Variable-Speed Pump Drive System for a 5000 kN Ring Expander

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Abstract

In cooperation with Bosch Rexroth and MAE Maschinen- und Apparatebau Götzen, a new hydraulic drive system has been designed using the example of a ring expander. Ring expanders are part of a process chain for the manufacture of high-precision rolled rings in particular for high-alloy steels. In this application a large differential cylinder acts on an expanding cone. This in turn acts via 9 or 12 segments, so-called expanding shoes, on a rolled ring to calibrate the inside diameter. Here, similar to a bending process, the ring is expanded beyond its yield point and brought to the required dimension.

The concept is based on a frequency-controlled servo-asynchronous motor with a drive power of 65 kW. An axial piston pump rotating clockwise and a second axial piston pump rotating anti-clockwise are mounted on a common drive shaft. Both pumps are designed as variable pumps with a mechanical torque controller. The swash plates are designed for two-quadrant operation. The reliable reduced set-up speed is attained without additional valve control by means of the reliable limitation of the servo-motor speed according to Cat. 3 via a module in the frequency converter.

"With the variable-speed pump drives, Rexroth now offers pump control in a highly dynamic intelligent electrical drive which only generates the volume flow actually required. Reduced speed during breaks in the cycle or when not running at full power mean a significant drop in the energy required, in noise emissions and in hydraulic power losses. The pump drive increases the speed of the highly dynamic motors as required as soon as the hydraulic system needs more power. All components come from the standard Rexroth product portfolio." /1/

KEYWORDS: ring expander, servo-asynchronous motor, variable-speed pump drives, axial piston pump

1. Introduction

A key topic at the Hannover Trade Fair in 2011 was the energy efficiency of hydraulic drives. Under the keywords "BlueHydraulics, variable-speed drives, Sytronix – for "Smart interplay of Hydraulics and Electronics", ECOdraulics, EPAI hybrid systems, SINAMICS servo-pump", concepts for variable-speed pump drives were presented by all leading hydraulics providers. This concept is really not new. Already in the middle of the nineties, injection moulding machines for plastics with speed-controlled pump drives (two-quadrant operation, pressure and volume control) were implemented by the company Battenfeld in cooperation with Voith/Eckerle. The systems are usually based on pump types such as gear pumps or vane pumps with a max. operating pressure of 200 .... 320 bar.

In the field of heavy mechanical engineering for presses and other forming units, hydraulic pressures of 350 to 600 bar are customary today for cost-optimised designs. Radial or axial piston pumps are typically used for these purposes. In cooperation with Bosch Rexroth and MAE Maschinen- und Apparatebau Götzen, a new hydraulic drive system has been designed for the operation of a ring expander. Ring expanders are part of a process chain for the manufacture of high-precision rolled rings in particular for high-alloy steels. In this application a large differential cylinder acts on an expanding cone. This in turn acts via 9 or 12 segments, so-called expanding shoes, on a rolled ring to calibrate the inside diameter. Here, similar to a bending process, the ring is expanded beyond its yield point and brought to the required dimension, taking into consideration the elastic spring-back (**Figure 1**).

Figure 1: Complete-assembled ring expander

The aim of the development is a significant increase in the efficiency of the drive power, simplification of the control circuit for controlling the position of the expanding cone and consequently the simple and quick commissioning and integration of a reliable reduced set-up speed (Cat. 3) in the frequency converter.

2. State of the art

The state of the art is represented by the familiar valve-based displacement control system with a high-pressure axial piston variable pump, low-pressure rapid-traverse pump with regenerative circuit, separate pump for maintaining the control oil pressure, proportional decompression valve, various directional valves, cooling and filter circuit and a safety valve control for reliable reduced set-up speeds.

By means of controlling the volumetric flow of the axial piston pump with a proportional valve (HS4 controller), it has already been possible to achieve significant energy savings as compared to proportional control valves. Variable-displacement pumps are swivelled back to nearly 0 l/min during operational breaks and on attaining the required operating pressure. Nevertheless, the use of an asynchronous motor with a constant mains frequency means that no-load losses of up to 30 % of the rated motor power arise even when the plant is in idle operation. During longer interruptions the motors can be switched off and, whenever necessary, be switched on again via reduced-voltage starting devices.

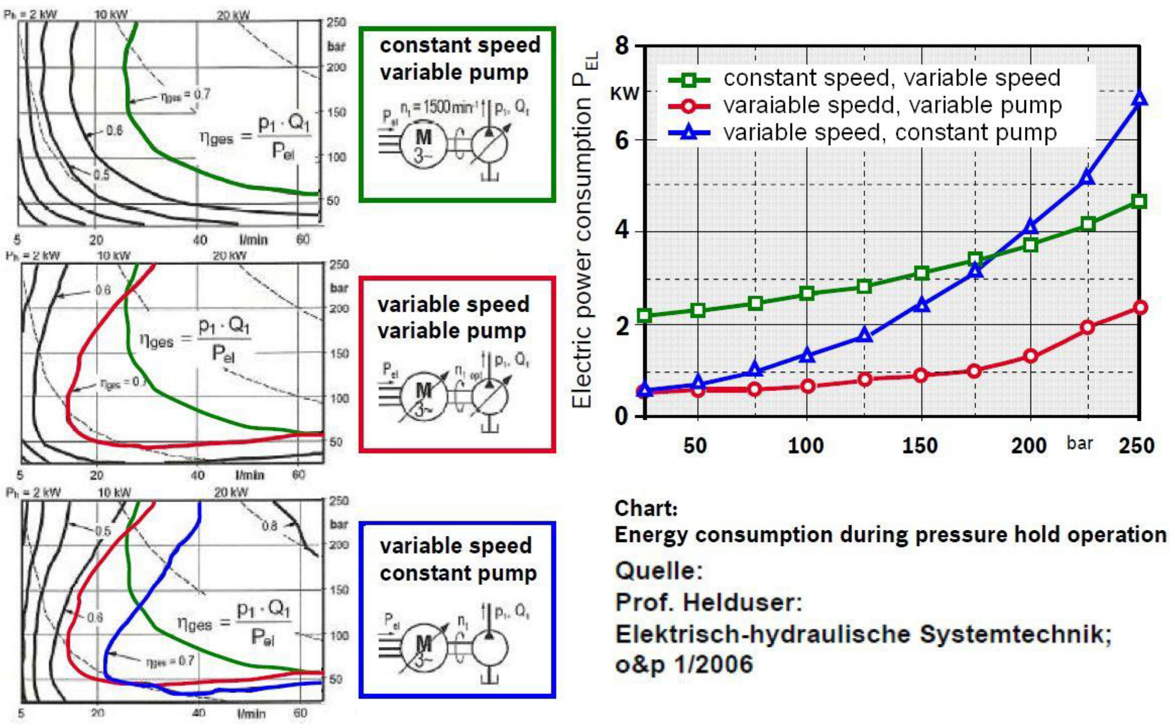

Figure 2: Energy consumption during pressure hold operation /2/, /3/

As early as 2006 Professor Helduser examined the various drive systems for their energy efficiency /2/. With regard to the pressure hold function, i.e. the regulation of a constant pressure with compensation of any possible leak oil losses, the combination of frequency-controlled motor and variable-displacement pump shows the best efficiency (**Figure 2**).

3. New drive concept for expander hydraulics

The price of a converter with servo-motor rises disproportionately with the drive power and/or with the required torque at the drive shaft. For the design of a speed-controlled pump drive, three different systems are possible for reducing the required torque at the servo-motor (**Figure 3**).

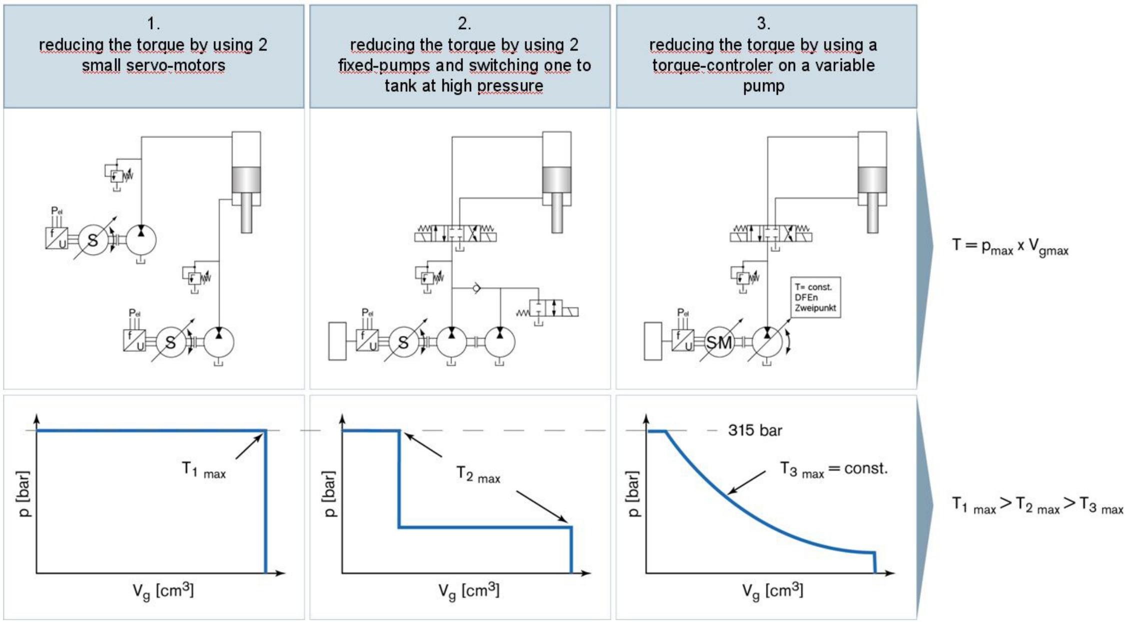

Figure 3: Torque reduction possibilities (costs) /3/

In case 1 the necessary drive power is distributed over two drives for the piston and rod ends of the cylinder. This, however, causes the total installed power to increase and two independent control circuits have to be harmonized with each other. In the second case, two fixed-flow pumps are each operated at a common drive shaft. When a specified operating pressure (normally the required travel pressure for rapid speed) is exceeded, the low-pressure pump is switched to pressureless circulation and the torque is thus limited to the drive torque of the high-pressure pump. There are therefore two operating conditions: a high operating pressure with a small oil flow for the load cycle and a lower pressure with a high oil flow for the rapid speed. In the third case, a variable-displacement pump with a mechanical torque controller is used instead of constant pumps. In this version the displacement of the pump is continuously adapted to the current operating pressure more or less according to the maximum available drive torque. This enables valuable seconds to be saved for many hot forming processes. In particular, the pressure contact times and consequently the heat losses and also the wear of the tools can be reduced.

The concept implemented for the expander is based on a servo-asynchronous motor with a drive power of 65 kW. An axial piston pump rotating clockwise with a mechanical torque controller and a second axial piston pump rotating anti-clockwise are mounted on a common drive shaft (**Figure 4**).

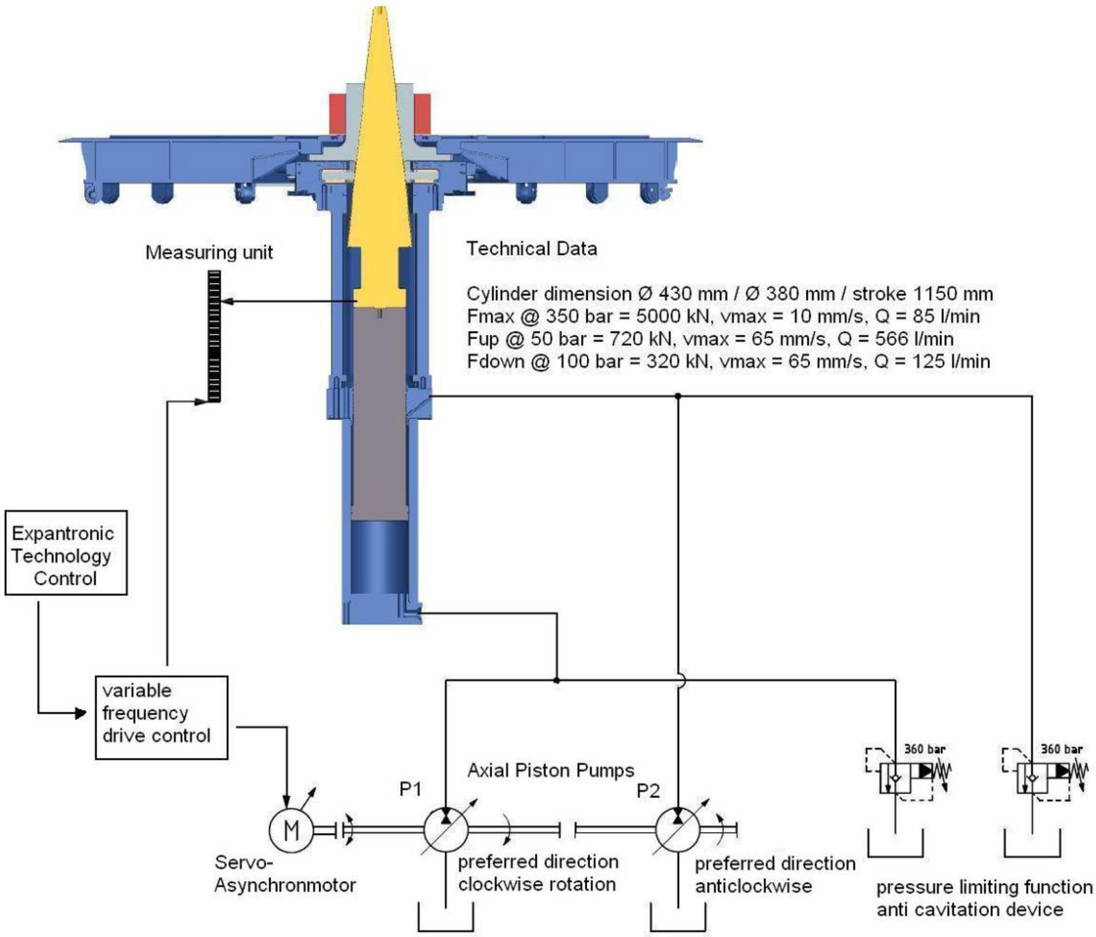

Figure 4: Schematic diagram of hydraulic ring expander

The torque controller ensures that, independently of the speed and the current operating pressure, the maximum torque at the servo-motor is not exceeded (not a power controller). Both pumps are reversed in their turning direction depending on the direction of movement of the cylinder. Decompression of the cylinder chambers is directly done via the relevant pumps. The cylinder is exclusively positioned via the speed control system of the motor. For this, the absolute value of the cylinder position (accuracy 0.01 mm) is fed back into the frequency drive controller.

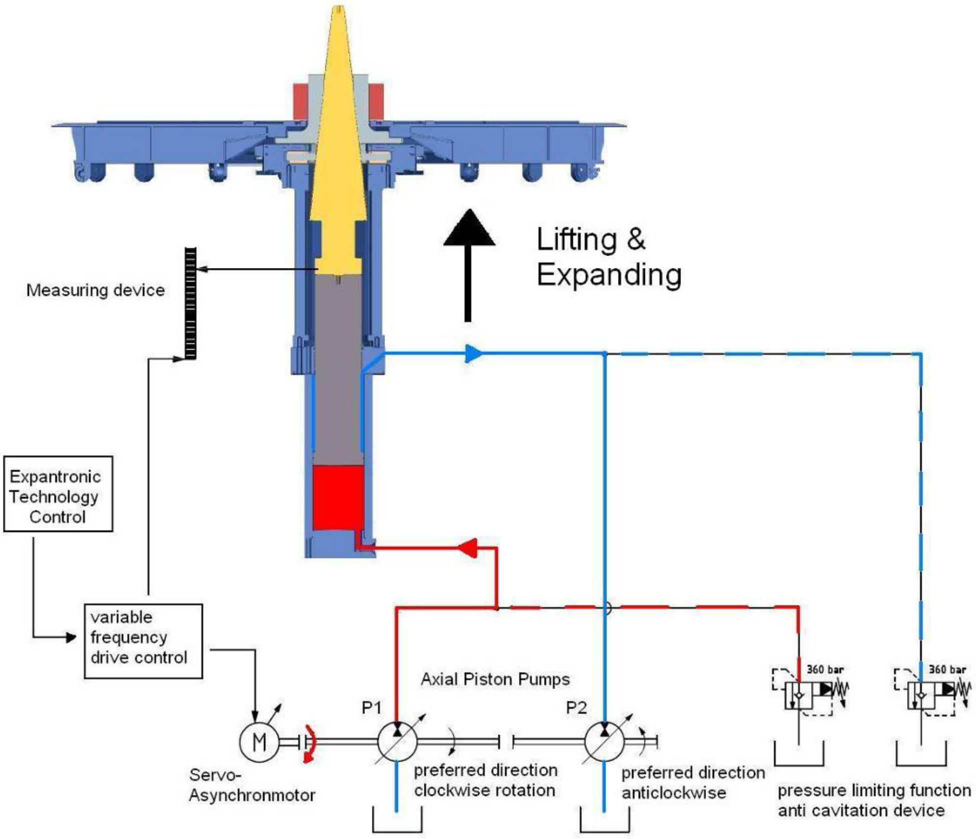

Figure 5: Motion sequence for lifting / rapid speed and for load stroke / expanding

Figure 5 shows the actuation of the servo-motor for the upwards movement at rapid speed and the operating movement "Expanding of the ring" under load. The transition from rapid speed to working stroke takes place continuously according to the available drive torque. Pump P1 conveys the oil from the tank into the piston end of the cylinder, while Pump P2 conveys the oil from the rod end into the tank. Pump P1 is sufficiently dimensioned for supplying the necessary delivery volume for the rapid speed movement. Upon increasing the operating pressure to expand the ring, the pump is swivelled back via the torque controller to limit the power input of the servo-motor. The delivery volume of the second pump is reduced via the mechanical stroke limitation in accordance with the area ratio between the piston and rod areas.

Figure 6 shows the actuation of the servo-motor for the downwards movement of the expanding cone. Following the decompression phase of the main piston via Pump P1, Pump P2 delivers into the rod area of the cylinder and Pump P1 conveys the oil out of the piston area into the tank.

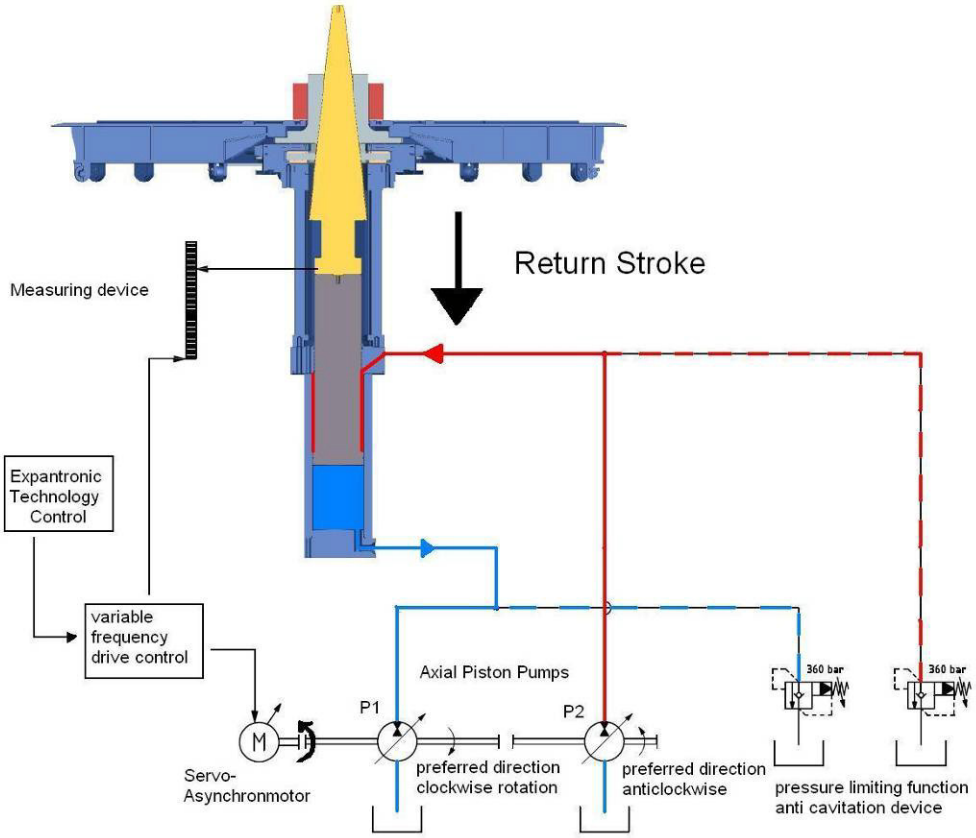

Figure 6: Motion sequence for retraction / lowering

4. Technical details of the system

4.1. Design of the unit

Figure 7 shows the structure of the hydraulic unit. For this control concept it is essential for the unit to be positioned as close as possible to the cylinder in order to obtain adequate control dynamics. For this reason, the most compact possible design was chosen.

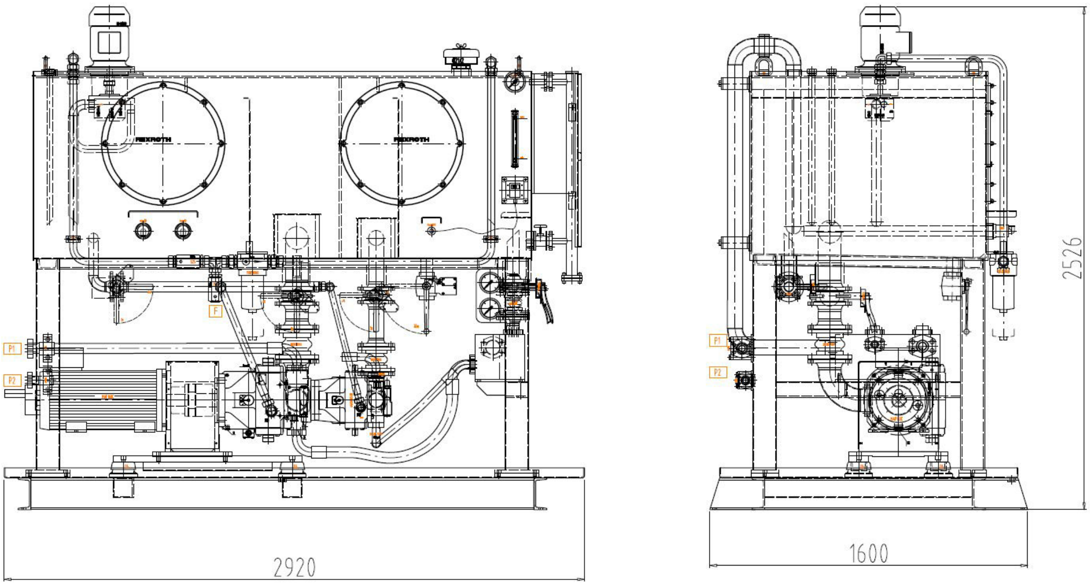

Figure 7: Structure of the hydraulic unit

4.2. Axial piston pumps A4VSO LR2D

The rated pump sizes are adapted to the cylinder dimensions. Pump P1 has the rated size 355 ccm for the piston chamber and Pump P2, which is smaller as regards the area ratio, has rated size 125 ccm for the rod end (**Figure 8**). This "splitting" is done very easily and very precisely via the Vgmax and Vgmin variability of variable-displacement pumps. In this case, the capacity of Pump P2 was limited to 78 ccm. The front pump rotates clockwise. At the rear an anticlockwise-rotating pump is mounted on the same shaft. This is a completely new and revolutionary design because axial piston pumps, when actually operating as pumps, have hitherto had only one defined direction of rotation. When acting as a motor, they do of course have alternating directions of rotation, but a hydraulic motor is always under pressure. Here, however, we are talking about a pump function (with a small additional motor component). For this, the pump must be able to change its direction of rotation: The front, clockwise-rotating unit becomes an anticlockwise-rotating unit and the rear, anticlockwise-rotating unit becomes clockwise-rotating. The direction of pressure does not change during this process. This is a great challenge for pump manufacturers.

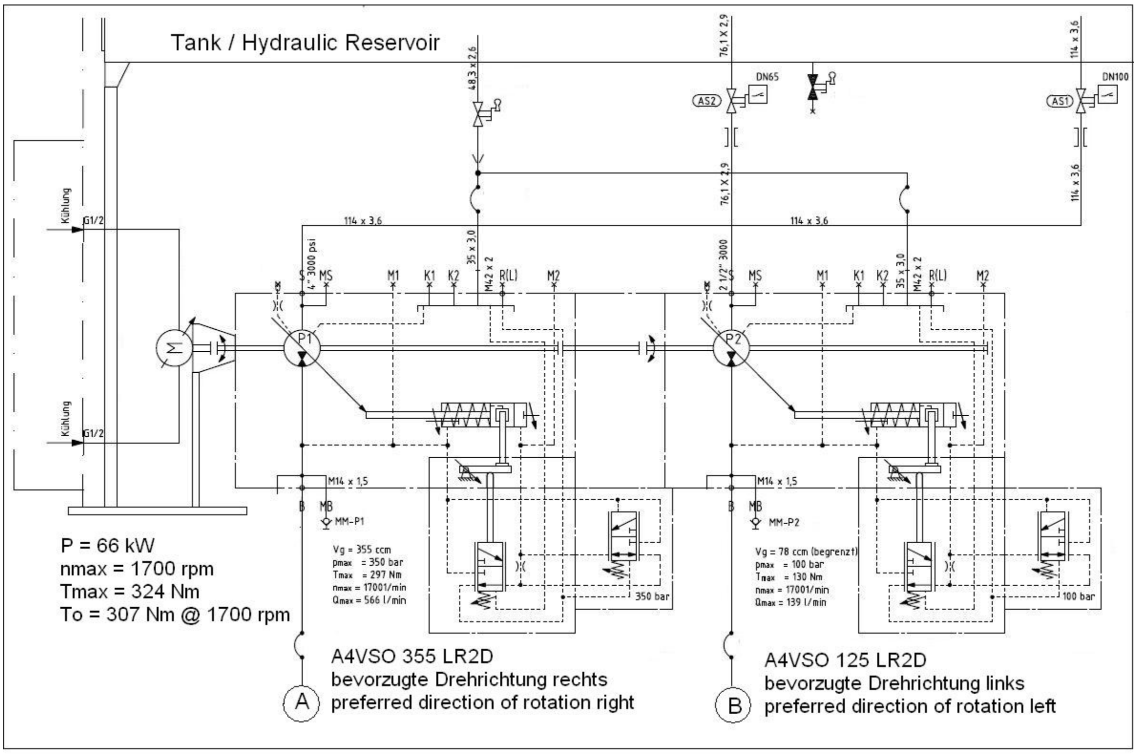

Figure 8: Arrangement of the axial piston pumps

The pump actuation functions in two-quadrant operation, though not by being swivelled through zero but instead by means of a reversal of direction of rotation. For this operating mode the distributor plate needs to have pre-compression slots at the high-pressure end in both directions. Nevertheless, the distributor plate has still to be twisted since there is only one pressure direction with a predominant pump function. Everything that is applicable to the high-pressure end is likewise applicable to the low-pressure and suction-pressure ends (slots and twisting). Further requirements made on the pump design are:

- Speed-controlled operation up to 1,800 revolutions per minute
- The pump combination must be able to operate at low speeds
- The pump combination must be able to be brought to a stop (though without being switched off).

4.3. Servo-asynchronous motor Intradyn MAF160C-0200

The liquid-cooled motors in the MAF series are particularly suitable for applications demanding maximum torques in minimum amounts of space. At the same time, the integrated cooling system design ensures the thermal isolation of motor and machine and therefore maximum handling precision. The reason for the choice with liquid cooling instead of external fan cooling is illustrated in **Table 1**. The same motor with liquid cooling has approx. 20 % less moment of inertia and about 30 % higher nominal torque. **Figure 9** shows the motor characteristic of the selected motor MAF 160C-0200 with liquid cooling. The operating range of the motor is within the speed rang 0 to 1800 rev./min. This means that the full drive torque is available at any time.

Type	Rated speed	Maximum speed	Rated torque	Maximum torque	Rated power	Rated current	Maximum current	Moment of inertia
	n_N	n_{Max}	M_N	M_{Max}	P_N	I_N	I_{Max}	J
	[1/min]	[1/min]	[Nm]	[Nm]	[kW]	[A]	[A]	[kgm²]
MAF160C-0200	2000	6000	285	677,4	59,7	136	290,7	0,26
MAD160C-0200	2000	6000	210	494,2	44	93,9	182,4	0,311

Table 1: Comparison of motor ratings between water-cooled (MAF) and air-cooled (MAD) /4/

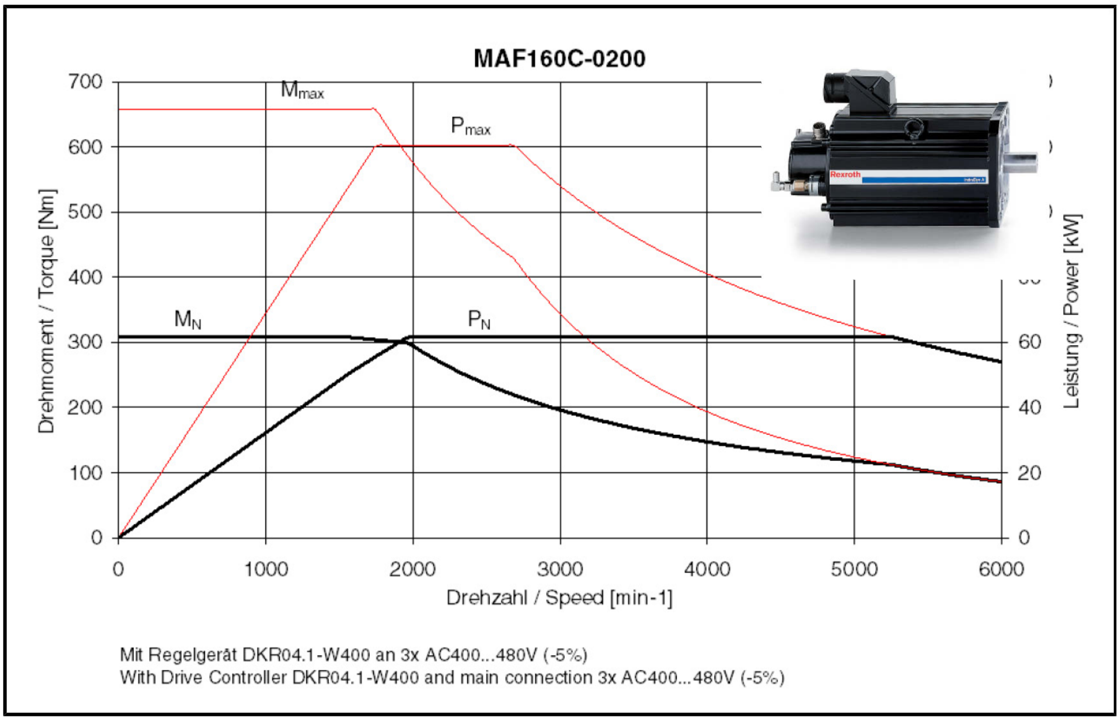

Figure 9: Motor characteristic curve Bosch Rexroth Intradyn MAF160C-0200 /4/

4.4. Typical working cycle of an expanding operation

Figure 10 shows a typical sequence of an expanding operation. First the cylinder lifts with fast approve speed until the expanding shoes get into contact with the inner diameter of the ring. During the expanding of the ring the pressure is built up. After calibration to the exact size there is a holding time to allow the material to stabilize. After decompression the expander returns back into the home position with high speed.

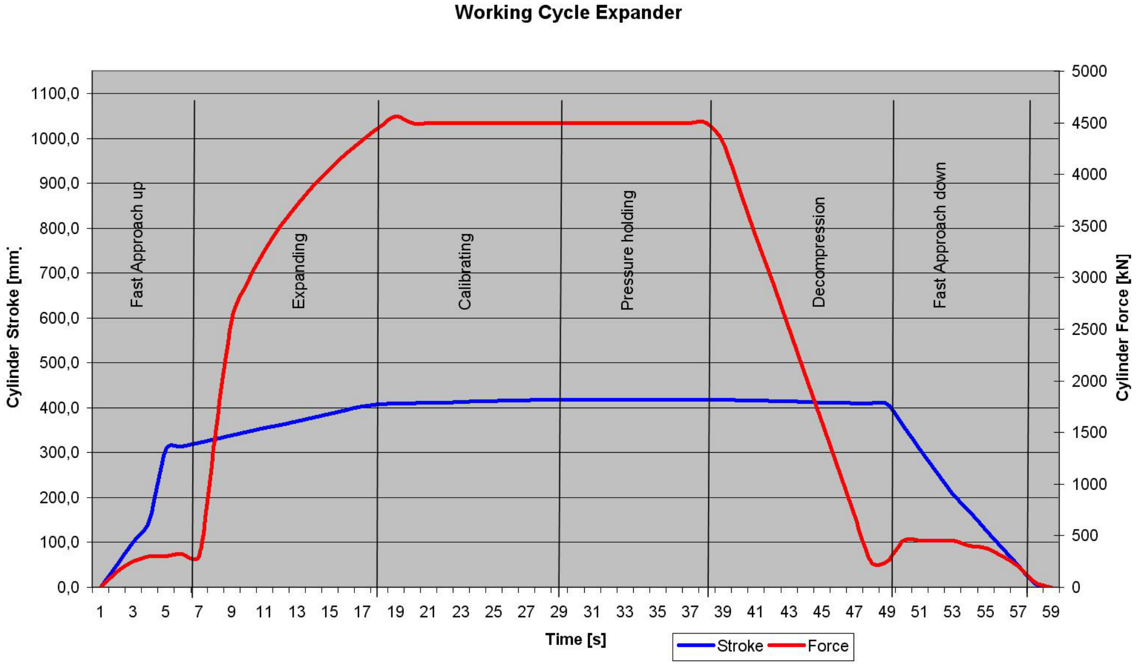

Figure 10: Cycle sequence of an expanding operation

5. Evaluation of advantages and disadvantages

What is really convincing is the highly minimalistic functional concept. All travelling movements and the necessary operating conditions for the forming process (Figure 10) are controlled via the speed control of the drive motor. The executed hydraulic drive is able to function fully without electrically actuated valves, i.e. the heating and abrasion of the oil caused by it flowing over the valve edges is greatly reduced. Thanks to the high efficiency, a cooling system for the hydraulic oil is not needed. In comparison with purely electrical systems, however, the advantages of the high power density of hydraulics are retained. The reliable set-up speed is attained through the reliable limitation of the servo-motor speed according to Cat. 3 via a module in the frequency converter. In the pauses between cycles and under part-load operation, the speed of the servo-motor is lowered down or brought to a halt by the frequency converter.

Depending on the working cycle, 40 to 70 % of the energy consumption can be saved. Since, theoretically, no noise arises when the servo-motor is at a standstill, the average sound pressure level measured, for example, over one working shift, is considerably reduced. Since no specific hydraulic knowledge is required, the start-up of the control circuit (speed control and monitoring of the drive via the frequency converter) can be carried out independently by the "electrical people".

However, compared with a central hydraulic system, there are some disadvantages or limitations of this control concept that need to be considered. Thus, for example, parallel movements of different actuators such as cylinders, oil motors, etc. are not possible, or only partly with additional controls. The attainable dynamics are influenced by the distance between the unit and the actuator. In heavy mechanical engineering, such a set-up close to the drive is not always possible. For highly dynamic processes, the tried-and-tested valve technology of "full acceleration and brake" is definitely superior to direct pump control.

In view of the advantages listed, however, a detailed check should be made whether a variable-speed pump drive system is applicable for new hydraulic systems or also for the modernisation of existing systems.

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