DEVELOPMENT OF A HYDROSTATIC LOAD BALANCING SYSTEM FOR 3-SPINDLE SCREW PUMPS

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In principle screw pumps are low-noise and theoretically pulsation-free positive displacement pumps. They are basically used in oil and chemical industry. Delivered liquids are e.g. jet fuel (circa 1 cSt) as well as very viscous liquids like heavy fuels (up to 100 000 cSt). For low viscosities and low rotating speeds the achievable pressure difference is limited due to the lifting force of the hydrodynamic journal bearing. The presented concept enables to increase the pressure operation limit of the screw pump.

Keywords: Screw pump, load balancing, hydrodynamic journal bearing
Target audience: Displacement pumps, simulation and validation

1 Introduction

Three spindle screw pumps are used in a wide field of application: E.g. as delivery pumps in oil and chemical industry, as cooling and lubrication pumps for ships, as fuel pumps in gas turbines or as low pulsation pumps in various hydraulic applications. The principle setup of a three spindle screw pump is given in /2/. The liquid enters the machine through the intake on the left hand side. The screw package charges the fluid from the left hand side to the right hand side. Closed delivery chambers are located between the spindles and the casing. Due to the kinematics of the spindles, the fluid is delivered continuously from the suction to the pressure side of the screw pump. This yields to a nearly pulsation free fluid flow. The screw package itself consists out of one centre drive screw and two sidewise idle screws. The spindle screws are supported by the journal bearing effect. During operation, hydrostatic forces affect the bearing reaction. They increase with the operating pressure difference of the screw pump. The pressure limit for this type of pumps is reached if the hydrostatic loads are bigger than the journal bearing forces. For that case the spindles touch the casing and damage the pump. Nowadays, a common way to increase the pressure difference between intake and outtake is to reduce the pressure rise per stage. Hence more stages are needed to reach the same pressure difference and the pump's length increases. Due to this geometrical lengthening, the hydrostatic pressure affects a larger area and the hydrostatic forces decrease. The major disadvantage of this approach is an immense increase in manufacturing effort. An example of a high pressure screw pump is given in /6/. It provides a maximum pressure difference of 175 bar and has a total length of approximately 2 meters. For this kind of pumps the spindles as well as the casing are divided in axial direction into several parts.

The increase in manufacturing effort and hence in manufacturing cost as well as the increase in weight for the screw pumps is the linking to the lateral load balancing system presented in this paper. Our system charges inlet and outlet pressure on the lateral surface of the spindle screws. In an ideal case, the presented system represents a hydrostatic bearing with a maximum bearing force that is proportional to the maximum discharge pressure of the pump. Hence, the hydrodynamic bearing is not used anymore and the pressure limit can be shifted to higher limitations.

In the following section "Reference System and its ", the basic concept of the screw pump as well as the reason for the limit of application is presented. Furthermore, an experimental approach to assess the limit of application of screw pumps as well as the results obtained with one pump at various rates of revolution for one kinematic oil viscosity is presented. In the section "Load Balancing", a concept to overcome the problem of the limit of use as well as the construction of the first prototype and the attained results are presented. The paper closes with a critical discussion of the obtained results and an outlook.

2 Reference System and its Limitations

In the following subsection, the screw pump and its functional principles are presented. Later on the limit of use due to the pressure rise across the pump is pointed out. In the second subsection, a new approach to estimate the operation limit as well as the accompanying pressure difference can be quantified. Furthermore, the results obtained with the described approach are presented for one type of screw pump.

2.1 Reference System Screw Pump

In *Figure 1* a principle sketch of the used three spindle screw pump is presented. The figure is used to define the principle parts and properties of the reference system. In the upper part of the sketch, the inlet, outlet, spindle casing and the drive screw are the main parts that can be identified. The fluid is delivered from the left hand side to the right hand side of the pump. In the lower part of the sketch, the two idle spindles as well as the bypass channel can be figured out. Concerning the working principle of the machine it has to be mentioned that the idle spindle and drive screw form closed delivery chambers due to the cycloidal tooth construction. The fluid is conveyed from suction to pressure side across the thread of the middle drive screw and the sidewise idle screws. In axial direction, the idle screws are supported by a thrust balancing. This means that the screws are promoted with the same pressure on both spot facings. The thrust balancing is realised through a bypass channel. This channel connects the suction side with a fluid chamber at the end of the pump. Inlet and outlet pressure are only separated by the balance piston of the drive screw. The bearing of the drive screw is realised by a ball bearing mounted in the shaft housing. In lateral direction, the idle screws are beard by a hydrodynamic lubrication film.

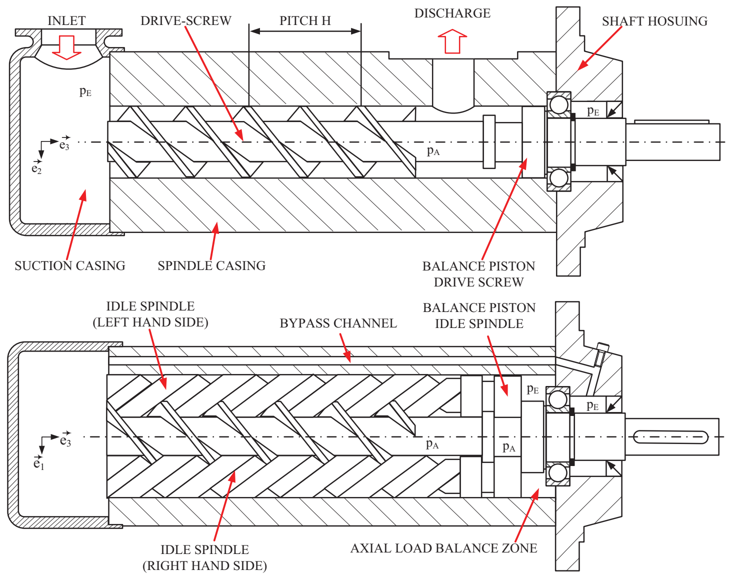

Figure 1: Setup of the three spindle screw pump.

Due to the pressure rise across the pump, hydrostatic loads affect the idle screws during the operation. Hence, the idle spindles are moved towards the spindle casing. The reason for this is presented in the schematic sketch of *Figure 2*. In this figure, a qualitative sketch of the pressure rise along the delivery chambers of the right idle spindle is presented. It can be depicted that the pressure rise at the upside and the downside of the idle screw vary due to the pitch H. Upside and downside are related to the above sketch of *Figure 1*. This means that the right idle screw is chased into the negative $\vec{e}_2$ direction due to the higher pressure at the downside. The left hand sided idle screw is chased into the positive $\vec{e}_2$ direction due to the radial symmetry of the assembly. Also due to the radial symmetry, the screw pump is free of transvers loads.

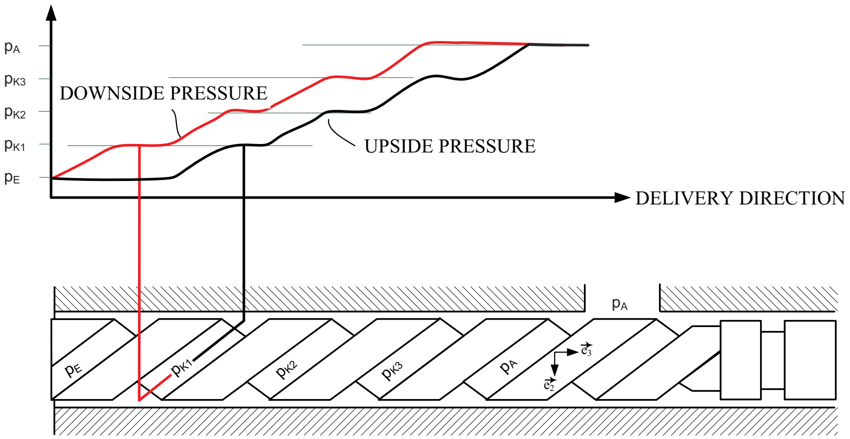

Figure 2: Sketch of the hydrostatic load across right idle spindle.

The lateral forces of a three spindle screw pump with cycloidal tooth constructions are the main topic of this paper, because they limit the maximum pressure rise that can be reached with a machine characterized by the viscosity of the fluid, the inclination of the screws H (delivery profile), the number of delivery chambers and the length of the pump. The maximum pressure of such a machine is reached if the hydrostatic loads are in balance with the maximum hydrodynamic friction bearing forces. For hydrostatic loads bigger than the maximum friction bearing forces, the idle screws grind partially at the spindle casing. In this case, further operation of the screw pump leads to a wear out.

2.2 Assessment of the Maximum Pressure Difference of Screw Pumps

In this section, a new approach to estimate the limit of application of screw pumps as well as the accompanying pressure difference is pointed out. The basic consideration is a balancing of torques. For the balancing, we take into account the three different torques acting on the pump shaft: The driving torque M provided by the electric drive, the nominal torque M_N to convey the fluid if the pump would be able to work without dissipation and the friction torque M_R that contains the losses of the pump. If these moments are balanced we end up with the following expression:

$$M = M_N + M_R \tag{1}$$

From the three torques in Equation (1), the driving torque is measured experimentally. The nominal torque can be determined by means of the principle of work and energy. The obtained result for the nominal torque is

$$M_N = \frac{\Delta p}{2\pi} V_G. \tag{2}$$

In the above equation Δp denotes the pressure difference between inlet and outlet and V_G is the geometric displacement volume. The pressure difference is obtained from measurements and the geometric displacement volume can be determined analytically as presented in /4/. Hence, we can compute the friction torque at a determined rotational speed Ω and for a given fluid of kinematic viscosity ν depending on the pressure difference

$$M_R = M - \frac{\Delta p}{2\pi} V_G. \tag{3}$$

From Equation (3) we expect a rise in friction torque if the hydrodynamic friction bearing changes its frictional state from liquid to mixed friction. This point is assumed to correspond to an operating condition where the idle spindles almost grind partially at the spindle casing. Thus, the point where the rise in friction takes place is defined as the *limit of application* for the investigated operating level.

In the following, we check the stated assumption about the increase in friction torque using the test rig presented in the following subsection.

2.3 Experimental Setup

In *Figure 3*, a hydraulic circuit diagram of the used test rig is presented. Key part of the presented test rig is the screw pump denoted by the red arrow. The pump was operated by an electrical drive with a maximum power of 18.5 kW and a maximum rotational speed of 1450 rpm. The rate of revolution is variable by means of the used frequency converter. For our investigations, we used a range of revolution speeds from 500 rpm up to 1450 rpm. The driving torque as well as the speed of revolution was measured by means of a Kistler torque meter. The measurement range of the torque meter (Type 4503A) is up to 200 Nm.

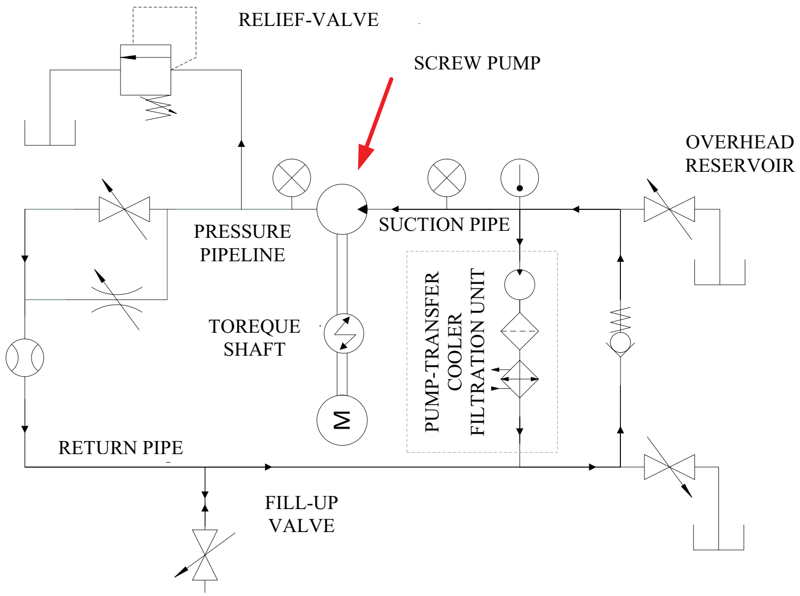

Figure 3: Hydraulic wiring scheme of the test rig.

The pump was supplied by an overhead reservoir in order to avoid cavitation in the suction pipe. The pressure rise across the screw pump was measured by means of two pressure transducers (Keller). At the suction side of the pump, a transducer with a maximum range of 5 bar was used and at the pressure side a transducer with a maximum range of 100 bar was used. At the suction side, the temperature of the fluid was measured with a PT100 thermocouple. A parallel connection of two valves was located behind the screw pump. The ball valve was commonly closed during the measurement. It served only for two tasks: On the one hand, its purpose was to reduce the flow resistance to a minimum when the geometric displacement volume was measured and on the other hand it was opened if the pressure rise among the pump reaches a critical level to release the pressure and avoid damages on the test rig. If the ball valve was closed during operation, the load for the rest rig was realised by adjusting the flow rate using a valve. Behind the valves, the flow rate was measured with a volume meter and afterwards returned into the overhead reservoir.

Due to the throttling losses, the fluid was heated up during the experiment. In a first step, these losses were used to heat the fluid up from environmental temperature to 40°C which is the standardised temperature for the kinematic viscosity indicated by the supplier of the investigated oil. Later on, whenever the temperature was higher than 44°C, the experimental investigations were stopped and the fluid was cooled down to 40°C using a pump-transfer cooler filtration unit. The filtration unit was equipped with a degree of filtration of 5 µm. As fluid a HLP hydraulic oil with a kinematic viscosity of 5 cSt was used. The mineral oil based hydraulic fluid contained additives to improve its resistance against ageing, the protection against corrosion and the wear pad.

2.4 Results

In the experiments, we measured the friction torque for the investigated screw pump at five different rates of revolution. We started at a minimum pressure difference across the screw pump and increased it step by step until a friction torque of approximately 1.5 Nm was reached. To obtained results were pointed out in *Figure 4*.

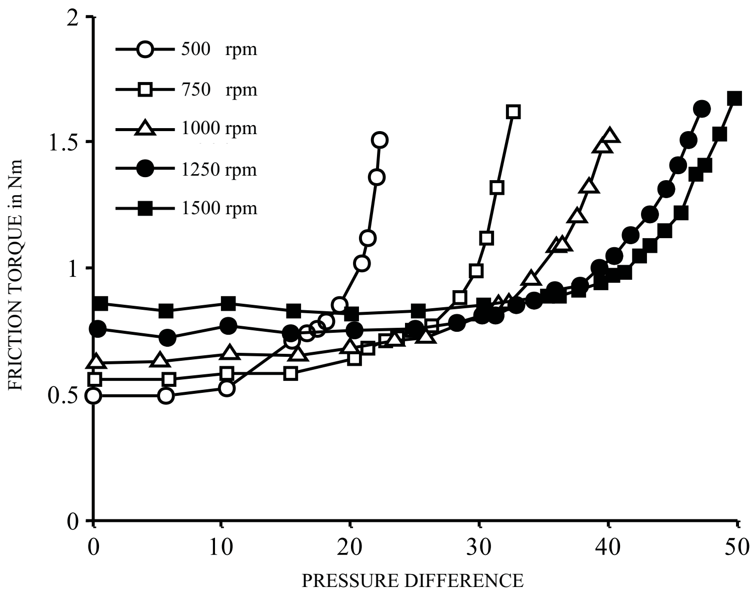

Figure 4: Measured friction torque for a three spindle screw pump.

In above figure two basic considerations have two be taken into account. At first, it can be figured out that for each rate of revolution a range exists where a disproportionately high increase of the friction torque occurs. Hence our assumption from section 2.2 that the transition from liquid to mixed friction can be figured out by measuring the friction torque is validated. Furthermore it can be seen from the first measurement that the pressure difference where the slope occurs switches with a higher speed of revolution to higher pressure differences.

The second task that has to be mentioned is the increase in the basic level of the friction torque with an increase in the rate of revolution. This seems also be reasonable due to the fact that the fluid friction increases with higher revolutions.

3 Load Balancing

In the following section, a concept to overcome the described limitation of use due to a lateral force balancing is presented. At first, the concept is presented. Following that, the design engineering of the prototype is pointed out. In the last subsection, the system is validated using the hydraulic circuit presented in section 2.3.

3.1 Concept

To improve the limitation of use of the three spindle screw pump presented in section 2.4, a lateral load balancing concept was developed. Aim of the presented approach was to obtain an entire load relieving of the idle spindles in lateral direction. In axial direction, a load balancing system is achieved due to the bypass channel.

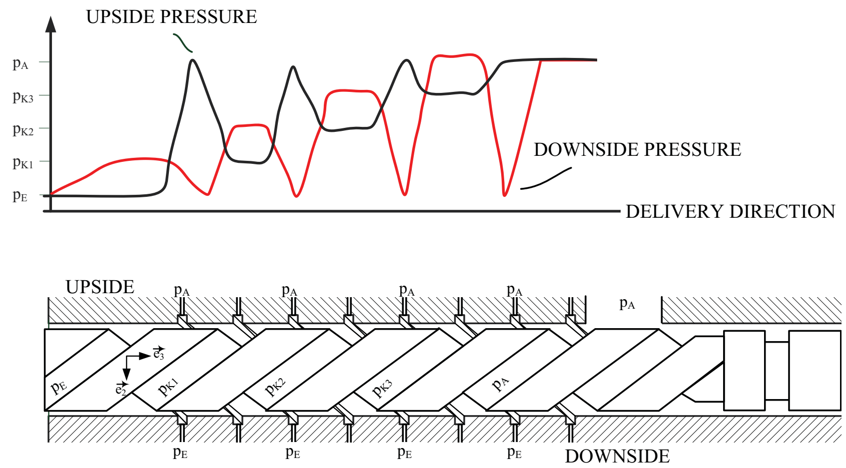

Figure 5 Concept of the lateral load balancing at the right idle spindle.

The basic idea of the system is a counter balancing of the transverse loads due to the pressure in the fluid chambers with a bearing whereby the bearing force of resistance is independent on the rate of revolution. But the deflection of the idle spindles as denoted in /1/ prevents a classical bearing approach using e.g. ball bearings or cylindrical roller bearings. Hence, a planar picking up of the bearing loads was taken into account. Therefore, a hydrostatic bearing system was designed. In this system, the tooth-side surfaces of the idle spindles acted as bearing areas. The pressure for the hydrostatic bearing is provided by the pump itself. We take *Figure 5* into account in which the right idle spindle is presented. The right idle spindle is deflected into the upside direction as it was pointed out in section 2.1. To balance the transverse loadings, pressure from the outlet p_A is provided at the upper side and forces the screw into the opposite direction. Simultaneously, the idle spindle is balanced into the downside direction through a connection with the inlet pressure p_E. From *Figure 5*, it can also be observed that after a 90° rotation of the spindle, the bore holes that are currently denoted by p_A and p_E are hydraulically decoupled from the fluid chambers while the not denoted bore holes have to provide the bearing pressure. If all bore holes would be opened at the same time, large leakage drain flows would affect the efficiency of the screw pump. This problem was solved by using a passive valve control as presented in the hydraulic wiring scheme of

Figure 6. Hence, it can be seen that due to two pressure-tap connections that are also switched due to the rotation angle

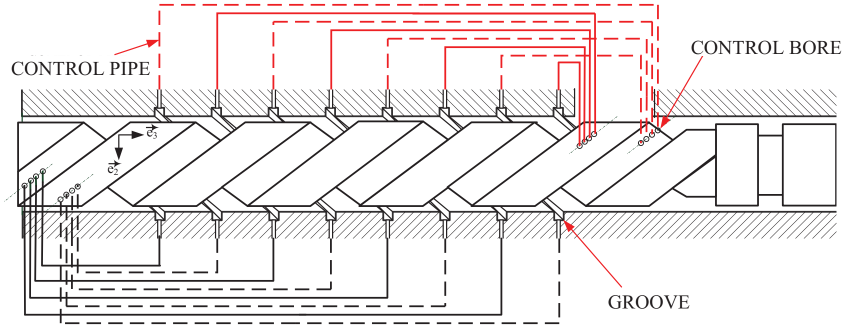

Figure 6 Hydraulic wiring scheme of the compensation system.

of the idle spindle as well the bore holes that point into the fluid chamber are closed by the tooth-side surface. In the following section, the design engineering of the prototype is presented. Due to the cycloidal tooth construction of the delivery profile, the design keeps some challenges that have to be solved. If an ideal transfer of the presented concepts could be achieved, the loading capacity of the hydrodynamic journal bearing would not be needed furthermore and the limitation of use would be omitted completely.

3.2 Prototype

The challenging task for the prototype was the manufacturing of the control grooves delivering the hydrostatic pressure to the tooth-side places of the idle spindles. In *Figure 6* the allocation of the pressure was already denoted by means of grooves that were connected via pipes. These grooves have the shape of a screw helix and had to be positioned into the interior channels of the screw pump. Due to the fact that the bearing load generated by the control bores of a diameter of 0.8 mm would be too small to balance the loads across the pump, additional grooves have to be established. The grooves have a length of 22 mm and a width of 0.5 mm.

At first it was tried to etch the grooves into the screw casing. This attempt failed due to the fact that the depth of the etched grooves was not sufficient. Hence, the more sophisticated design engineering presented in *Figure 7* was used for the prototype. The modified pump consists out of the basic pump and the compensation system. The basic pump contains the drive spindle, the idle spindles and the screw casing. For the compensation system, the discharge cartridges, the casing pipe, the suction side window, the pipe fittings as well as the not presented pipes to connect the fittings are needed. Furthermore, the screw casing has to be modified so that the cartridges can be adapted.

Initial point for the below presented design was the fact that with the available equipment of the workshop it was impossible to manufacture the grooves inside the screw casing. Hence, the cartridge design was evolved. For the cartridges, as presented in *Figure 8* it was possible to manufacture the grooves using a computerized numerical control (CNC) milling machine. The next task was the fitting of the cartridge into the screw casing. For the fit of the casing and the cartridges as presented in *Figure 9*, a very high fitting quality is recommended due to the fact that the fitting of the casing holes as well as the fitting of the spindles is in the order of some microns compared to the whole length of the pump. This fitting quality was not reached due to the manufacturing limitations related to the milling machines in our workshop. Hence, after the assembly, a contact fitting between screw casing and cartridges could be sensed by bare hand contact. Thus, with this system no satisfying results could be obtained. Therefore, the inner surface of the casing with the fitted cartridges was filled with a feed opening filling. Due to this modification, the tribology system of the pump was influenced in a meaning full manner and we could not expect a higher limit of use than obtained with the unmodified pump. But we expected to obtain a higher limit of use with our compensation system than with the remaining journal bearing effect. By means of that we can proof that our system works in principle.

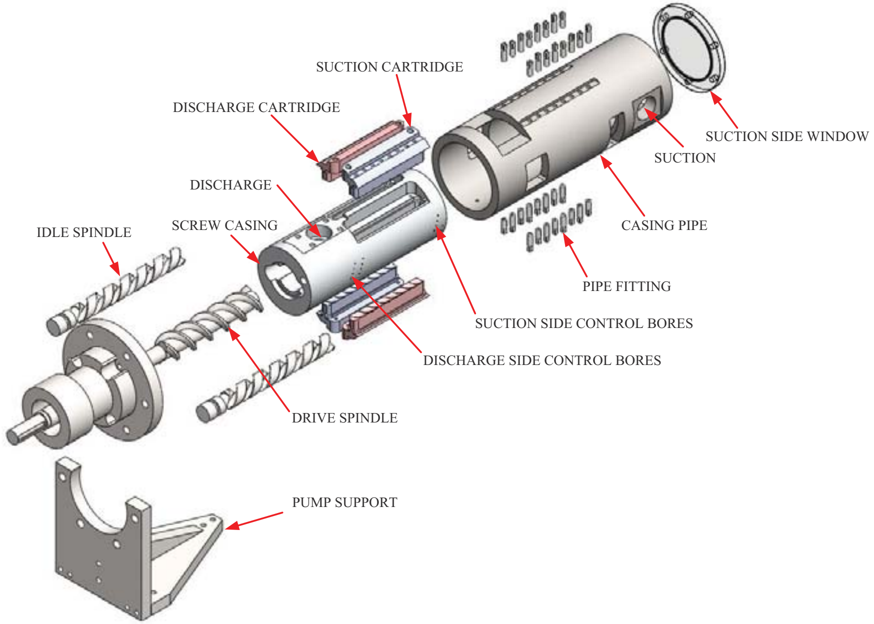

Figure 7 Exploded assembly drawing of the screw pump with the added compensation system.

From a structural analysis it was known that due to the pockets in the screw casing, the stability was not sufficient anymore. Hence, the casing pipe was used to improve stability of the whole construction. The screw casing and the casing pipe were mounted by means of an amount of shrinkage fitting. The fitting is adjusted to the operating temperature of 40°C. The casing pipe is closed by a window on the suction side to check the operational state for cavitation. The fittings are inserted into the casing and the control bores are connected via pipes as presented in *Figure 10* where an image of the prototype is presented.

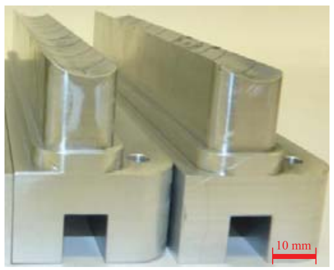

Figure 8 Cartridge for the compensation system

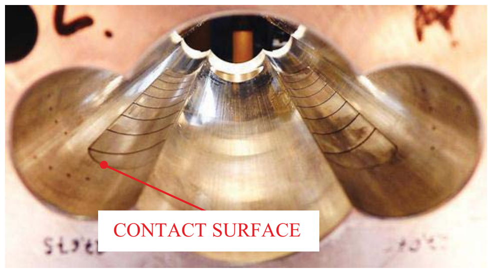

Figure 9 Cartridges fitted into the screw casing

The obtained prototype is presented in *Figure 10*, where the complex piping from the hydraulic wiring scheme can be figured out.

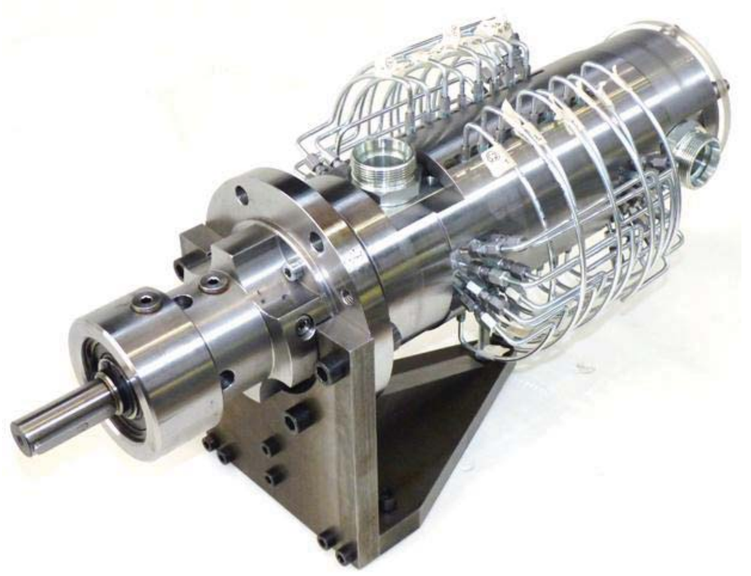

Figure 10 Image of the prototype

3.3 Results

For the presented prototype, the limitation of use was measured in the same manner as in section 2.4. In a first step, the pipes of the compensation system were disconnected and the pipe fittings were sealed with blind plugs. During these measurements, the remaining journal bearing effect was estimated. As expected in section 3.2, the fit of the inner casing of the screw pump was influenced by our modifications in a meaningful manner. The limit of use was reduced from 36 bar in *Figure 4* of the basic screw pump down to 8 bar presented in *Figure 11*.

With the compensation system it was possible to shift the limit of use from 8 bar up to 12 bar with compensation bores of 0.8 mm as presented in *Figure 11*. Hence, we increased the limit of use by 50%. In the next step the compensation bores were opened to 1.6 mm. Due to the fact that the bore holes are the smallest diameter in the system, we expected a higher application limit due to a dethrottling of the system. With this setup, we could not enlarge the limit of use in the region of lubrication friction as it was defined in section 2.2. But in the region of mixed friction at a friction torque of 2 Nm we were able to increase our maximum pressure from 12 bar up to 36 bar and obtained the former limit of use.

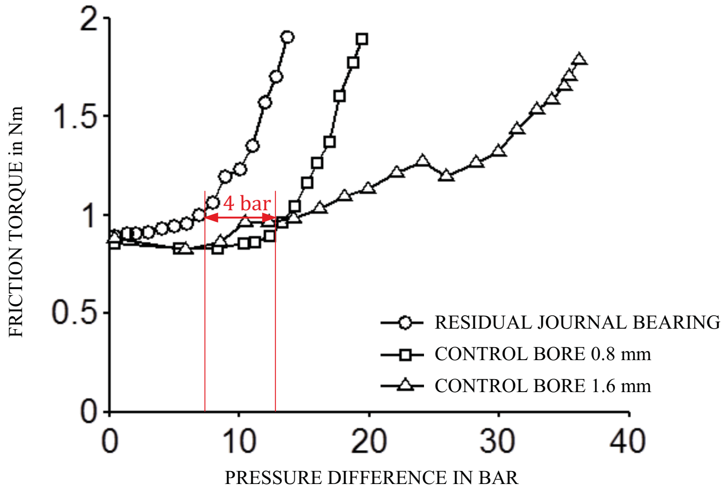

Figure 11 Friction torque of the modified screw with and without compensation system as 1500 rpm.

4 Summary and Conclusion

In this paper, for a three spindle screw pump the limitation of use as well as its enhancement was presented. At first, the basic principle of a three spindle screw pump and the reason for the limitation in pressure rise across the pump was pointed out in detail. Furthermore, an approach to estimate the limitation of use in an experimental manner depending on the operation point was presented. This method was evaluated through an experimental investigation for one type of screw pump, one fluid and at different rates of revolution. In the still ongoing project, this method will be applied and validated for different types of screw pumps and different fluids.

Additionally, an invention to enlarge the limitation of use was introduced for the first time. The main idea of the presented concept is to replace the hydrodynamic journal bearing by a hydrostatic bearing. In this concept, the pressure supply of the bearing is realised by the screw pump itself. Furthermore, the design engineering of the prototype was presented. Key task for this construction was the application of grooves into the inner casing of the screw pump to distribute the bearing pressure. This was realised by using a cartridge concept. With the prototype, it was possible to enhance the limitation of use by 50%. Unsatisfactory was the fact due to the manufacturing process of the prototype – the fitting system of the screw pump was affected negatively. Hence, the limitation of use for the modified screw pump was about 8 bar whereas the limitation of use for repetitions parts is up to 36 bar. Due to the already mentioned negative side effects, mainly resulting from manufacturing quality, with the presented compensation system from an absolute point of view it was unfortunately not possible to enhance the limitation of use significantly. But nevertheless, it can be figured out that the suggested and patented concept /5/ has a great potential with regard to the design engineering of high pressure screw pumps.

Nomenclature

Variable	*Description*	*Unit*
H	Inclination of the screw	[m]
M	Driving torque	[kgm^2/s^2]

M_N	Nominal torque	[kgm^2/s^2]
M_R	Friction torque	[kgm^2/s^2]
p_A	Outlet pressure	[kg/ms^2]
p_E	Inlet pressure	[kg/ms^2]
Δp	Pressure difference	[kg/ms^2]
V_G	Geometric displacement volume	[m^3]
ν	Kinematic viscosity	[m^2/s]
Ω	Rate of revolution	[1/s]

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