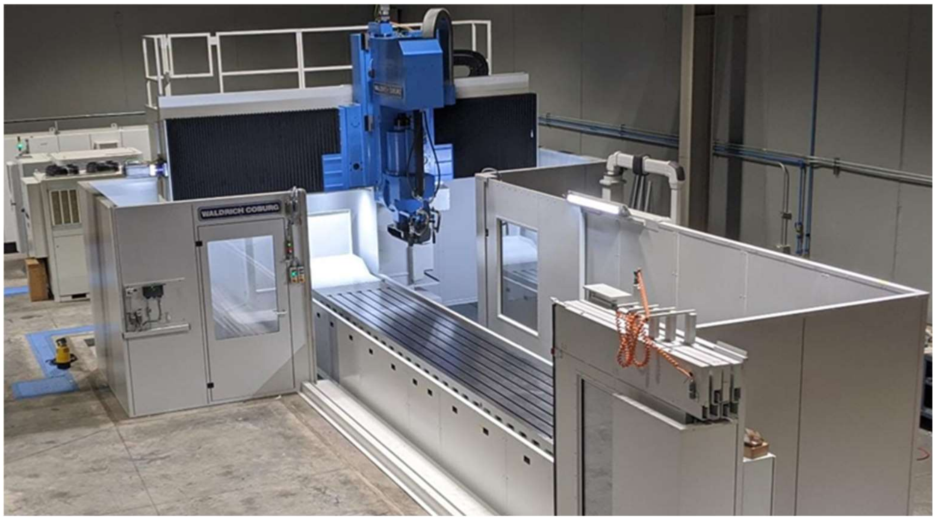

Figure 11: Grinding machine by Waldrich Coburg Company (6 m grinding length)

The size of the grinding table may give an impression of the drive stroke needed and the moved masses, even without workpiece. While processing forces are not that high in grinding, axis load is dominated by acceleration and friction. With moderate load and large stroke, bending resonance becomes a major issue for spindle drives, while hydraulic cylinders may be a cost-efficient, robust alternative. At the same time, the working conditions make the use of valve control undesirable for efficiency and thermal reasons. Altogether, an electro-hydrostatic linear drive is a superior solution.

4.1. Machine Configuration

The WayGrind series of Waldrich Coburg comprises grinding machines of various sizes in portal configuration with setup lengths of up to 15 m and clearance height and width of up to 3.5 m and 4 m. The workpiece is mounted on a grinding table, which realizes the working stroke (x-axis), while the grinding support carries and positions main drive and grinding disc in y- and z direction.

Figure 12: Table, portal and grinding support of machine chosen as application example.

A medium size WayGrind 1000 FC of this series shall be presented as an application example. The machine dimensions are 8,800 x 6,400 x 4,200 mm (LxWxH). Table mass is approx. 5.5 to, the table dimensions are 2,000x1,000 mm, carrying a workpiece with maximum weight of 4 to. Grinding width and height are 1,200 mm x 1,000 mm. Travelling distances of the axes: 3,000 mm (X), 3,140 mm (Y) and 1,000 mm (Z). The grinding disc diameter may be up to 600 mm, the spindle power up to 22 kW. The feed velocity of the table ranges between 2 m/min and 45 m/min (750 mm/s), while the minimal infeed of 0.001 mm of y- and z-axis may give an impression of the machine precision.

4.2. Drive Configuration for the Grinding Table

For driving the machining table (x-axis) of the machine, Bosch Rexroth has developed and built a preloaded electro-hydrostatic linear drive, consisting of single-rod cylinder, serial hydrotransformer and supplementary flow matching unit as shown in **fig. 13**.
Cylinder dimensions are 80/56-3000 (α = 1.961), the eigenfrequency of the unloaded table is around 3 Hz. The sizes of the double-pump are 100.2 ccm and 64.7 ccm respectively, so β = 1.549. The flow-matching unit size is 20.1 ccm. Motor speeds are nominal 2,260 rpm and 1,530 rpm at v = 750 mm/s. The transformer is built in serial configuration for the main reason, that the same type of internal gear pump in open circuit version can be used for all 3 displacement units. The pressure level is increased to 90 bar of sum pressure, resulting in 30.4 and 59.6 bar chamber pressure at the load free cylinder. The double-pump drive implements position control and active damping, while sum-pressure control is implemented by the FMU-drive. Cross-communication of the drives runs over SERCOS CCD.

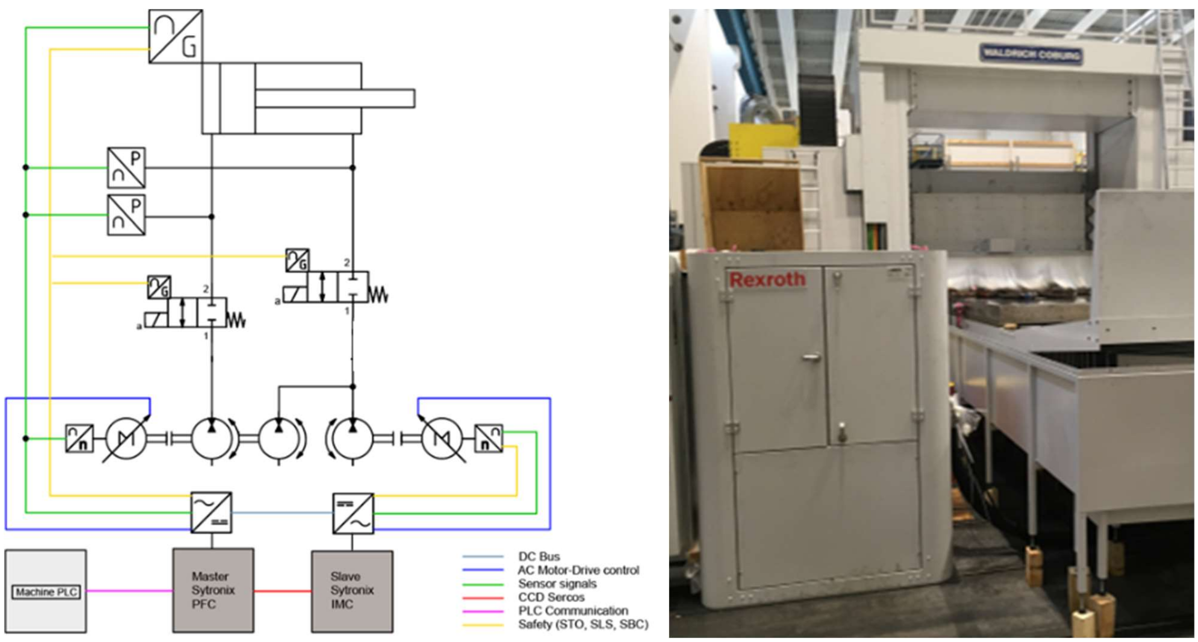

Figure 13: Principal hydraulic scheme of hydrostatic gear and hydraulic cabinet of grinding table drive.

4.3. Performance of the Hydrostatic Drive within the Grinding Cycle

During the grinding process the table with the working-piece is moved with constant velocity between two end-positions that ensure that the table has reached constant speed when the grinding wheel engages. The uniformity of the movement ensures process quality, while the distance needed to accelerate from standstill to constant velocity determines productivity and the maximal size of the working piece. In the following example the travel distance is 2.5 m The x-axis accelerates to v = 500 mm/s with a nominal acceleration of 1.0 m/s^2. **Fig. 14** shows the measured performance of the drive in a time plot of velocity and pressures. The nominal acceleration and deceleration distance given in the set-value path is 125 mm each and sufficient to reach feed and zero velocity. High static friction initially excites the axis and causes an overshoot of 6.6 % in velocity, leading to another travel

of 235 mm necessary to stabilize the axis to an accuracy < 1%.
After stabilization the <u>velocity deviation is 0.84 mm/s in average</u> (0.168 %) and 3.7 mm/s at maximum (0.74 %). Even given the high-quality table guides, the accuracy of the axis is remarkable. The stability shown also in pressures underlines the meaning of pre-tension for 4-quadrant-ability.

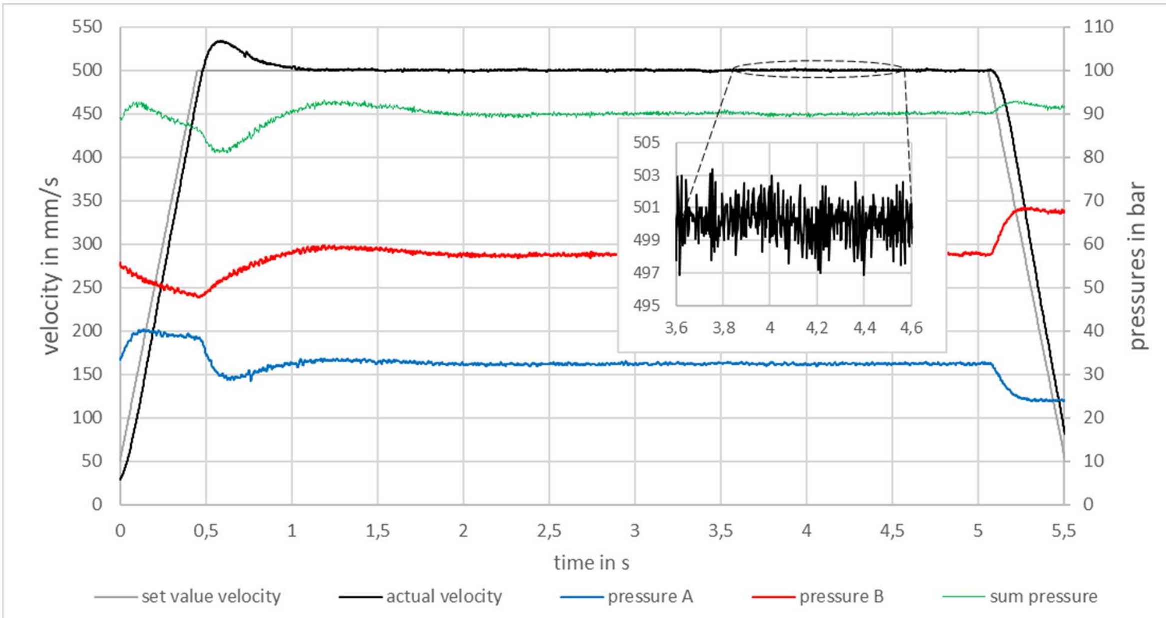

Figure 14: Velocity and pressures of grinding table drive during grinding process over 2.5 m travel.

5. CONCLUSION AND OUTLOOK

This paper concerns the design of primary controlled cylinder drives for applications that require continuous 4-quadrant ability. It focuses on the consistent use of electric power distribution and speed-variable motor-pump units. Hydraulics is used for power transmission only, but not for control. Double-pumps can be used as hydro transformer, thus being an efficient means for the build-up of hydrostatic gears involving single-rod cylinders. The basic variance of hydraulic setups is restricted to two transformer types, serial and parallel transformer. In general, hydrostatic gears are prone the loss of pressure level over time due to leakage. An additional motor-pump unit allows for actively increasing the pressure level, the control of the sum of pressures being an appropriate approach. With this, both chamber pressures may change concurrently with varying load force, thus preventing from reaching pressure limits in the complete range of operation and maximizing the stiffness of the drive. Moreover, the additional motor-pump unit allows for compensating non-matching flow balances of double-pump and cylinder. The number of circuit variations combining such a flow matching unit with a double-pump is limited to three basic configurations, two of them having a variation each. A modular approach to a generic "mechatronic software architecture" restricts software-variance due to different actuator configurations to one single layer, involving an inverse model of the pump network. A hydrostatic gear developed accordingly is able to meet all expectations in terms of functionality, accuracy and efficiency in the demanding application of an advanced feed of a large-scale grinding machine. It drives a grinding table of 5.5 to with a maximum velocity of 750 mm/s over 3 m of stroke. The shown approach can be seen structurally consistent and overall promising. Open questions concern a more detailed consideration of structural variations under the aspect of power balances, especially of the flow matching units. This should lay basis for a systematic selection and sizing of components and a systematic system design based on application demands. Moreover, a systematic examination of the implemented multi-variable control in terms of stability, robustness, and static and dynamic decoupling should contribute substantially to a successful use in demanding applications.

NOMENCLATURE

A	area	m^2, cm^2
α	area ratio	-
β	ratio of displacement volumes of double pump	-
f_0	eigenfrequency	Hz
F	force	N, kN
m	mass, weight	kg, to
M	Torque	Nm
n	speed	rpm
p	pressure	N/m^2, bar
P	power	W
Q	flow	m^3/s, l/min
$\dot{\varphi}$	angular speed	rad/s
v, $\dot{x}$	velocity	m/s, mm/s
V	volume	m^3, l, ccm
x	position	m, mm
ω_0	natural frequency	1/s

Cyl	cylinder
DoPu	double-pump
FMU	flow matching unit
A,B,T	Indices for hydraulic ports and related values
act	Abbreviation for “actual” in Indices
int	Abbreviation for “internal” in Indices

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