René Brückner and Burkhard Verhülsdonk, Essen, and Meno Türk and Thomas Zenke, Potsdam

Calculating Characteristic Curves of Helical Toothed Rotary Lobe Pumps Considering Wear

Double shafted rotary lobe pumps are often used to pump suspensions with rough particles, as in liquid manure, biogas and sewage treatment plants. Pump characteristics are usually measured by water-based test equipment, although the efficiency curves change considerably under abrasive wear conditions. An algorithm for calculating internal leakage flow (slip) has been developed to determine the actual characteristic curves, depending on viscosity and wear. A new pump sizing program aids in practical application.

Dipl.-Ing. (FH) René Brückner and Dipl.-Ing. Burkhard Verhülsdonk are employees of Hugo Vogelsang Maschinenbau GmbH, Holthöge, 49632 Essen (Oldbg.); e-mail: *verhuelsdonk@vogelsang-gmbh.com*
Dr.-Ing. habil. Meno Türk and Dipl.-Ing. (FH) Thomas Zenke are employees in the division „Technik in der Tierhaltung" of the Institute für Agrartechnik Bornim e. V. (ATB) (Director: Prof. Dr.-Ing. J. Zaske), Max-Eyth-Allee 100, 14469 Potsdam; e-mail: *mtuerk@atb-potsdam.de*
This report was supported by the "Bundesministerium für Wirtschaft und Arbeit BMWi".

Summarized contribution to LANDTECHNIK. You will find the long version under LANDTECHNIK-NET.com

Keywords

Rotary lobe pumps, wear, slip, pumping behaviour, leakage calculation, software

Literature

Literature references can be called up under LT 03502 via internet http://www.landwirtschaftsverlag.com/landtech/local/literatur.htm.

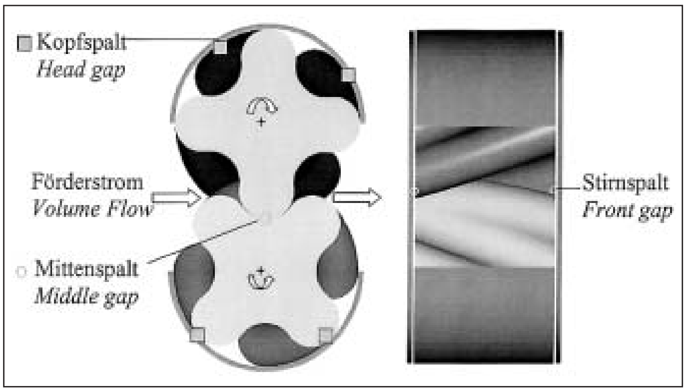

Fig. 1: Gaps in a rotary lobe pump with helical toothed lobes

There are always certain clearances between rotating and housing parts of a rotary lobe pump. An internal leakage flow (slip) is created reducing the pump's flow rate. During operation these clearances are increasing because of friction wear and jet wear caused by hard particles in the fluid. As a result the pump characteristic changes. This effect mainly depends on the flow characteristics of the fluid [1]. Therefore it should be helpful for pump users and designing engineers to be able to calculate characteristic curves with help of new software covering the all relevant factors.

Calculation of gap leakage

Using helical toothed lobes pressure pulsations are reduced to a minimum. Pulsation at spur-toothed lobes is caused by permanently moving the sealing line between the two lobes forward and backward during rotation [2]. Continuous flow with no pressure fluctuation achieves a vibration-free working behaviour of the pump.

The calculation formula assumes the total internal leakage to be caused by three different sorts of gaps (*Fig. 1*),

- the head gap between the lobe tips and the housing,
- the front gap between the lobe ends and the wear plates and
- the lobe gaps between each lobes.

The total volume flow Q is calculated from the theoretic flow rate Q_{th} reduced by the leakage rate Q_S.

$$Q = Q_{th} - Q_S \quad (1)$$

$$Q_S = \sum (A_S \bullet v_S) \quad (2)$$

$$A_S = b_S \bullet h_S \quad (3)$$

The gap areas A_S have by simplifying a rectangular shape. The leakage rate of a certain gap cross section is forced only by the differential pressure ($\Delta p = p_d - p_s$ between suction side and discharge side [3]. Accordingly Δp can be calculated:

((Gleichung einsetzen)) (4)

The pressure loss coefficient ζ is a function of the Reynolds' number (5) and - assuming constant gap dimensions - a function of flow velocity v_S and fluid viscosity η.

((Gleichung einsetzen)) (5)

The hydraulic diameter of gaps is

((Gleichung einsetzen)) (6)

If $h_S << b_S$ it is possible to calculate $d_{hyd\,S} = 2 \bullet h_S$. Slip rate Q_S and the corresponding velocities are not known yet. So the pressure loss factor ζ and v_s have to be calculated iteratively.

((Gleichung einsetzen)) (7)

((Gleichung einsetzen)) (8)

The pressure loss in a gap is assumed to be created by friction loss at the pump housing (described by the friction coefficient λ_S) and other losses, particularly diverting losses, summarised by the coefficient ς_S.

An important question is the geometrical specification of gap dimensions.

With increasing wear the *head gaps* are increasing in length l_K and height h_K. The length l_K can be calculated as a segment of circle determined by the gap height. The width of the gap b_K is determined by the number of wing per lobe. With increasing numbers of sealing lines (respectively wings) the pump's "tightness" is enhancing.

Even at multi-wing lobes there is only one sealing line in the *lobe gap* between the two lobes. Therefore its width is left constant. Even in a pump with worn-out tips the lobe gap l_M is not increasing and remains in a size range of about 0.1 to 0.3 mm (manufacturing tolerances).

Because of their length the *front gaps* are area spaced gaps. To define its width the shaft diameter has to be taken into account (particularly at the gear box side of the lobes). The length of a front gap is calculated from a wing width and the number of sealing lines.

In order to calculate the Reynolds' number (5) it is necessary to know the fluid's viscosity. Newtonian Fluids have a constant viscosity which only depends on the temperature. As at lot of organic suspensions have non-Newtonian fluid flow, behaviours depend on the current shear rate. The flow behaviour of a Non-Newtonian Fluid can be described by means of the formula from Herschel and Bulkley [4]:

$$t = t_0 + k \cdot \dot{g}^n \qquad (9)$$

When calculating the leakage flow in rotary lobe pumps, classic formulas and measuring methods of have to be modified. Separation of the fluid components in a suspension with rough structure is possible. In this case water can segregate from the solids and is going to become the main part of the slip. On the other side the gaps can be locally and temporary clogged. In addition to this high velocities and shear rates inside the gaps create low fictitious viscosities of non-Newtonian suspensions. Theses phenomena all come together and cannot be calculated in detail. Those phenomena only can be verified by measurements.

Knowing the pump dimensions, its working condition and the flow characteristic of the pumped medium, the flow rate of rotary lobe pumps now can be calculated for regularly fluids [5].

((Gleichung einsetzen)) (10)

The cross sectional area of a gap A_s and lobe width b_p are known values as well as the speed n_P. The total delivery area per lobe $A_{Fö}$ given by the pump manufacturer has to be reduced by the delivery area loss according to lobe wear-off.

The fluid velocity v_S in the gap has to be iteratively calculated by formula (7). Here λ_S and ς_S are unknown. By means of the first calculated fluid velocity v_S a first Reynolds' number can be calculated. The result gives the first value λ_S. Now ς_S can be calculated using the first value of λ_S. Now the next fluid velocity can be calculated etc. This loop can be finished if the difference between v_s and v_{s-1} does not exceed a set value (e.g. Δ_{vS} < 0.1m/s). Usually 4 loops are sufficient.

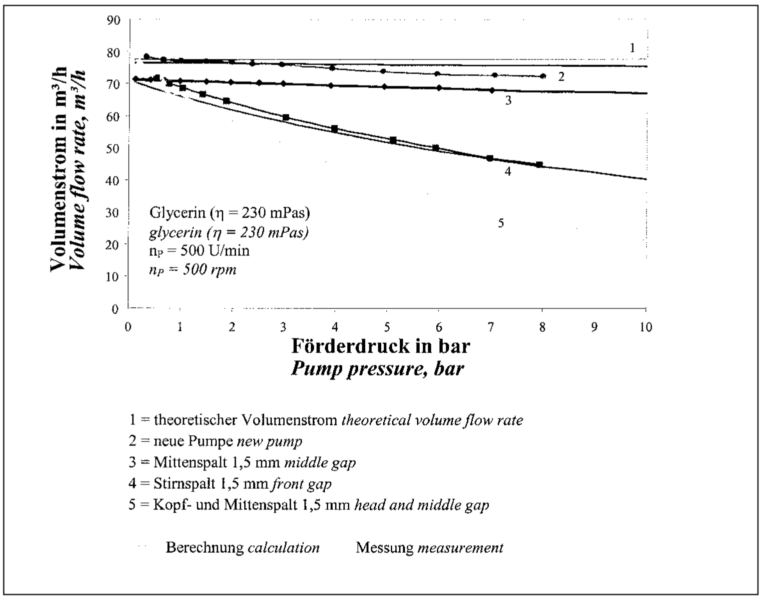

Fig. 3: Comparing experimental and calculated pump characteristic curves with glycerin

The friction loss factor λ_S for a laminar gap flow (Re < Re_{krit} = 2300) in small gaps [5] for both Newtonian or non-Newtonian fluid is calculated by

$$\lambda_S = 96 / Re \qquad (11)$$

To calculate turbulent flow conditions, diverting, shock and acceleration losses inside the pump adequate equations are known.

Knowing the single fluid velocities of the different gaps it is now possible to calculate the slip rate Q_S and the effective volume flow rate Q of the pump under the predetermined working condition (pressure difference, fluid behaviour, wear-off condition of pump).

Discussion

The calculated curves and the measured curves nearly meet each other (*Fig. 2*). Even a new pump with very small gaps can not achieve the theoretical flow rate. New pumps have to be built with clearances to avoid too much friction torque. Figure 2 shows characteristic curves for glycerine (viscosity η = 0.15 - 0.25 Pa s) and pumps with different wear-off condition. So pairs of capacity curves are showing the influence of wear at *lobe gap, front side gaps* and at the *tip gaps* together with the lobe gap. It has to be recognised that the influence of the lobe gap (middle gap) is small. The reason is that the sealing line between the lobes is permanently valid, even in the state of wear.

The effect of increasing tip gaps on the capacity is the most important one. So, technical measures to reduce capacity loss under abrasive fluid condition will have maximum success if those measures improve the abrasion behaviour of the lobe tips.

The pump selection program

In order to make this theoretical calculation concept applicable for designing engineers and users a pump selection program was developed. As a result of this computer program the user gets a range of appropriate pumps according to specified application conditions like pressure, flow rate, viscosity.

Within a list of pumps the most appropriate one is selected. The list contains the characteristic data for both a new and worn-out pump. The characteristic curves of each pump, even in worn-out state, can be calculated and is shown. With all its features the program increases the quality of customer advisory service and this is a basis for adapting pumps optimally to real application conditions. Pumps designed with this computer program have a reduced sensitivity to wear and their operation time between services is extended.

Abstract

Conventionally rotary lobe pumps will be designed for water condition although, in most cases, they are pumping different media. Calculation proposals for practically oriented characteristic curves are described. All calculations are based on tests mostly carried out with helical toothed rotary lobe pumps. The tested pumps were fitted with systematically changed internal clearances and are tested with fluids having different viscosities. A pump sizing program was developed the calculation results of which are verified by tests. Pump behaviour can be calculated in advance even for pumps working under abrasive conditions. The sizing program is valid for pumps with straight and helical toothed lobes.

The flow properties of the fluid can differ very much. Therefore additional laboratory tests are necessary particularly at non-Newtonian fluids. If those flow characteristics are known as well as the pump dimensions the working behaviour of a rotary lobe pump can be determined.

When using additionally an already existing program for pipe flow calculations [4], pump equipment fitted with a rotary piston pump can be designed perfectly.