# A new Approach on a Hydrostatic Motor for Applications in Mobile Cranes

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Mobile hydraulic linear actuators are a fixed part of many applications. Especially in mobile cranes, they are used for the movement of the booms and are characterized with a light power to weight ratio. The kinematics can be seen as a restriction of linear motor in mobile cranes. On one side the possible range of the motion and on the other side the unfavourable constellations of the triangle of force are essentially restrictions in the construction of mobile cranes. For the avoidance of these restrictions exists approaches by in the joints arranged hydrostatic rotational motors. At present these solutions fails by a to high weight to force ratio, or they give no benefit in the kinematics. In this article, a new approach of a hydrostatic rotational motor will be presented, which is characterized by low weight and high torque. By the possibility to rotate endless these rotation hydrostatic motors are predestined as a direct drive in the joints of mobile cranes, to get new possibilities in the kinematics and the construction of the booms.

**Keywords:** mobile cranes, hydrostatic rotary motor
**Target audience:** mobile hydraulics, components

## 1 Introduction

The kinematics of mobile cranes, especially in applications for the agriculture and forestry industries, also at mobile concrete pumps is mostly restricted by the linear motors of the booms. The use of lever to increase the swivel range of single booms for more freedom in motion leads to a higher weight of the crane and to unfavourable forces at the linear motors (see figure 1).

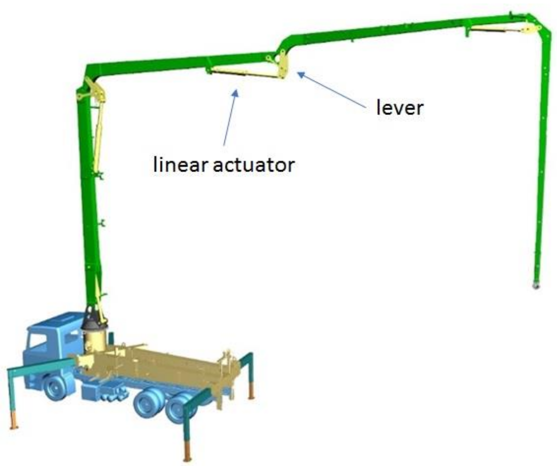


*Figure 1: Principle sketch of implementation of linear actuators on mobile cranes, source: own illustration*

The special requirements of the concrete industry represent a big challenge on the kinematics of automotive concrete pumps. When a customer buys an automotive concrete pump, he must decide the length of the crane and how the crane is folded on the vehicle. Depending on the type of folding, there are advantages when working outdoors or when working in buildings or in pits. The booms of concrete pumps can be fold in two different types. By the z-fold, the booms are applied either from above or from below to the previous boom (see Figure. 2 left). The roll folding type applied all booms from below to the previous boom (see Figure 2, right). Depending on the type of folding, the kinematics result in different working areas of the concrete pump crane.

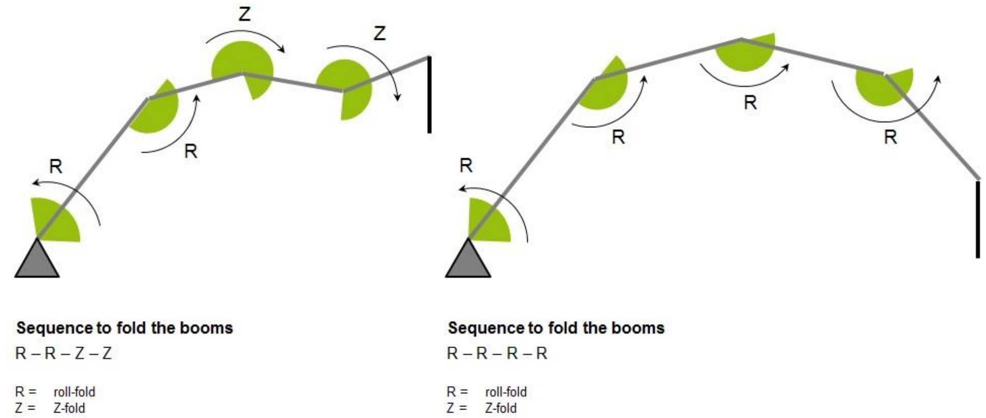


*Figure 2: Different sequences to fold the booms of mobile cranes, source: own illustration*

By replacing the linear motors in the individual booms through rotary motors in the respective joints of the booms many advantages are given in the kinematics. The main advantage is, that the customer must make no decision which type of folding he wants to use. Depending on the design, the individual booms can cover large angular areas, or even rotate endless. Thus, there are no restrictions in the kinematics. However, the implementation of rotary motors in the joints of the booms of cranes cannot be implemented without problems. The greatest problem is the required torque in the joints, which is necessary to move the booms. Depending on the joint in which a rotary motor is used, are torques of about 50 kNm upwards to about 1500 kNm necessary. Thus, motors with high torque are required, but with little weight that the structure of the booms is not overload and the required torques of the previous joints not increase disproportionately.

Due to the required power weight, only the use of hydrostatic motors is possible. There are already existing concepts, but these concepts are limited on the one hand with the maximum torque, the power weight or the maximum swivel angle.

One concept uses a worm gear in combination with a hydraulic motor /1/. With this combination the desired swivel angles can be achieved, but the required torque is limited by the power weight of worm gear and hydraulic motor. A concept with a spur gear unit and a hydraulic motor was also presented /2/. In this case is a limitation of the maximum gear ration of the spur gear. A further possibility is the use of a steep-thread drive /3/. However, this results in limitations in the kinematics and in the power weight (see Table 1). The above-described concepts can only be used in the last joint of the booms of automobile concrete pumps, because the power weight or the maximal achievable torque is a limit in the use in all joints of the booms of automobile concrete pumps.

The in this article presented hydrostatic motor don't have the above described limitations in the kinematics, power weight or in the achievable torque and can be used in all joints of the booms of automotive concrete pumps /4/.

| *hydrostatic rotary motor* | torque in kNm | max operating pressure in bar | weight in kg | operating angle in ° |
|---|---|---|---|---|
| Eckart SM4/300<br>see: https://www.eckart-hydraulics.com/ | 85 | 250 | 1456 | 360 |
| Hägglunds Drive CB280<br>see: https://www.boschrexroth.com | 92 | 350 | 705 | ∞ |
| direct drive, Schwing GmbH | 120 | 300 | 120 | ∞ |

*Table 1: Comparison of different construction principles of hydrostatic rotary motors.*

In Table 1, three hydrostatic rotary motors are compared with respect to power weight and swivel angle. The motor marketed by the company Eckart /5/ as the construction of Martin /6/ which is also described by Holmes /7/ is also mentioned in the patent of the Putzmeister Engineering GmbH /3/. In the points of power weight and kinematics the hydrostatic rotary motor of the company Hägglunds is close to the here presented direct drive of the company Schwing GmbH.

## 2 Direct Drive

### 2.1 Construction

The design of the direct drive is shown in Figure 3. The rotary motor consists of two hydraulic cylinders (2) with the same area ratio, each of them has plane gear teeth on the two piston surfaces. The housings (1) are likewise designed with plane gear teeth on the planar surface. Both pistons are mechanically connected to each other via a splined shaft (3). Either the toothings of two associated housing or the pistons have an offset by half a tooth width. (see Figure 4). By the splined shaft have both pistons an offset of a quarter tooth. While the offset by a half tooth is necessary to generate the torque of one cylinder, is the offset by a quarter of a tooth is necessary to generate torque in the death point of the cylinders during the reversal of the direction.

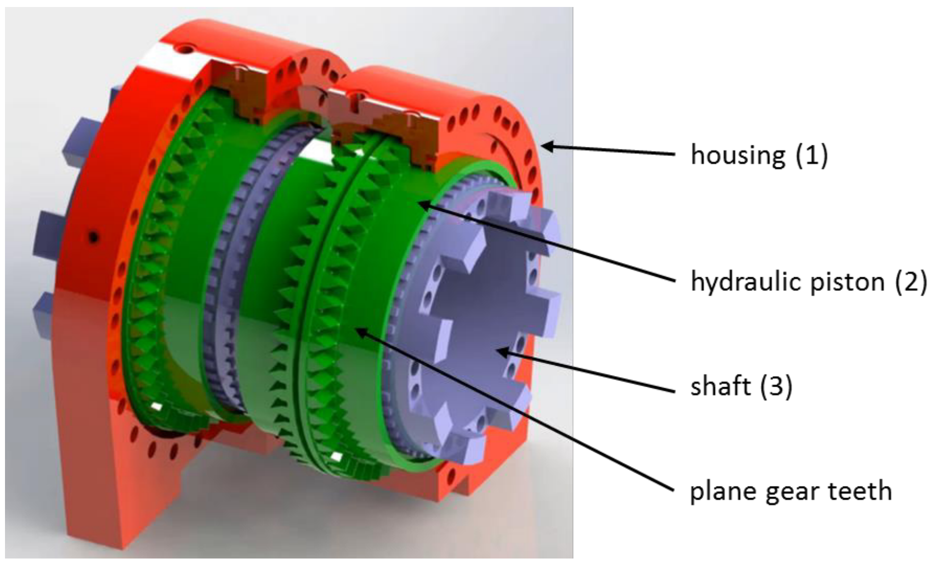


*Figure 3: direct drives, parts, source: own illustration*

### 2.2 Kinematics

The linear motion of the individual pistons is converted into a rotary movement by the plane toothing (see figure 4). Both pistons are mechanically connected to each other via the splined shaft (see figure 3, shaft (3)). This means, that the pistons can move independently linearly, but any twist which a piston undergoes through the plane toothing is also transmitted to the second piston. This mechanical coupling is especially in the reversal points necessary.

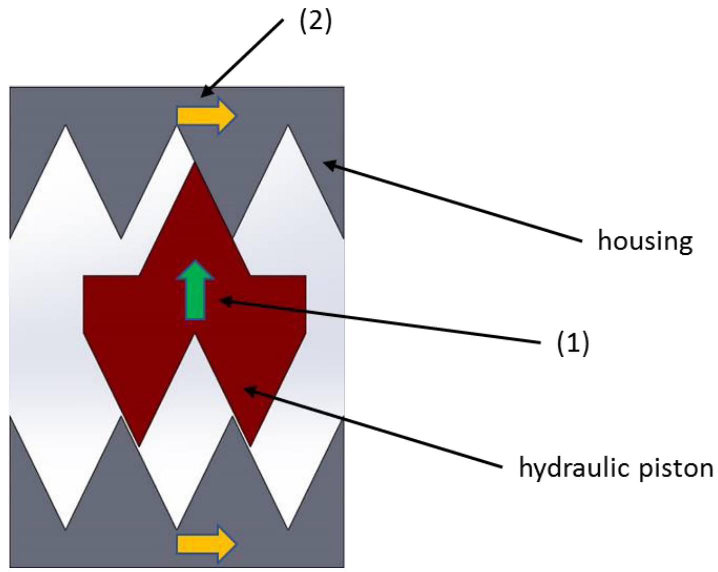


*Figure 4: unwound plane toothing, source: own illustration*

Figure 4 shows a simplified piston chamber of a cylinder with plane toothing. The simplified piston moves along the spline shaft (1). Due to the plane toothing, the longitudinal movement of the piston is converted into a rotational movement of the housing (2). When the piston is at a dead point, it has no longer any contact with the opposite plane toothing of the housing. The piston has threaded out of the toothing. In order to be able to thread into the next tooth gap during the movement reversal, the second piston is necessary. During the reversal movement of a piston is the second piston in its middle position between both housings. Therefore, the second piston can generate the necessary torque to move the boom. Thereby, the second piston ensures that the piston which is just in the movement reversal, can thread into the next tooth gap.

The resulting motion profile is shown in Figure 5. It shows the stroke of both pistons over the time. It can be seen, that the directional reversal of the pistons must take place in an infinitely short time (peak at 0 mm and 20mm strike). The following measure is taken to ensure a secure threading into the next opposing tooth gap. The pistons are not moved into the maximum possible position (dashed line in Figure 5), thereby the teeth have more space which is necessary to thread into the opposing teeth gap. As a result, it is also necessary to reduce the height of the individual teeth as well (dash-dotted line in Figure 5). This results in a trapezoidal movement profile (see Figure 6, top) of the individual pistons, whereby the speed of the individual pistons as well as the position of the individual pistons must be coordinated with each other.

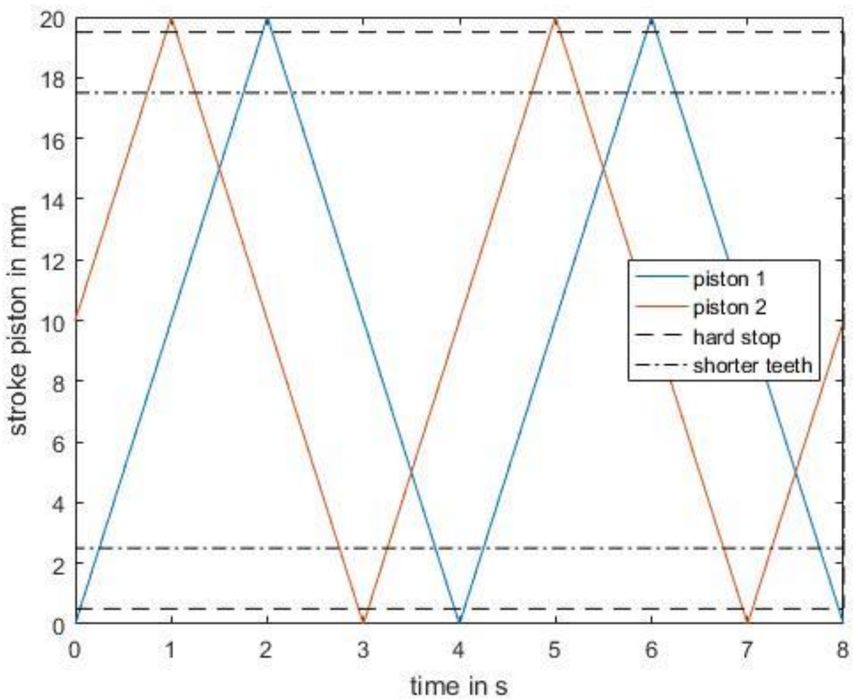


*Figure 5: Movement profile of individual pistons, source: own illustration*

An ideal motor is a hydrostatic rotary motor with ideal geometry, without backlash and without tolerances. In addition, an infinitely fast movement direction reversal is assumed. This results in the previously described ideal trapezoidal movement profile (see Figure 6, top). The volumetric flow can be calculated by the area of the piston and the velocity of the piston (see equation 1).

$$Q = v_{Kolben,1} * A_{Kolben} + v_{Kolben,2} * A_{Kolben} \tag{1}$$

It can also be seen, that the volumetric flow to the motor can change by a factor of 2, depending on whether both pistons are moving or when one of the pistons is in the reverse phase (see Figure 6 below). The black vertical lines represent the movement reversal and the dwell time, which is necessary to thread into the opposing tooth gap.

The torque of the motor can be calculated by the difference pressure of the piston, the area of the piston and the angle of the plane gear teeth (see equation 2 – 4).

$$F_{Kolben,axial} = \Delta p \cdot A_{Kolben} \tag{2}$$

$$F_{Kolben,radial} = \frac{F_{Kolben,axial}}{\tan \alpha} \tag{3}$$

$$M_T = F_{Kolben,radial} \cdot r_{wirk} \tag{4}$$

On the other way with the weight and the length of the boom the difference pressure $\Delta p$ of each cylinder can be calculated (see Figure 7). The corresponding motion profile is shown again in Figure 7 top. In this case, the black vertical lines represent the start of engagement of the individual pistons. Due to the shortening of the tooth heads, the start of engagement of the toothing deviates from the beginning of movement of the individual pistons.

Depending on the situation, whether a piston is standing still, moving without load, moving together with the second piston, or generating the required torque alone, results different pressure differences (see Figure 7 below).

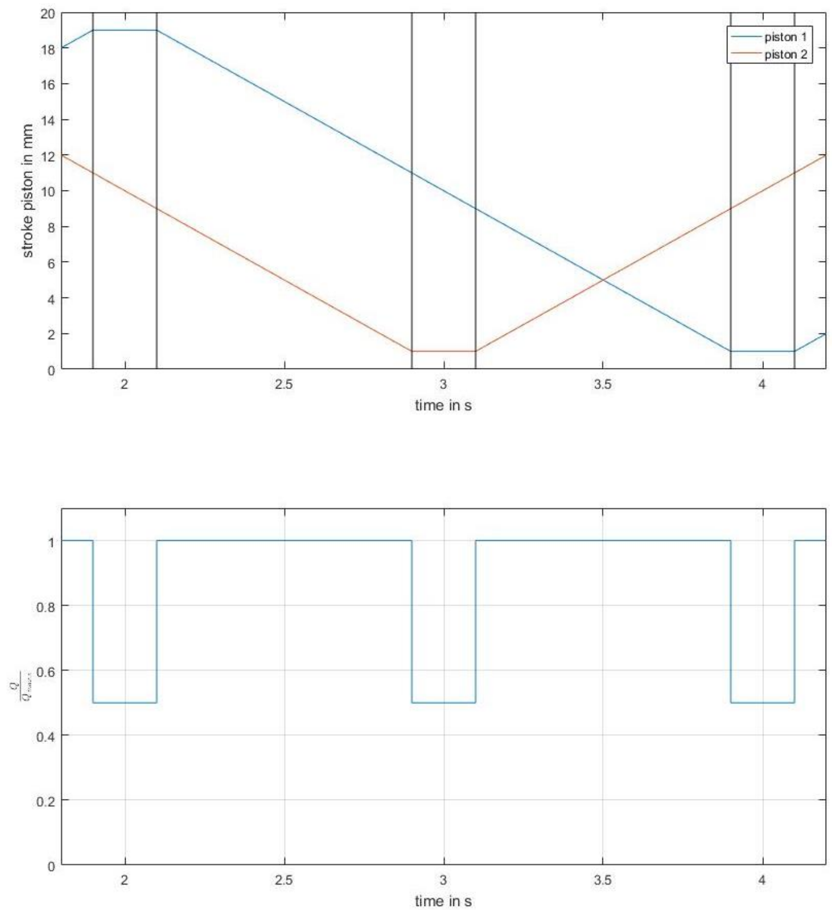


*Figure 6: Total volume flow via one stroke of the ideal motor, source: own illustration*

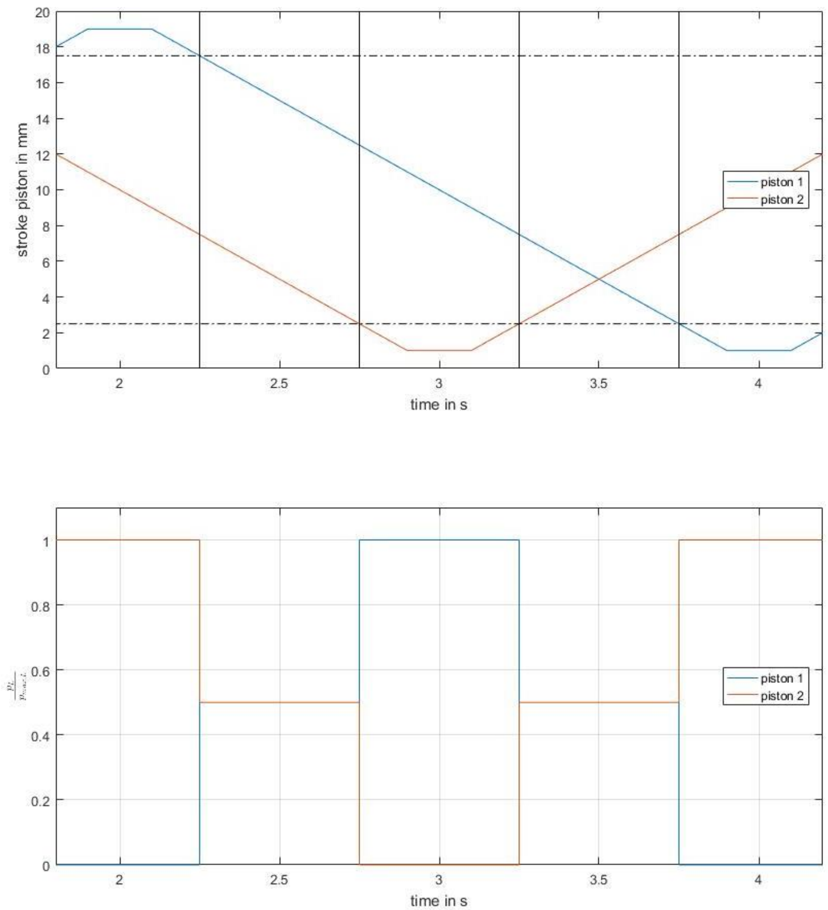


*Figure 7: Pressure difference of the ideal motor, source: own illustration*

## 3 Realized Motor

In the case of the realized motor, the pressure curves differ to the ideal motor. These deviations have ''internal'' constructive and manufacturing reasons, as well as ''external'' reasons such as long supply pipes, fluctuating loads during one piston stroke, or vibrations of the entire structure of the crane.

Furthermore, the switching points of the individual pistons must be precisely matched to one another in order to be able to achieve a smooth movement of the boom. For each moving direction of each piston results a stop point and a starting point. Each point can be set either too early or too late. In principle, it can be said, that a deviation from the ideal movement results in a blocking of the motor. Figure 8 shows the measured pressure profile of each cylinder chamber of a prototype. The corresponding movement profile of the individual pistons corresponds to Figure 7 top.

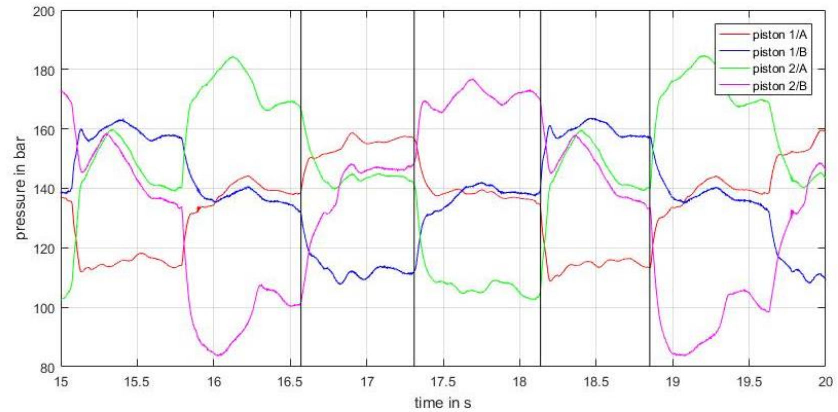


*Figure 8: Pressure of the cylinder chambers of the real motor, source: own illustration*

The vertical lines represent again the load transfer of the individual pistons. It can also be seen, that depending on the direction of movement of the pistons, different pressure differences are required to generating the torque. This can be explained by the manufacturing accuracies of the individual components and undesired relative movements of individual components relative to one another of the prototype. Figure 9 shows the required torque and the actual torque during an entire crane boom rotation. From this, the efficiency of the motor can be calculated with 65%. Further, the torque reversal during downward travel can be seen in a range of 120°. This is also a safety issue, because the boom can not fall down when the torque of the motor is positive. When the motor is operating the control unit prevents the boom to fall down, when the control unit is not working safety valves are needed.

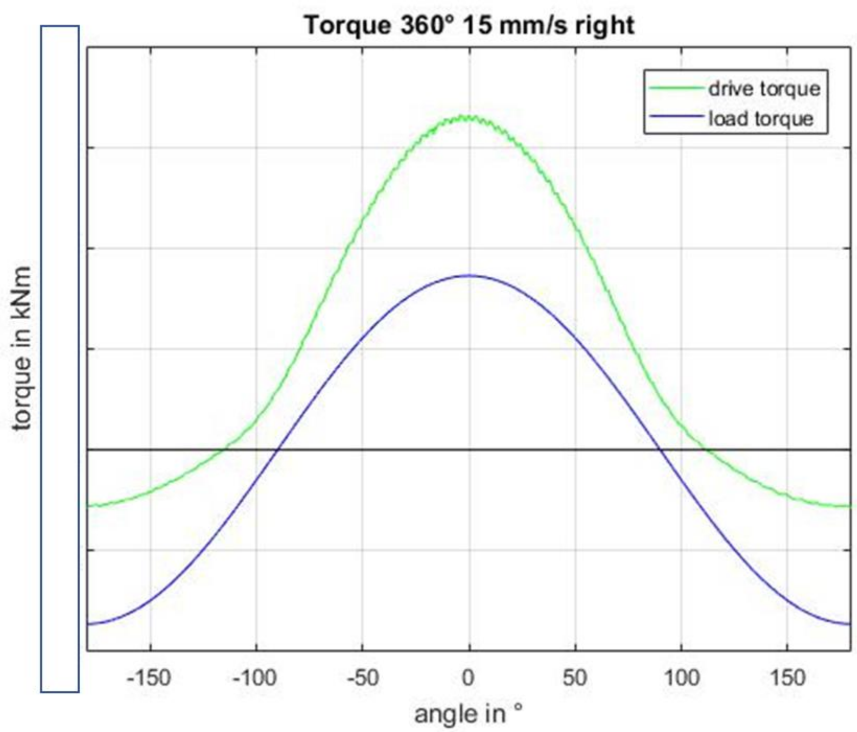


*Figure 9: Torque profile of the real motor over 360°, source: own illustration*

## 4 Summary and Conclusion

The consistent use of hydrostatic rotary motors in the joints of the booms of concrete pumps is with existing solutions of hydrostatic rotary motors not or only with limitations possible. Either there are limitations due to the swivel angle which can be achieved or the power weight represents the limiting factor.

The presented direct drive hydrostatic rotary motor can surpass existing solutions both with the swivel angle to be achieved and with the power weight. The combination of plane toothing and an integrated hydraulic cylinder makes the construction compact and light in weight. At the same time, however, high torques can be generated via the plane toothing.

The production of the components as well as the exact synchronization of the movement sequence certainly represent a challenge in the implementation. That is nevertheless possible to integrate the concept into automotive concrete pumps is shown with the presented prototype.

## Nomenclature

| *Variable* | *Description* | *Unit* |
|---|---|---|
| $\Delta p$ | Pressure difference | [N/m^2] |
| $A_{Kolben}$ | Piston area | [m^2] |
| $r_{wirk}$ | Effective radius to generate the torque | [m] |
| α | Angle of the teeth | [grad] |
| $F_{Kolben,axial}$ | Axial force of the piston | [N] |
| $F_{Kolben,radial}$ | Radial force of the piston | [N] |
| $v_{Kolben,n}$ | Velocity of the piston n | [m/s] |
| $M_T$ | Torque of the motor | [Nm] |

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