Experimental studies on the volumetric efficiency of triple screw pumps

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Abstract

Screw pumps are positive displacement pumps. The fluid is moved forward by the motion of in theory closed displacement volumes from suction to pressure side. Due to the pressure difference there is, in practice, always a leakage flow from pressure to suction side resulting in a drop of the hydraulic efficiency. The back flow from upstream to downstream displacement volumes is restricted by a metal to metal line-like contact sealing. For screw pumps there are five different types of gaps and it is the objective of the presented investigation to measure and model the associated leakage flows by direct or indirect methods. In most cases it is impossible to measure the internal leakage flows directly. Hence a suitable leakage model for each gap is necessary to evaluate the particular leakage flows. Based on measured flow rates of the screw pump, the leakage flows in the screw pumps are determined by subtracting their theoretical volume flows. These theoretical volume flows are given by the products of the geometrical displacement volume and the relevant speeds of rotation. The variations in the leakage flows through the different gaps are achieved by modification of the spindles. Due to the fact that nearly each spindle modification affects more than one gap type, an experimental concept to evaluate the leakage flow through each gap will be elaborated. At the current stage of work the different gaps are presented as well as the spindle modifications to evaluate the different leakage flows. Furthermore the variance of the leakage flows of three identical screw pumps was investigated.

Introduction

Triple screw pumps are used in a wide field of applications: 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 theory and setup of triple screw pumps is given in various textbooks that are engaged with the subject of positive displacement pumps e.g. from Ivantsyn [1]. In order to explain the nature and scope of the paper an introduction of the working principle is useful. Therefore the working principle of triple screw pumps in brief: First, the liquid enters the machine through the intake on the left hand side. Then 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 consists of one drive screw and two sidewise idle screws. The spindle screws are supported by the journal bearing effect. The sealing between the spindles is achieved by sufficiently small fit clearances. This kind of metallic sealing is derived from basic principle, where under no circumstances, can be hermetically sealed. Hence the key part of the current investigations is to quantify the statement “sufficiently small” due to the fact that the manufacturing effort as well as manufacturing cost increases with increasing demands on the fit clearances.

At this point in time the only approach to quantify the leakage flow of positive displacement pumps is to quantify the reduction of delivery flow that accompanies the rising pressure. In this paper we introduce a method to measure the leakage flow in the overflow pipe as well as an approach to quantify the different profile losses across the various gaps of the delivery profile.

To fulfill our objectives we must first introduce the geometry and nomenclature of the screw pump that is used in the paper. Therefore, we will explain the geometry of the different gaps that occur in the profile of triple screw pumps. Furthermore we introduce our approach to evaluate the leakage flow among the different gaps. Afterwards the section “Experimental Setup” presents our test rig that is used to evaluate the leakage flows. Then we will present the first measurements related to the variance of the manufacturing process of the pumps and the leakage flow across the throttle gap of the drive screw and the idle screws. The paper closes with a conclusion and an outlook about the ongoing work.

Geometry and gaps of triple screw pumps

In a first step the geometry as well as the different gaps of triple screw pumps are illustrated in Fig. 1. The liquid enters the screw pump left hand sided with inlet pressure p_{IN} and is charged by the delivery chambers successively to the outlet located on the right hand side of the sketch. With each delivery chamber the pressure rises successively from inlet to outlet pressure p_{OUT}. The pressure rise per stage depends upon the pitch of the screw. The closed delivery chambers are formed between the idle screws, the idle screws and the casing. At

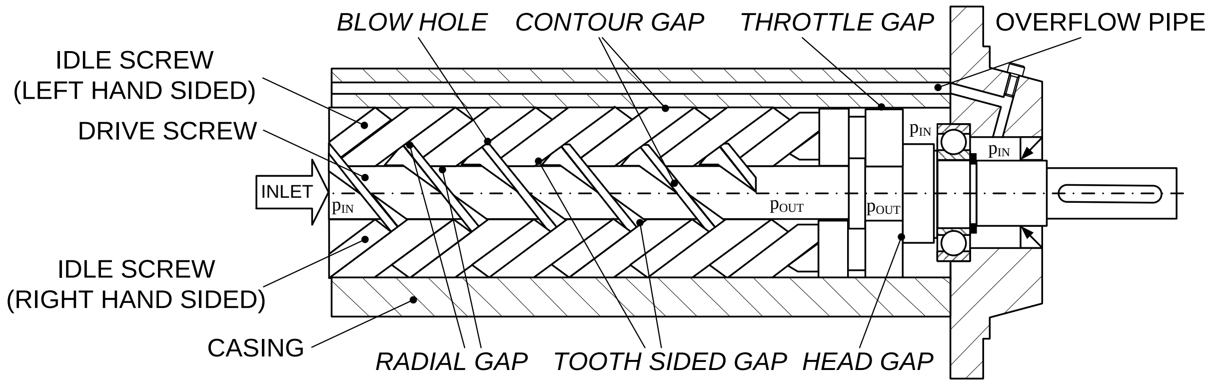

Fig. 1: Sectional view of a triple screw pump

which the fluid is charged in between the idle screws and the casing.

The idle screws are (ISC) powered by the drive screw (DSC) and are beard by the journal bearing effect. The working principle of screw pumps accounts for six different gaps. The gaps and their location are denoted in Fig. 1 by italics.

We can distinguish three different types of gaps: Line gaps, planar gaps, and the so called blow hole. The contour gap, the throttle gap, and the head gap can be enumerated among the planar gaps. Whereas the radial gap and the tooth sided gap can be enumerated among the line gaps. The blow hole can neither be classified as a planar gap nor as a Line gap. So it represents a class of its own.

For the entirety of this paper, profound understanding of the lobe profile is required so that the gaps can be described in clarity. The lobe profile of a triple screw pump is presented in Fig. 2. On the left hand side of the sketch the screw package of one delivery stage can be recognized and on the right hand side a view on the face of the screw packages is represented. From the face few illustration it

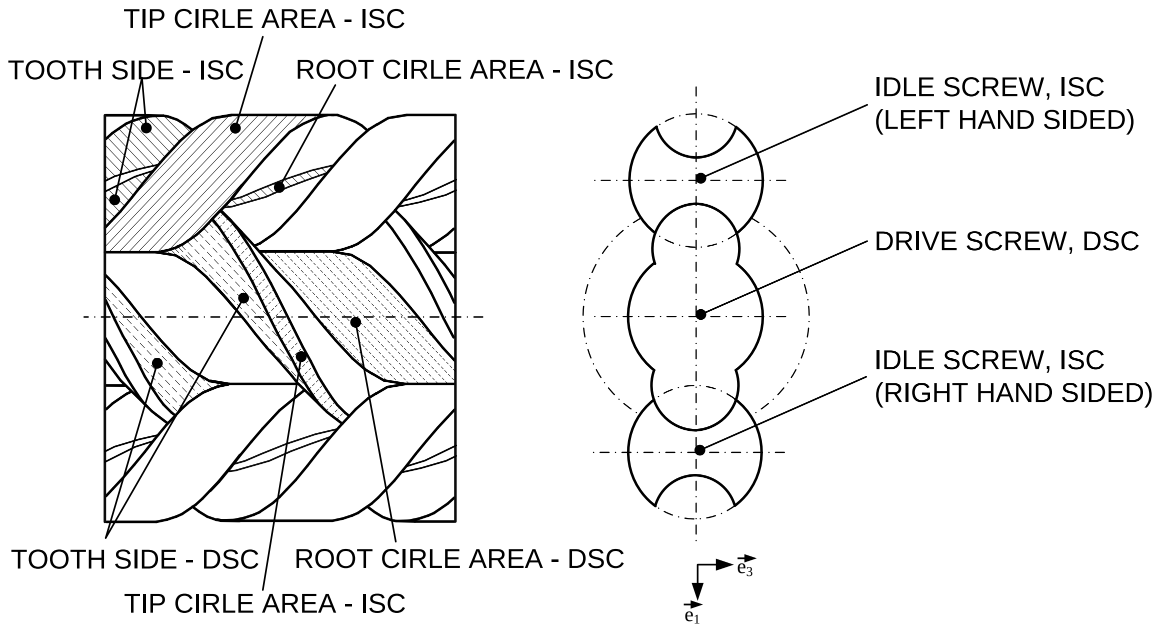

Fig. 2: Lobe profile definitions of triple screw pumps.

can be figured out that the investigated screw pump has a cycloidal toothing. On the left hand sketch, the three different parts of the cycloidal toothing are denoted: The tip circle area, the root circle area and the tooth side. The idle screws describe an epicycloid orbit and the three different areas of the toothing are denoted by the solid lines. The drive screw describes a hypocycloid orbit and the three different areas of the toothing are denoted by the dash-dot lines. A detailed theory of the cycloidal toothing are given in in the monographs by Thurner [3] and Žmud [4]. The leakage paths can be subdivided into line gaps that occur mainly in the lobe profle and into the planar gaps that occur mainly between the spindles and the casing.

Planar gaps

Planar gaps are the contour gap, the throttle gap and the head gap. The planar gaps can be subdivided into two different groups of gaps. The leakage flow among the contour gaps results in a backflow within the lobe profile. Whereas the leakage flow of the throttle gap and the head gap are floated back across the overflow pipe. The task of the overflow pipe is an axial load balancing of the idle screw. Because of the bypass the idle spindles are relief rinsed by a fluid with inlet pressure. Hence there is in an ideal case no resulting axial force that affects the idle spindles. A major disadvantage of this load balancing approach is a leakage flow with maximum pressure difference across head and throttle gaps. In the following section the geometrical context of the three planar gaps is explained in detail using appropriated illustrations.

Contour gap

Fig. 3 shows a face view of a triple screw pump with the drive screw and two idle screws as well as the casing. The contour gaps are located between the tip circle area of the screws and the casing. At triple screw pumps the Counter gaps have to be separated in the contour gaps of the drive screw and the contour gaps of the two idle screws. The drive screw has two contour gaps due to the fact that the surrounding of the casing is interrupted by the two idle spindles. Each idle screw has only one contour gap cause the surrounding of the casing is interrupted only once by the meshing of the drive screw. Hence the screw pump has two different kinds of contour gaps if the rotational length of the two contour gaps of the drive screw and the idle screws are summed up.

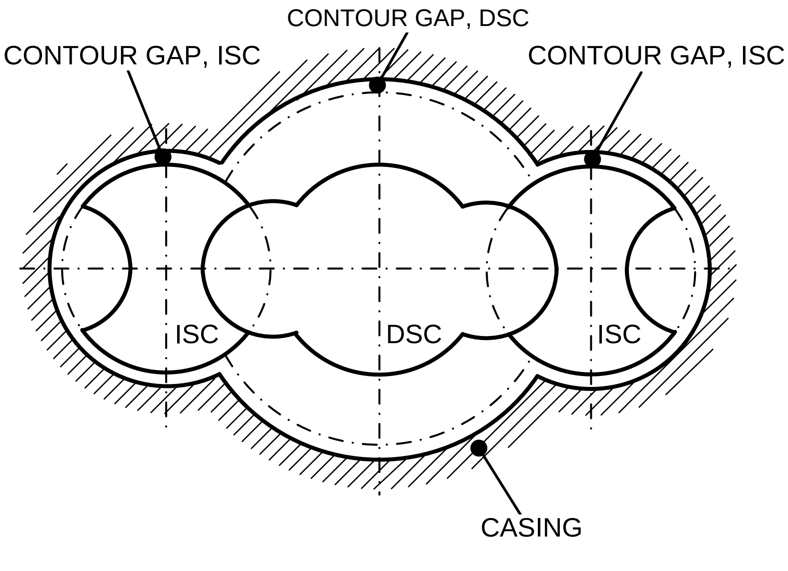

Fig. 3: Face view of a triple screw pump – contour gaps

Throttle and head gap

Fig. 4 represents a detailed view of the rear part of a triple screw pump. Apparitional are the front (the idle screws are omitted in this view) and the bottom view of the multi-view orthographic projection. The idle screws as well as drive screw end with the choke plunger. The choke plunger reduces the leakage flow from pressure to suction side and forms two different gaps. The throttle gaps have to be differentiated into the throttle gaps of the drive screw and the throttle gaps of the idle screws. In principle they behave similarly to the contour gaps except that the head loss affects one stage and the fluid flow has the opposite direction – from the left hand side to the right hand side.

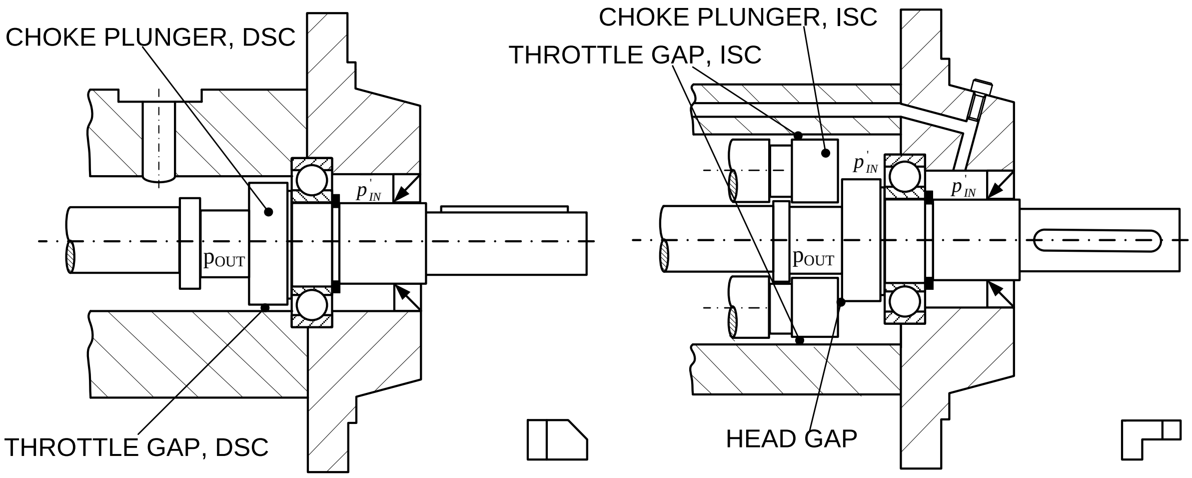

Fig. 4: Throttle and head gap in detail.

The head gap is accomplished by the spacing of the reverse side of the choke plungers of the idle screws and the front side of the choke plunger of the drive screw. Its spacing arises by the axial force equilibrium of the spindles.
Due to the toothing of the spindles and without any pressure influence the reverse side of the choke plunger is moved toward the front side of the choke plunger of the drive screw.

Line gaps

Line gaps occur in between the lobe profile where the screws unroll of another's. We differentiate two types of line gaps the radial gap and the Tooth sided gap. Fig. 5 shows the sealing lines of a pair of drive screw and idle screw. The two different kinds of gaps are represented by the solid and the dashed sealing line.

Radial gap

The radial gap emerges between the contact line of the tip circle and the root circle of the screws as presented in Fig. 5. The sealing line of the radial gap is denoted by the line. In fact, the drive screw as well as the idle screws have tip and root circle. Hence we have to consider for both types of gaps:
The radial gap of the drive screw is defined by the root circle of the drive screw and the tip circle of the idle screw. The radial gap of the idle screws is defined by the root circle of the idle screw and the tip circle of the drive screw.

Tooth sided gap

The tooth sided gap develops between the tooth side of the drive screw and the idle screw at the contact line as presented in Fig. 5. The sealing line of the tooth sided gap is denoted by the dashed line. A delivery chamber consists out of two tooth sides hence we distinguish the preparatory and the lagging tooth side and double the leakage flow. In a model concept the tooth sided gaps of each delivery stage can be summed up.

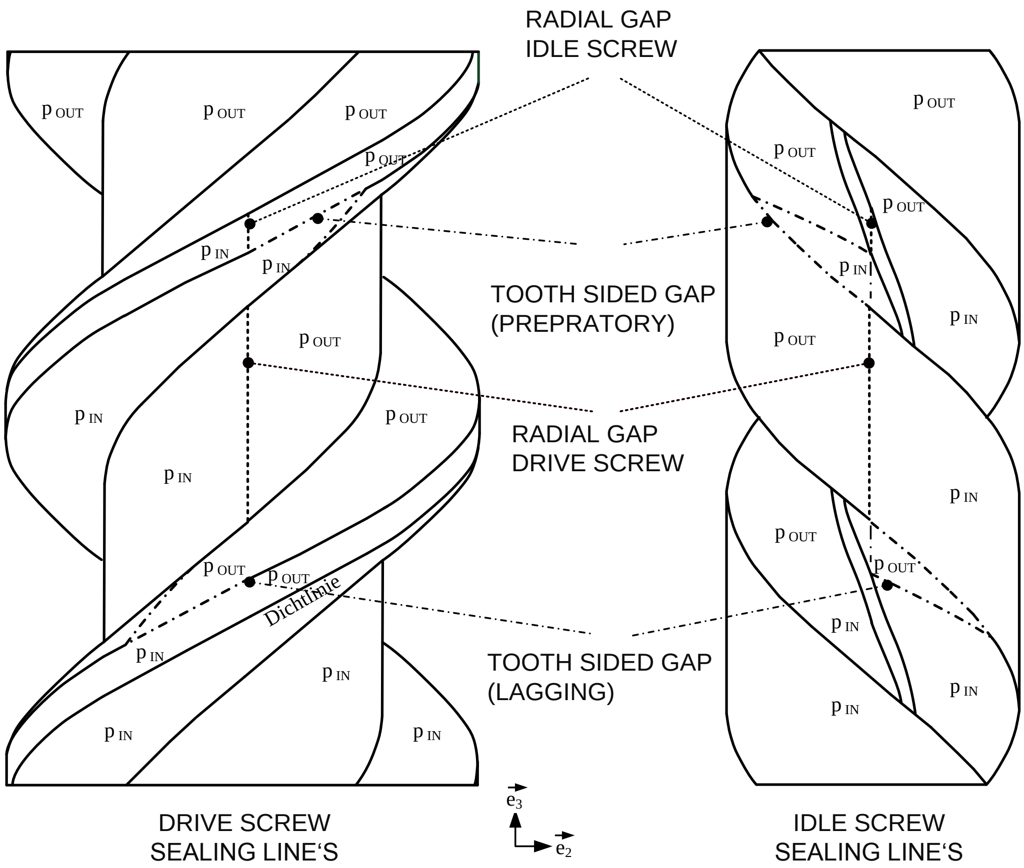

Fig. 5: Sealing lines and line gaps of drive screw and idle screw.

Blow hole

The blow hole cannot be clearly identified neither as a line gap nor as a planar gap. It occurs at the intersection line of the casing, the drive screw and the idle screw.

Procedure to evaluate the leakage flow of each gap

As addressed in the introduction nowadays the common approach to quantify the efficiency of positive displacement pumps is to measure the reduction in delivery rate with increasing pressure difference. Our new approach is based on an additional flow meter and a more staged measurement process.

Leakage flow across the overflow pipe

For our measurement process we add a second volumetric flow meter in the overflow pipe of the screw pump as presented in Fig. 6. Therefore the original overflow pipe is hermetically sealed by a plug and a bypass pipe with a volumetric flow meter is added to the screw pump. This allows us to measure the sum of the leakage flow across the head gap and the throttle gaps directly. Now we know three different kinds of flows: The ideal volume flow of the screw pump that is given by the rate of revolution multiplied with the geometrical delivery volume, the measured delivery rate, and the flow across the overflow pipe. Now it is possible to separate the profile losses. The profile losses are the ideal volume flow minus the delivery flow rate and the flow across the overflow pipe. With this information the change of leakage across each gap can be evaluated by screw modifications. The screw modifications are explained in detail in the ongoing passage.

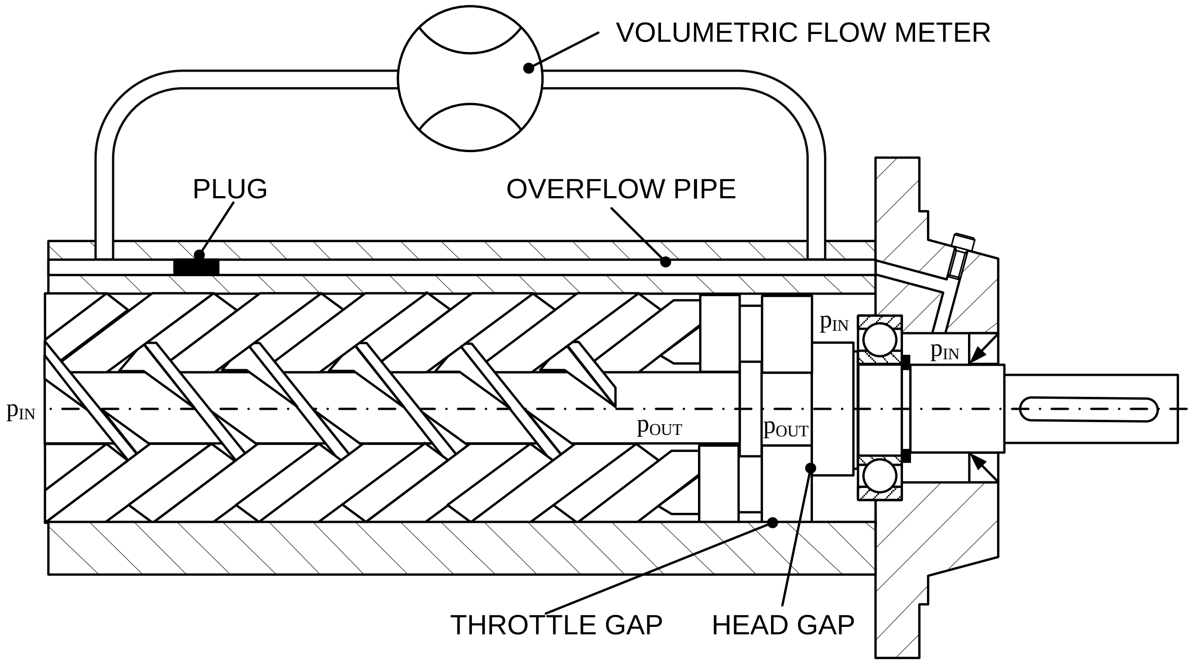

Fig. 6. Measuring the leakage flow through the overflow pipe

Screw modifications – evaluation of each gap loss

With the separation of the losses into profile losses and losses across the overflow pipe Leistritz proposed to evaluate the change in leakage flow across each gap by distinct modifications of the gaps. The modifications are listed in Table 1. In a first step the original screws have to be measured due to the fact that we can only measure the differences due to the spindle modification. The first two modifications are related to the gap losses across the overflow pipe. Modification three till seven are related to the profile losses. In the ongoing the losses across the overflow pipe and the profile are considered separately.

STATE OF MODIFICATION	MODIFICATION	AFFACTED GAP's	INVESTIGATED SET OF SCREWS	EVALUATION
ORIGINAL	-	-	DSC_{ORIG}, ISC_{ORIG}	
1st	REDUCED TIP DIAMETER OF THE CHOKE PLUNGER (DSC_{MOD1})	*THROTTLE GAP - DSC*	DSC_{MOD1}, ISC_{ORIG}	DIRECT MEASUREMENT
2nd	REDUCED TIP DIAMETER OF THE CHOKE PLUNGER (ISC_{MOD1})	*THROTTLE GAP - ISC*	DSC_{ORIG}, ISC_{MOD1}	DIRECT MEASUREMENT
3rd	REGRINDED TOOTH SIDE OF THE DRIVE SCREW (DSC_{MOD2})	*THOOTH SIDED GAP - DSC*	DSC_{MOD2}, ISC_{ORIG}	DIRECT MEASUR-EMENT
4th	REDUCED INSIDE DIAMETER OF THE DRIVE SCREW (DSC_{MOD3})	THOOTH SIDED GAP – DSC, *RADIAL GAP - DSC*	DSC_{MOD3}, ISC_{ORIG}	
5th	REDUCED TIP DIAMETER OF THE IDLE SCREW (ISC_{MOD2})	RADIAL GAP – DSC, *CONTOUR GAP - ISC*	DSC_{ORIG}, ISC_{MOD2}	
6th	REDUCED INSIDE DIAMETER OF THE IDLE SCREW (ISC_{MOD3})	THOOTH SIDED GAP – DSC, RADIAL GAP - ISC	DSC_{ORIG}, ISC_{MOD3}	
7th	REDUCED TIP DIAMETER OF THE DRIVE SCREW (DSC_{MOD4})	RADIAL GAP – ISC, CONTOUR GAP - DSC	DSC_{MOD4}, ISC_{ORIG}	

Table 1: Screw modifications to evaluate the gap losses.

Leakage flow across the overflow pipe

The leakage flow across the overflow pipe is influenced by the leakage flow across the head gap and the throttle gap. The head gap cannot be influenced due to the fact that it is adjusted by operational state and hence it is a given system parameter and not a geometrical one. The throttle gap of the drive screw and the idle screw has to be modified in two steps. In a first step the changed volume flow across the throttle gap of the modified drive screw and the original idle screws is investigated. In a second step vice versa modification is investigated. Based on the changes in volume flow an analytical model is setup to evaluate the change in leakages flows for both modifications. With this analytical model it is possible to evaluate the losses across the throttle gap in the original modification. If we assume that the modifications of the choke plunger do not affect the losses across the head gap these losses can be evaluated by a simple subtraction. After modification one and two gap losses of the head gap and the throttle gap are evaluated.

Leakage flow across the profile

The leakage flow across the profile cannot be evaluated as easy as the losses across the overflow pipe. Due to the fact that nearly all geometrical changes in the profile influence more than one gap. Only the tooth sided gap can be influenced in a standalone mode. Therefore the modification of this gap is the first step to evaluate the profile losses. After measuring the change in leakage flow due to the modification of the tooth sided gap of the drive screw, an analytical model for these losses has to be developed.

If the losses across the tooth sided gap of the drive screw are known the fourth modification can be investigated. The fourth modification includes a change of the inner diameter of the drive screw. This modification affects the tooth sided gap of the drive screw as well as the radial gap of the same screw. With the analytical model of the third modification and the geometrical knowledge of the modification the leakage flow across the radial gap can be evaluated.

In the fifth step the tip diameter of the idle screws is reduced. This modification affects the radial gap of the drive screw and the contour gap of the idle screws. And again with the analytical model from modification four and the geometrical knowledge of the modification the leakage flow across the changes in the contour gap can be evaluated. In the sixth step the inner diameter of the idle screws is reduced. This modification affects the tooth sided gap of the drive screws as well as the radial gap of the idle screw. And again with the analytical model from modification five and the geometrical knowledge of the modification the leakage flow across the changes in the radial gap can be evaluated. In the last modification the tip diameter of the drive screw is reduced. This modification affects the radial gap of the idle screw as well as the contour gap of the drive screw. The last leakage flow can be evaluated with an analytical model that describes the leakage flow across the radial gap of the idle screws and the geometrical information about the modification of the tip diameter of the drive screw.

The losses across each gap of a triple screw pump can be evaluated using seven screw modifications and six analytical models to describe the leakage flow across each gap. The analytical models are validated with hydraulic fluids of four different viscosities at various different rates of revolution and various pressure differences.

Experimental setup

The experiments were conducted with a triple screw pump of the Leistritz Pumpen GmbH. For reasons of confidentiality and due to the fact that method is more important than the results, the investigated type of screw pump, the geometrical modifications, the viscosities of the used hydraulic fluids as well as the investigated rates of revolution will not be published in this paper.

The hydraulic system of the experimental test rig is presented in Fig. 7. Key part is the screw pump and the drive line. The drive line consists out of an electric motor and a frequency converter in order to vary the rate of revolution of the screw pump. The driving torque as well as the rate of revolution was measured by using a KISTLER torque sensor with dual range option of type 4503 A. The torque sensor is connected with the screw pump by a KTR torque overload protection clutch. The screw pump is supplied by a high level tank of the investigated hydraulic oil. The pressure difference across the screw pump is adjusted by a throttle valve. After the throttle valve, a pipe filter protects the flow meter. The bypass is used to avoid the pipe filter and measure at minimum pressure differences. In the hydraulic system, only the volumetric flow meter for the delivery flow is denoted. The flow meter for the leakage flow across the overflow pipe is not mentioned in the presented hydraulic scheme. Both flow meters are positive displacement flow meters based on the meshing gear principle and are matched with the investigated volume flows.

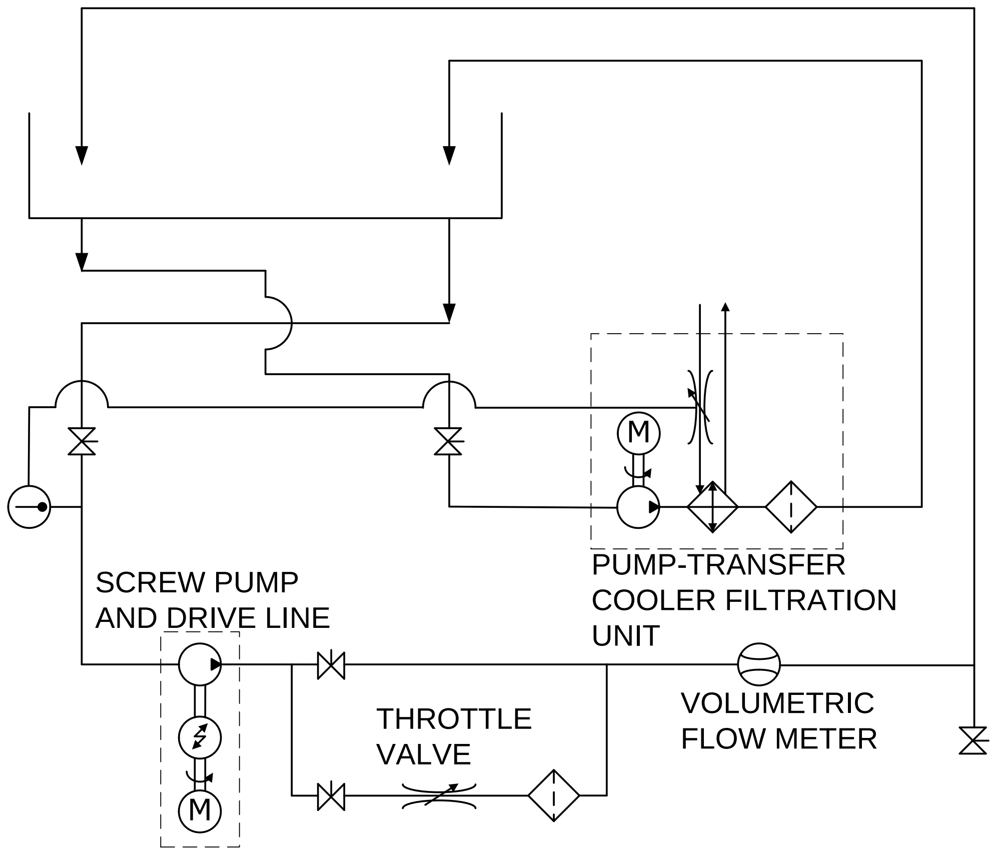

Fig. 7 Hydraulic system of the test rig.

The temperature of the system was kept constant at 40°C (± 0.2°C) using a HYDAC pump-transfer cooler filtration unit coupled with an AVTA valve that controls the water-cooling flow combined with a heating system.

Results

Before the investigations with the modified screws were conducted, several preliminary investigations have to be conducted to understand the system and its influence parameters. In the first step, the variance of the manufacturing process of the screw pumps for three different pumps was investigated. Therefore the delivery flow as well as the flow across the overflow pipe was measured for several different pressure differences and various rates of revolution. In Fig. 8 are the profile losses as well as the losses across the overflow pipe for the three different pumps for one kinematic viscosity and one rate of revolution illustrated. The plot shows the volumetric inefficiency depending on the normalized pressure difference. The leakage flow on the vertical axis is normalized with the theoretical delivery flow and the pressure difference on the horizontal axis is normalized with the kinematic viscosity of the investigated fluid and the rate of revolution. The dimensionless product that emerges by this definition is the so called “Leakage Flow Number” and was first introduced by Pelz in [2]. From Fig. 8 one can see that pump one and two have the same profile losses whereas the losses across the overflow pipe varry by approximately one percent. The profile losses from pump three varry compared to pump one and two by approimatley 1.5 %. Whereas the losses across the overflow pipe are inbetween the results of pump one and two. Overall it can be summarized that the manufacturing influence on the leakage flow is negligible small.

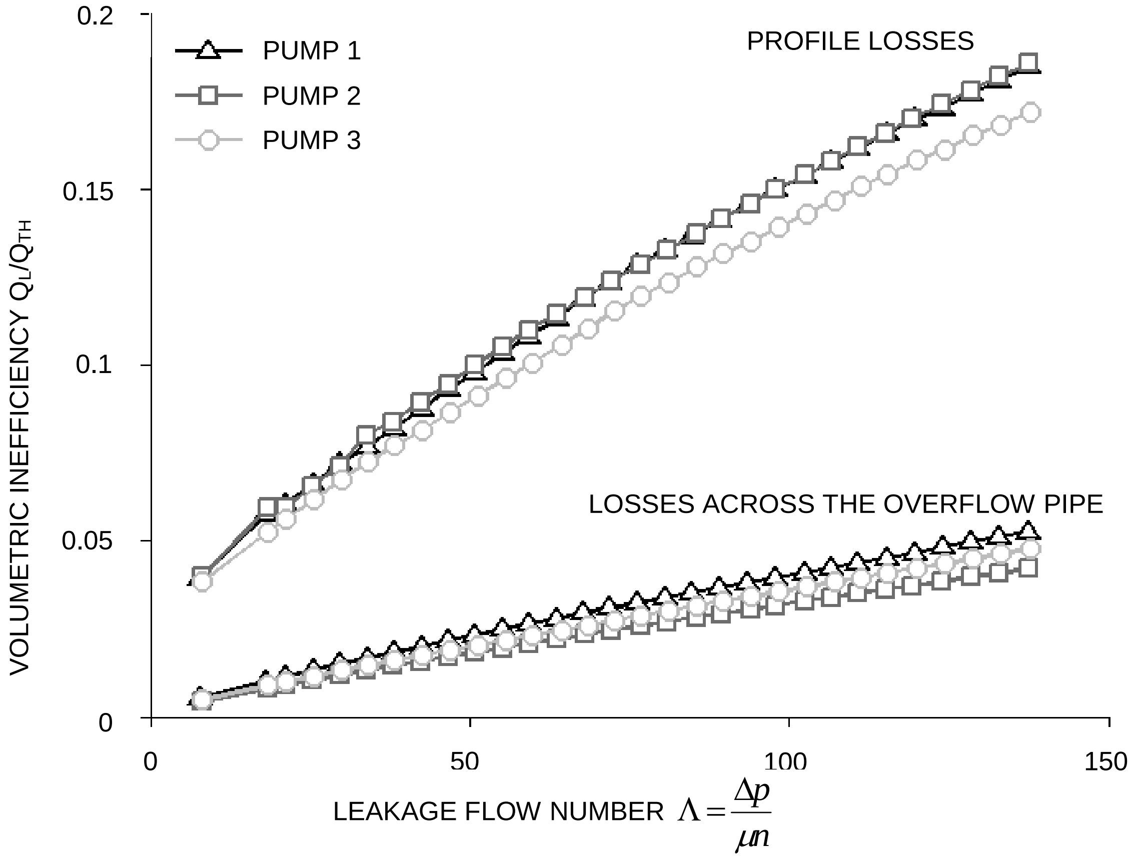

Fig. 8: Variance in the manufacturing process of three identical screw pumps.

The preliminary investigations supported us by the modification of the spindles in such a manner that the geometry change of the gaps has to be minimum in the same order of magnitude as the origin gaps to obtain a significant change in the leakage flow. To ensure that the delivery flow of the brand new pumps does not change during the measurements due to a running in characteristics the pumps were run several hours at the limitation of use until the driving torque as well as the delivery flow did not change anymore in time. All modified screws will be subjected with this running in procedure to ensure that the obtained results are reliable.

In Fig. 9 are the losses across the throttle gap for the original set of screws as well as for the modified throttle gaps of the drive screw (modification 1) and the idle screws (modification 2) represented. Each modification of the choke plungers results in a doubling of the leakage flow. With the geometry of the screw changes an analytical model for each gap modification has to be derived to evaluate the losses among the head gap. Then the leakage flow of each gap across the overflow pipe is known.

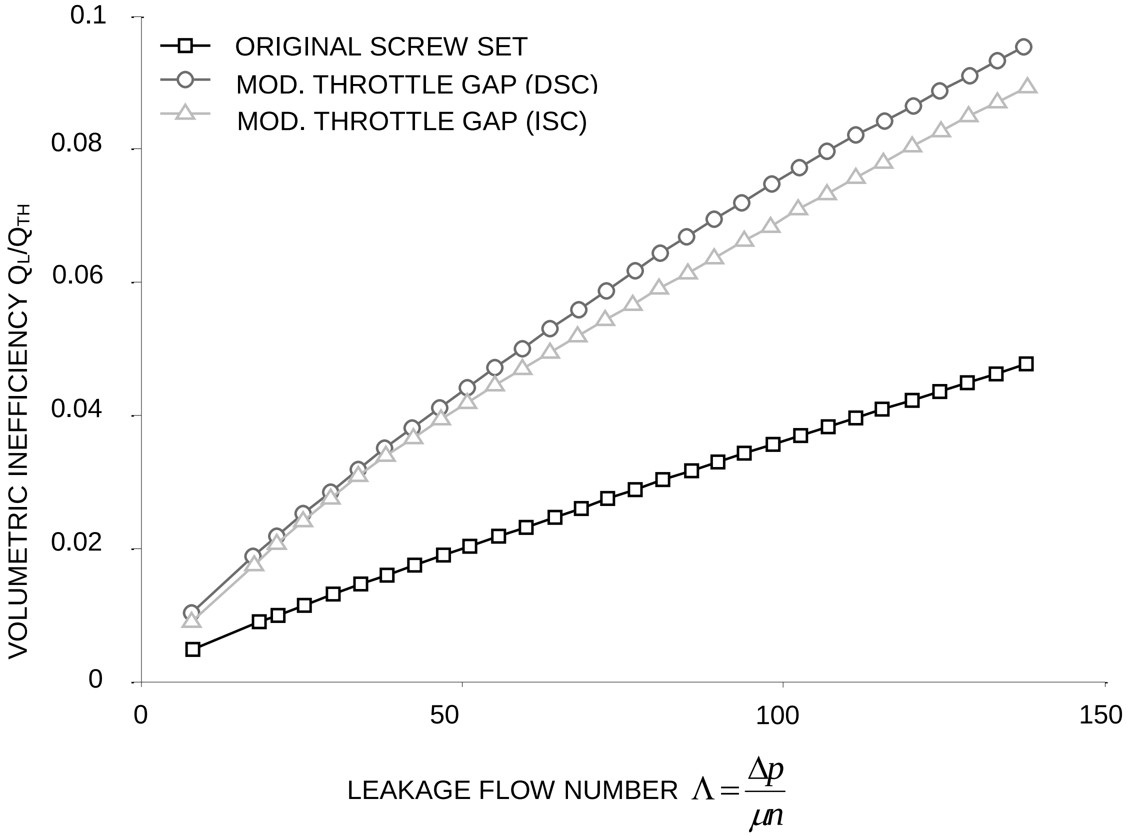

Fig. 9: Losses across the throttle gap

Summary and conclusion

In the presented paper a new method to evaluate the leakage losses across each gap of triple screw pumps was presented. In order to get the approach in detail a profound understanding of the leakage flows is essential. Therefore the internal loss mechanisms of triple screw pumps were provided in detail within the paper. Furthermore the test rig for a validation of the method and first measurements were presented. In a first step the manufacturing variance of triple screw pumps was evaluated to get a feel about the order of magnitude that the gaps have to be opened to measure a significant higher leakage flow after the modification of the screws. The modification of the choke plungers increases the leakage flow across the overflow pipe by a factor of two. It can be assumed that the order of magnitude for the gap opening was properly selected. In the ongoing work all gaps of triple screw pumps will be modified as presented in Table 1 to evaluate the leakage flow across each gap. At the end of these investigations analytical models for each gap losses can be established and validated.

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