Optimized Design of Screw-Type Vacuum Pumps

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Abstract

This paper concentrates on the efficiency-optimised design of machine parameters for dry-running screw-type vacuum pumps. To model screw-spindle vacuum pumps, an automatic chamber model generator is presented. With these models the variation of the geometrical parameters (rotor crown and root diameter, number of lobes, rotor gradient) and operating point variables are simulated. The simulation results are discussed to point out possible design criteria for such pumps. Therefore the influences of the different parameters on the delivery rate, the volume flow rate and the specific power are analyzed.

Symbols und Indices

Symbol	Meaning	Unit
a	centre distance	m
d	diameter	m
l	length of rotor	m
s	pitch of rotor	m
n	rotor speed	s^{-1}

Symbol	Meaning	Unit
S	suction capacity	m^3/s
u	circumferential velocity	m/s
V	volume	m^3
Φ	degree of a lobe on the rotor	-
λ_L	volumetric efficiency	-

1. Introduction

The aim of the research project is the development of a method to optimize the efficiency of dry running vacuum pumps. Characteristic values to describe the geometrical dependency of the thermodynamic behaviour as a quantitative criterion are developed. The geometrical machine parameters and their physical-technical effects are analysed using a screw-spindle vacuum pump.

The mathematical modelling of the screw-spindle geometry (based on the real machine) and the following thermodynamic simulation make possible the simple and cost-efficient development of efficiency-optimised machine geometries. As benchmark criteria the delivery rate, the volume flow rate and the specific power consumption are chosen. The geometries developed geometries should be optimal for all criteria at the same time.

2. Modus operandi

One possible option for the search for the best combination of geometric parameters is an analysis of the mathematical functions describing the machine. But this approach in isolation is very complex in thermodynamic terms, and the potential for success is small.

Another way to analyse the influence of geometrical parameters is using the thermodynamic simulation system "*KaSim*". The simulation results are analysed according to the criteria specified above, i.e. volumetric efficiency, suction capacity and specific power input. The results will also be evaluated with reference to geometrical limitations and development potential for the design.

3. Automatic chamber model generator

Converting the rotor geometry into a chamber model is difficult. The size and gradient of the chamber volumes, the gaps and the charge areas are variable and have to be described so that they can be calculated with the simulation system *KaSim*. This is the main reason for creating an automatic model generator. This new program reads all necessary input parameters from a text file and checks the plausibility of the parameters. Subsequently all necessary input files for the thermodynamic simulation system are created.

3.1. Variation parameter

There are numerous possible parameters for the screw spindle geometry, depending on the different rotor types. The parameters are classified by different basic rotor types.

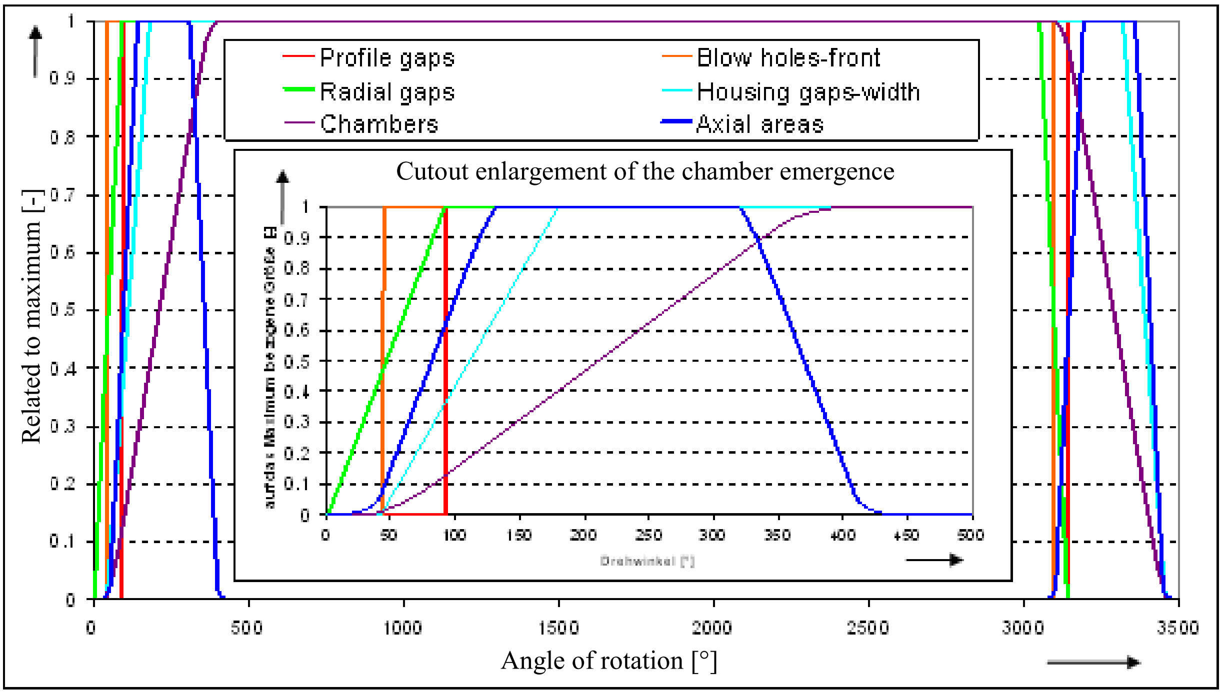


Fig. 1: **Diagram of the chamber model produced automatically by the chamber model generator as a check on the gradient of chamber volume, in- and outlet areas and gap gradients**

The rotor types are classified by the number of rotor lobes, the type of profile and of rotor pitch. With basic rotor types specified in this way, the parameters rotor crown circle, rotor root circle and rotor pitch can be varied. The rotor length l is set as a constant parameter. Besides the geometric parameters for the rotor, the gap height, width and length, have to be specified for all gaps, if they do not result from the geometry of the rotor.

The model files can be graphically prepared with a table calculation program and can thus be simply checked for possible errors, **Fig. 1**.

3.2. Inquiry systematic

The input of the geometric parameters is divided into three parts. First the fundamental geometrical dimensions of the rotor geometry are necessary

1. number of rotor vents
2. number of rotors
3. number of lobes per rotor
4. type of rotor pitch
5. rotor pitch function
6. rotor crown diameter
7. rotor root diameter
8. rotor length

In part 2 gap sizes are examined:

1. height of the housing gaps
2. height of the radial gaps
3. height of the profile gaps
4. gap contour and if necessary parameters for:
   a. housing gap
   b. radial gap
   c. profile gap
   d. blow hole

In part 3 the possibility exists to define a profile.

In the current program version a profile is defined by the choice of a lobe profile pitch angle. By this means a common real profile can be approximated by a trapezium profile. The lobe profile pitch angle is the angle between the lobe profile and the perpendicular to the rotor axis in a radial direction.[1]

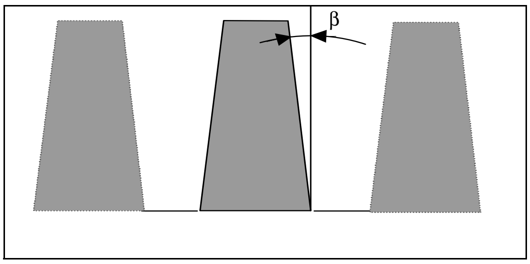


Fig 2. Lobe profile pitch angle at the rotor lobe in sectional view

4. Modeling of gaps

Gaps significantly influence the operating performance and the simulation results of the chamber model. Therefore a precise description of the gaps is necessary, because the clearance mass flow determines the operational behaviour of the vacuum pump, Fig. 3 [4].

For the simulation, the gaps are illustrated by the following outlines [1] which are deposited in the simulation system *KaSim* as a measured data basis, Fig. 3. The housing gap is illustrated by the contour VR3, the radial clearance by contour VR4. The radial gaps have a special characteristic. The VR4 contour describes only a single curvature paired with a plane

[1] *For more information about the automatic chamber model generator and the first simulation results see [7]*

wall. However the real radial gap in the machine is defined by two curvatures, the rotor crown and root diameter. Conversion to the double curvature follows automatically within the program module. The profile meshing gap is described by the contour VR3. The profile gap connects up to 4 capacities, i.e. working chambers to one another.

However the simulation system can calculate only one connection between two capacities. Therefore the profile meshing gap is divided into 4 single connections to ensure a realistic model of the gap. Finally the blow holes are illustrated by the contour row VR9.

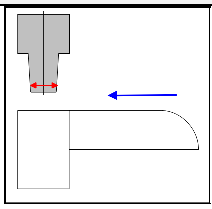
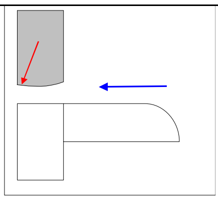
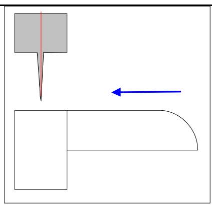
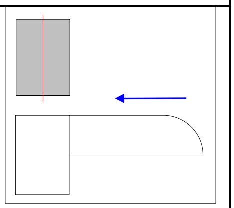
Clearance type and outline form:	Illustration of the outline
housing gap VR3	
radial gap VR4	
blow holes VR9	
profile gap VR9	

Fig. 3: Illustration of the graps of a screw-typ vacuum pump in the chamber model, by different outlines [1]

4.1. Selection of the operating parameters

With the choice of the operation parameters the focus is on the sensitive range of the volumetric efficiency characteristic. In this operating pressure range a change in clearance vacuum flow type occurs inside machines gaps, causing a major change in the volumetric efficiency curve as rotor speed increases. There are different flow forms in the vacuum and the gas flows can be subdivided into three ranges. The viscous flow range is present with high suction pressures and relatively large gap heights. The flow is determined by the pressure ratio at the inlet and outlet clearance. The mean free path of a gas particle is rather short, so that there are more impacts between the gas particles than impacts of particles with gap boundaries. With low suction pressures e.g. < 1 mbar and rather small gap heights molecular flow conditions occur. The molecular clearance flow rate is determined by the pressure difference between the inlet and outlet clearance. The mean free path of the gas particles is rather large and there are only a few impacts between the gas particles compared to impacts between gas particles and gap boundaries. The transition between the viscous and the molecular flow regime is called the “Knudsen” flow or transitional flow regime.

5. Simulation results for the variation of geometric parameters

5.1. Variation of the head and root diameters

The investigation of the rotor geometry is subdivided into the variation of the crown and root circle diameter, the number of rotor lobes and the variation of the rotor pitch. At this point the special characteristics of the thermal simulation system *KaSim* should be explained. The simulation system only calculates the leakage mass flows through machine clearances on the data basis of experimental results. Therefore effects of adsorption and desorption as well as the influence of moving clearance boundaries are not included and modelled. However a verification of the simulation system on a reference machine shows a good degree of consistency of measured values and simulation results in a suction pressure area above 1 mbar (see [5] and [6]). Consequently, in this article only simulation of the operating behaviour with a suction pressure above 1 mbar is considered. The values of the gap heights remain unchanged in all models.

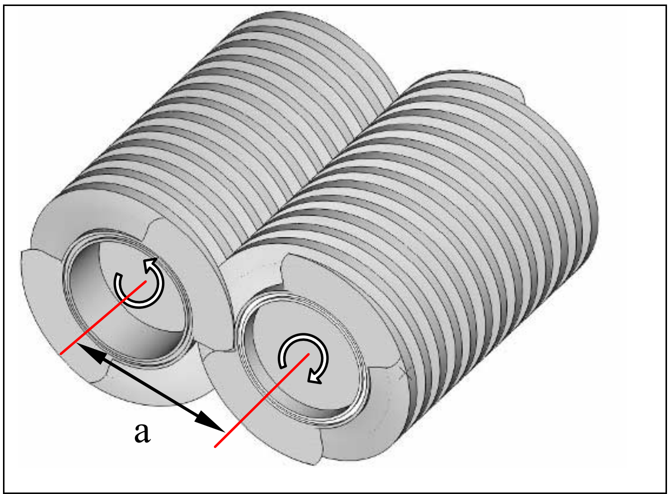


Fig. 4: Centre distance as a result of the given head and root diameter of the rotor

The rotor crown diameter is varied between 80 and 500 mm, the rotor root diameters within a range of 50 to 400mm. The chosen width of the variations is orientated to a test machine and reflects technically meaningful geometric parameters. To analyse the effects of rotor geometry parameters the first calculations of the thermodynamic processing are given for the following fourteen combinations of crown and root diameter. The centre distance (**fig. 4**) of the rotors describes a dependent parameter due to the chosen independent geometric rotor parameters. The given equation (**eq. 2**) shows the calculation instruction for the axle rotor distance.

$$a = \frac{d_K + d_F}{2} + h_{Radialspalt} \qquad \textit{\textbf{eq. (2)}}$$

5.2. Variation of the tip circle diameter

Table 1: Models with a common root diameter

Model:	Tip cylinder diameter:
D3	150 mm
D4	160 mm
D7	ca. 200 mm[2]
D10	300 mm
D12	500 mm

For the investigated variations of the crown cylinder diameter the root diameters remains unchanged, **table 1**. Here only 5 of the fourteen models which have the same root diameter will be considered. The definition of the operating parameters with the variation of the tip cylinder diameters encounters the problem of the comparability of the different

[2] *Due to confidentially no accurate value for the tip cylinder diameter are presented here.*

models. For various rotating machine parts the operation limit is determined not only by the number of revolutions, but also by the centrifugal force that results from the circumferential velocity. Therefore the comparison of the rotor models is presented at constant revolutions and also with a constant circumferential velocity.

A comparison of the simulated volumetric efficiencies as a function of the intake pressure at 3000rpm results in **fig. 5.** All models with a larger tip cylinder diameter than the reference machine show a higher volumetric efficiency. The models with a small tip cylinder diameter indicate a smaller volumetric efficiency and for the given number of revolutions an ultimate suction pressure level above 800mbar is attainable.

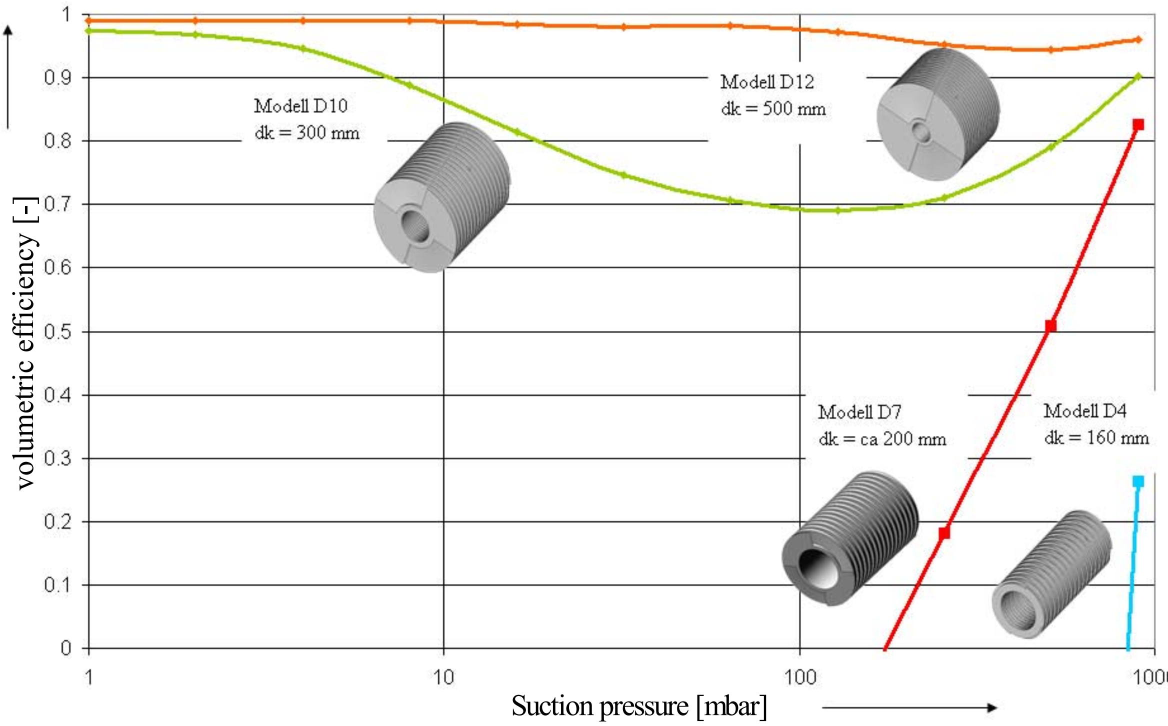


Fig. 5: Simulation result with variation of the tip cylinder diameter and rotor speed of 3000rpm

5.2.1. Comparison of the simulation results at constant rotor speed

A comparison of the volumetric efficiencies with a circumferential velocity of 40 m/s for the head of the lobe is shown for the volumetric efficiency characteristic in **fig. 6**.

The models with a tip cylinder diameter below 200 mm result in less volumetric efficiency than the reference model D7. The reasons for the differences in the volumetric efficiency depend on the number of working chambers and the contour, number and area of the clearances between the chambers. For the variation of the tip cylinder diameters the number of clearances remained unchanged. The form of the clearances along the main flow direction only changes slightly with variation of the tip cylinder diameters. Therefore the influencing parameters which can be examined are reduced to the chamber volume and the clearance area perpendicular to the main flow direction. Here a first definition of design criteria for a screw-spindle vacuum pump emerges.

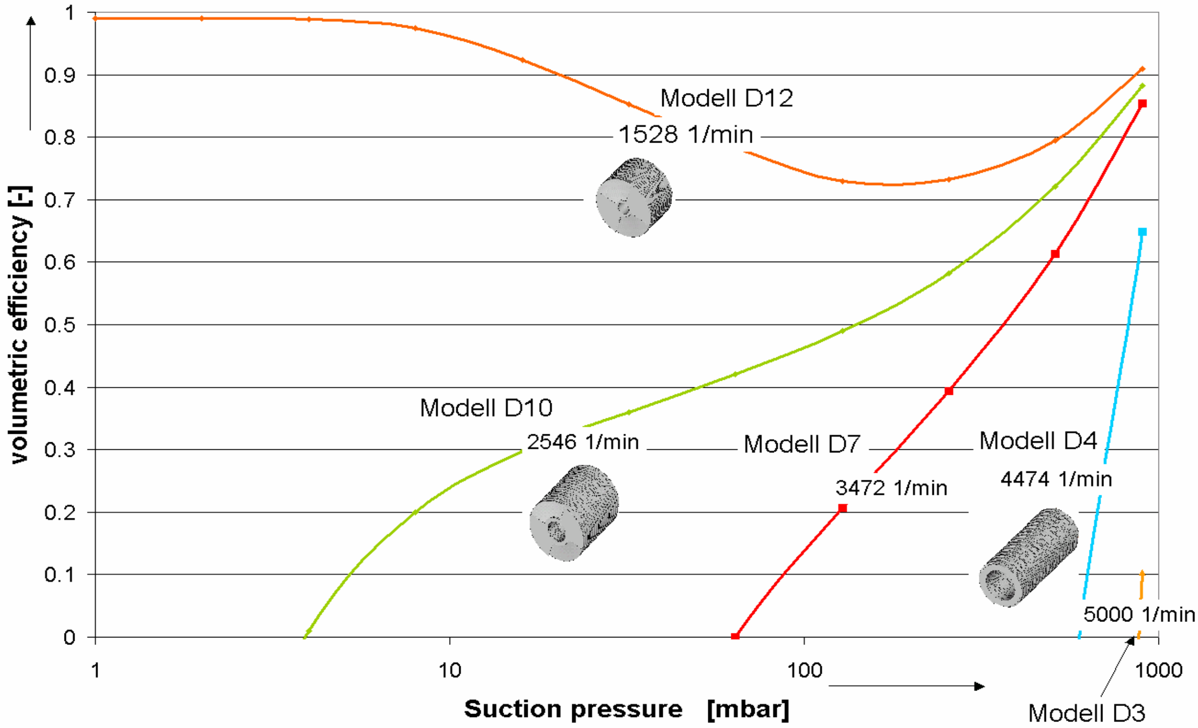


Fig. 6: Simulated volumetric efficiency for a constant rotor speed of 40 m/s and rotor geometries with varied tip cylinder diameter.

5.3. Variation of the root circle diameter

Table 2: Models with a constant head cylinder diameter of 220 mm

model:	***root diameter***
D5	**50 mm**
D6	**100 mm**
D7	**ca. 150 mm**
D8	**180 mm**
D9	**200 mm**

In order to analyse the influence of the root circle diameter of the rotors, the models listed in **table 2** are investigated. The simulation results are presented for a constant rotor speed of 4000rpm (**fig. 8**). The rotor geometries with a smaller root diameter than the reference machine show a higher volumetric efficiency and the ultimate suction pressures are calculated below 1 mbar.

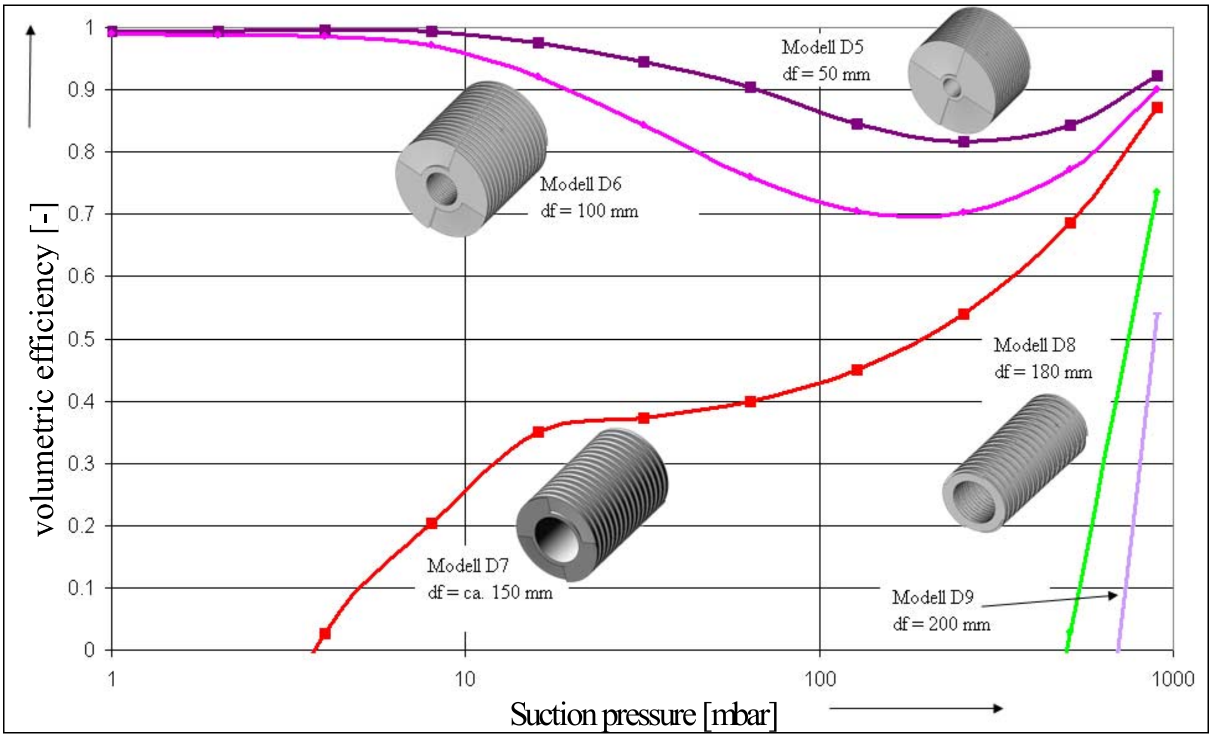


Fig. 8: Simulation result with variation of the root diameter at 4000rpm

5.4. Optimisation of crown and root circle

The chamber volume changes with the square of the rotor crown circle. **Figure 9** shows that the combination of the largest crown diameter and smallest root diameter does not always result in the largest chamber volume. The three volume curves based on different representative root circles intersect. The centre distance of the model with a theoretical root

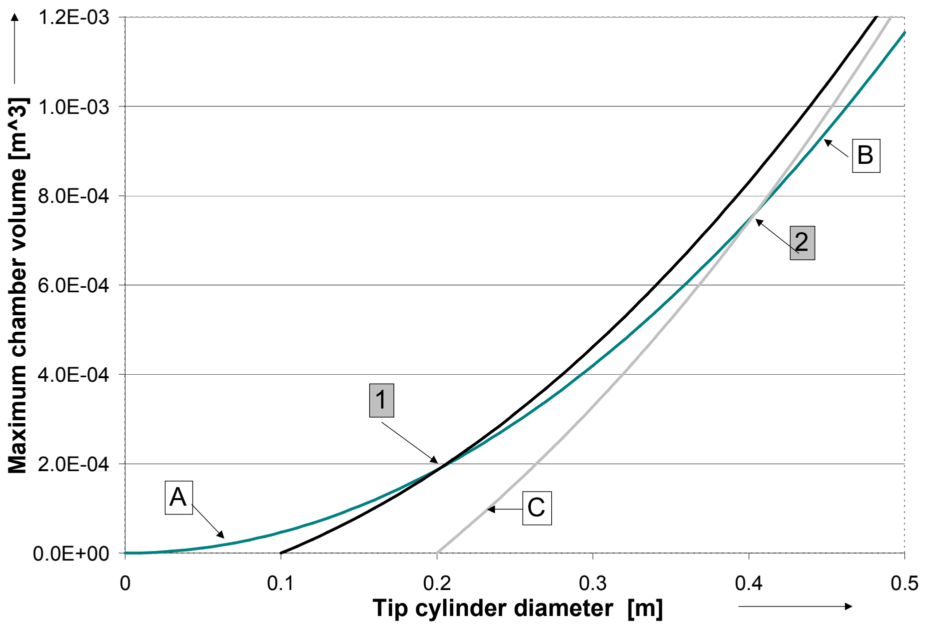


Fig. 9: Chamber volume depending on the crown diameter, for different root diameters

A: root diameter ***d_f =0 mm***
B: root diameter ***d_f =100 mm***
C: root diameter ***d_f =200 mm***
1: Intersection between the volume line A and B
2: Intersection between the volume line B and C

circle radius of zero equals a crown diameter of 200mm. At the first intersection point the model with a root diameter of 100mm and a head diameter of 150mm has the same chamber volume. These results suggest an optimal combination of crown and root diameters regarding the ratio of chamber volumes to clearance area. To determine this optimum the chamber volumes and the clearance areas based on multiple combinations of crown and root circle are calculated. The clearance heights are assumed to be constant in the calculation of the clearance area, and they are not varied in conjunction with the crown and root diameters. **Fig. 10** shows the ratio of chamber volume to clearance area in relation to the diameters of the crown and root circles of the rotors.

The ratio of chamber volume to clearance area rises with increasing crown circle diameter as expected. For a given crown circle diameter an optimal root diameter can be determined, which leads to a maximum ratio of chamber volume to clearance area. The maxima increase in line with the crown circle diameter, (fig. 10). For example the crown circle diameter of the reference machine has an optimal root diameter of 60mm. The models with a crown diameter of 500mm result in an optimal root diameter of 130mm. The root diameter of 130 mm is within the range of simulated models and can therefore be evaluated by the simulation.

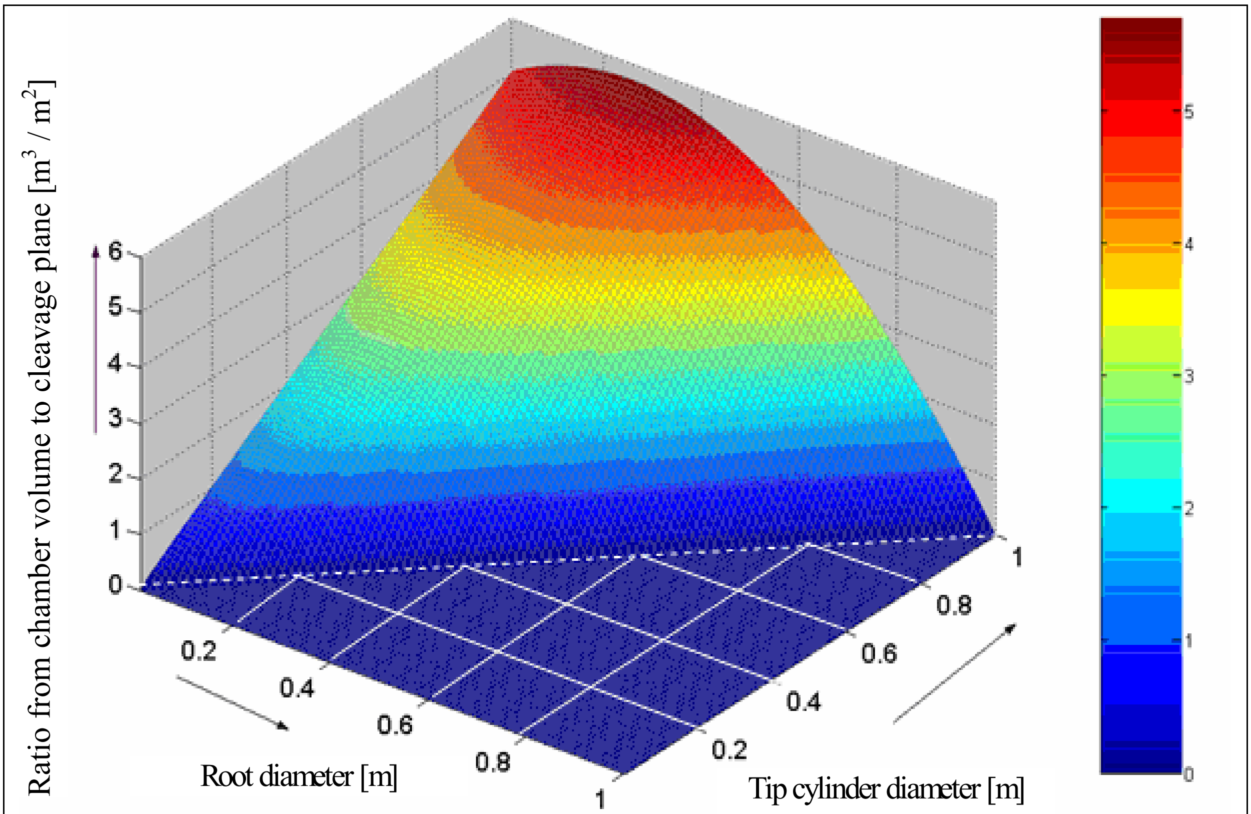


Fig. 10: Ratio of chamber volume to clearance area depending on the root and crown diameter, assuming constant casing gap and radial gap heights

The analysis of the simulation results of the models with a crown circle diameter of 500mm at a rotor speed of 1910rpm shows differences between the volumetric efficiency of the individual models (**fig. 11**).

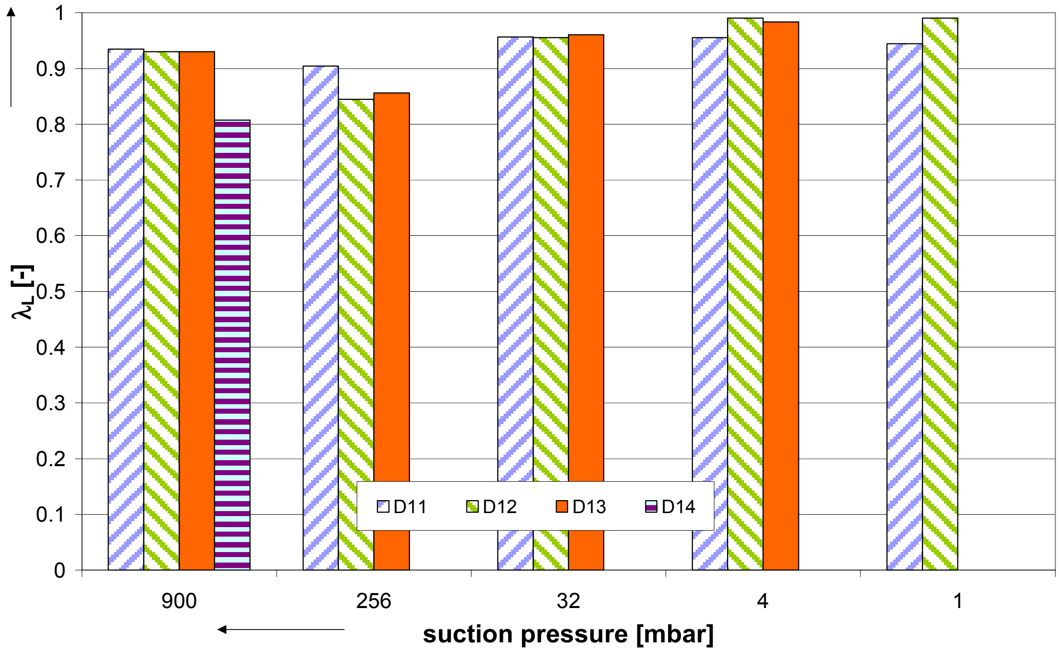


Fig. 11: Simulation results of the models with a crown circle diameter of 500 mm at a rotor speed of 1910 rpm. Constant housing gap height for all models

While the simulation results show a negative volumetric efficiency for model D14 at a suction levels of 256mbar, the volumetric efficiency of all other models exceeds 80%. The simulation shows that the model D 12, using the theoretically optimum root diameter, is not the best in the simulation at a suction level of about 32mbar. Models D11 and D13 have a better volumetric efficiency in this area of suction level than D12. Below a suction pressure of 32mbar, model D12 achieves the highest volumetric efficiency. This results from the viscous flow through the clearances at a suction pressure of over 32mbar. It seems that the length of clearance in flow direction cannot be disregarded in the calculation of the optimum ratio of chamber volume to clearance area, when it comes to viscous flow.

6. Number of lobes

All simulation results described in the previous sections are based on a two-lobe-trapezoidal-profile of the screw spindle, which is derived from the reference machine at the Fachgebiet Fluidtechnik. The market for screw-spindle pumps consists of one and two lobe rotor profiles. The question is, which of these two different profiles is thermodynamically more favourable and thus achieves the better operating results. To obtain accurate statements about the achievable suction capacity or volumetric efficiency, a three-lobe rotor profile was added to the simulation models. For all selected models the rotor length and the diameters are the same, leading to equal theoretical suction characteristics for all models. The computation of the suction characteristics for different numbers of rotor lobes results in the suction characteristics diagram, **fig. 12**, calculated at a rotor speed of 5000 rpm.

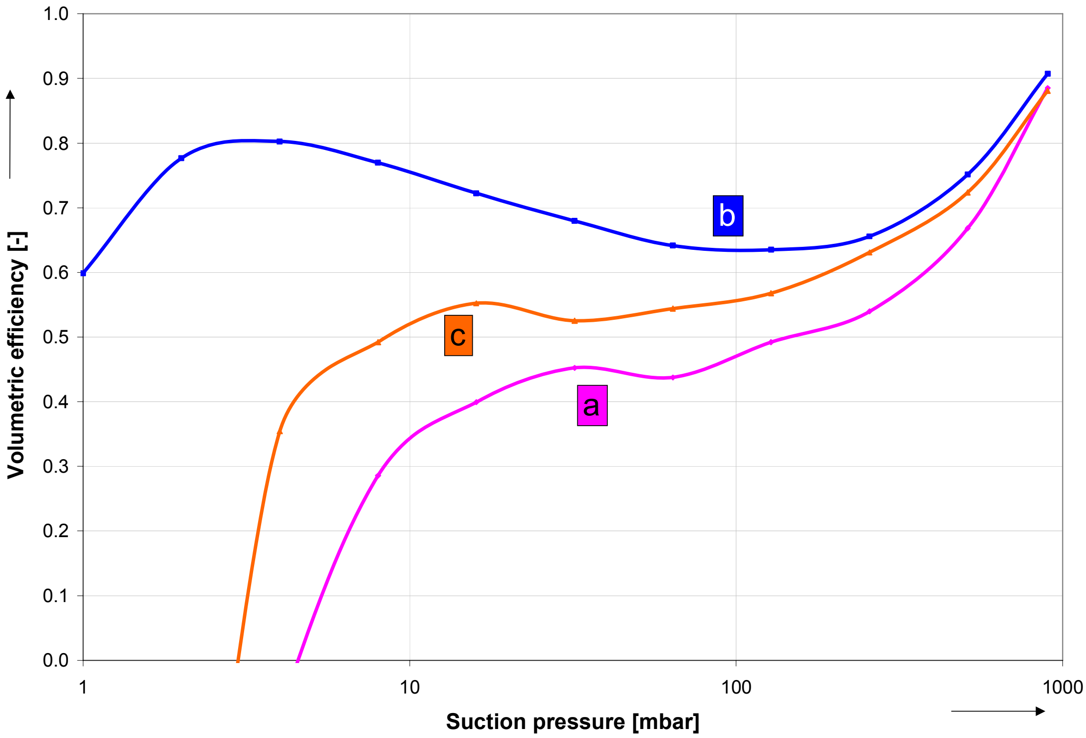


Fig. 12 Volumetric efficiency depending on the intake pressure for rotors with different numbers of lobes at a rotor speed of 5000 rpm

- **a. rotor with one lobe**
- **b. rotor with two lobes**
- **c. rotor with three lobes**

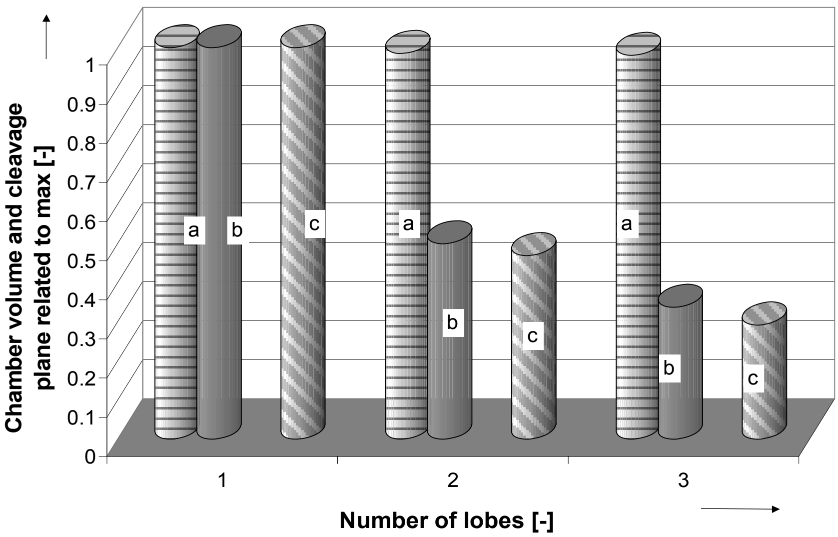


Fig. 13 Comparison of the chamber volume, the clearance area and the length of the housing gap related to one chamber depending on the number of rotor lobes.

- **a. Clearance Area**
- **b. Chamber volume**
- **c. Housing gap with**

The chamber model of the two - lobe rotor shows the highest volumetric efficiency in this comparison. An explanation for the low volumetric efficiency of the other two models lies in the number of clearances between high pressure side and low pressure side and the ratio of volume to clearance area for each chamber, which decreases with a rising number of lobes, **figure 13**.

The reason for this behaviour is particularly the reduction in the chamber volume along with nearly constant clearances. The number of radial and profile gaps between the high pressure and the low pressure side is also constant and does not therefore lead to a decreasing pressure ratio at the individual gaps. The number of housing clearances decreases as the number of lobes increases. Presumably the pressure ratio at the gap increases. If we examine housing clearances, we find that a decrease in the number of lobes leads, in addition to an increase in the length of the gaps, to reduced flow rates. For example, the standardized mass flow through a housing gap is at a pressure ratio of 0,5 and a gap inlet pressure of 32mbar for a one lobe rotor 0.18, for a two-lobe rotor 0.33 and for a three-lobe rotor 0.40. A reason for the decrease in volumetric efficiency with smaller numbers of lobes is the duration of the gas exchange. The longer the gas exchange takes, the more gas from the preceding chambers may flow back through the gaps. A short gas exchange duration on the other hand may lead to an incomplete charge of the chamber resulting in a decrease in volumetric efficiency. The angle of the chamber volume varies in the models between 540° for a one-lobe rotor, to 450° for a two-lobe rotor and down to 420° for three-lobe rotor. **Figure 14** gives an overview of the direction of flow between the suction area and the chamber.

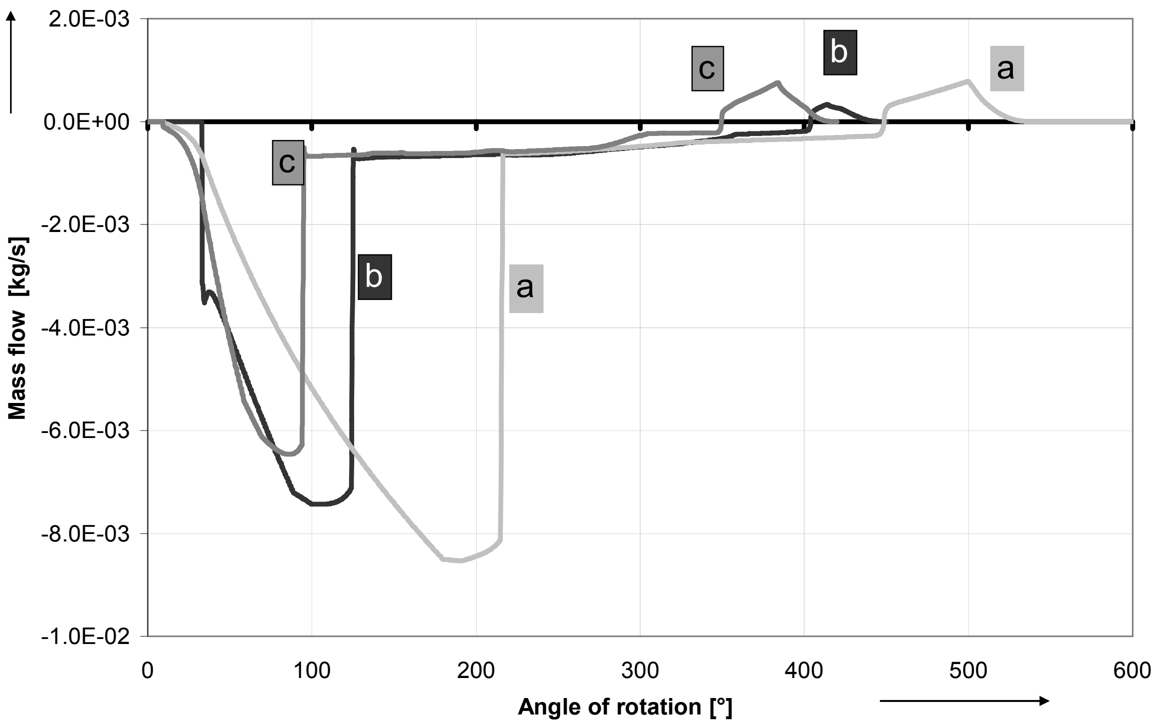


Fig. 14 Currents within the suction area of the pump for different numbers of lobes on the rotor:

a) one lobe rotor
b) two lobe rotor
c) three lobe rotor

Negative values mean that there is a flow from the suction area into the chamber. Positive values mean that there is a backflow from the chamber into the suction area of the pump.

The picture shows the total mass flow through the inlet area. Negative values represent a mass flow into the chamber, positive values show a backflow into the inlet port. All simulations show an initial mass flow into the working chamber, with a noticeable jump at the transition from the axial to the diagonal inlet surface. At the end of the chamber configuration the backflow into the inlet port predominates.

This backflow becomes minimal for a rotor with two lobes compared with the other rotor designs. These results suggest that the gap flow in relation to the chamber volume has lowest value for the two-lobe rotor. Therefore it makes sense from the point of the thermodynamics to choose a rotor with two lobes.

7. Variation of the pitch of the rotor

Within the framework of the variation of the rotor gradient the rotor length is kept constant, so the wrap angle of the lobe is changed. Through the modification of the wrap angle the number of working chambers also changes on the rotors. The number of working chambers

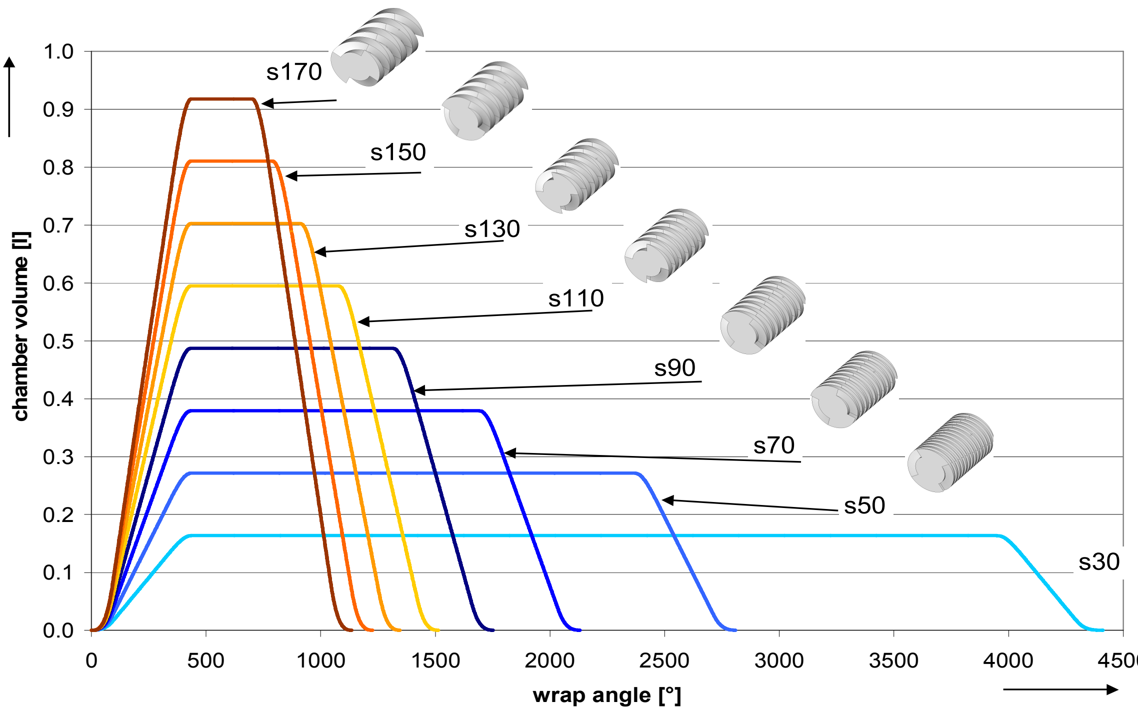


Fig. 15: Models with variation in the rotor pitch and a constant rotor length l_{rotor} = const.

by given gradient is changed according to equation **eq. (2)**.

$$j_{AK} = \left(\frac{\frac{l \cdot 360°}{s} + \varphi_{chamber\_creation\_angle}}{360°} \right) \cdot n_{Rotors} \cdot n_{teeths} \qquad \textbf{eq. (2)}$$

The range of rotor gradient is varied between 30 and 170mm. This range is divided up into nine more models (**fig. 15**).

It is to be expected that the decrease in working chambers with rotor gradient becoming larger affects the volumetric efficiency negatively because the number of gaps switched in series is reduced and thus the pressure ratio increases above individual clearance. The widening of the rotor-lobe-crown enhances this dependency in the case of increasing rotor gradient. The broadening of the rotor-lobe crown enhances this tendency in the case of increasing rotor gradient. A broad rotor-lobe crown entails longer housing clearances, which leads in turn to a greater throttle effect. The increase in the working chamber depends on the relationship between chamber volume and the are of the gap. A view of the modifications to chamber volume and wrap angle is given in **fig. 16**, which also presents the simulation results for the calculated suction capacity above suction level. The maximum at suction capacity is near high suction levels for models with a rotor gradient of 170mm.

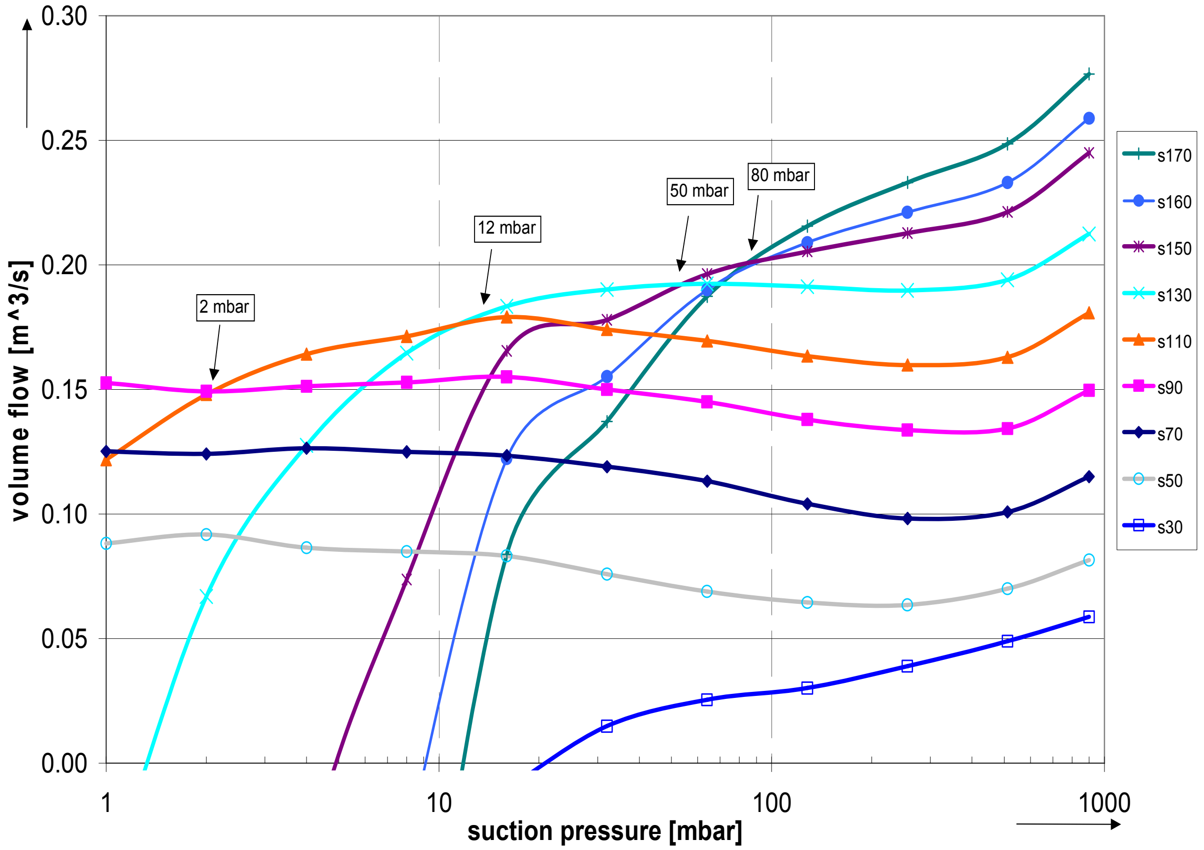


Fig. 16 **Relationship of chamber volumes, total gap plane and the relationship of chamber volumes to the gap plane for the models with variation in the upward gradient**

Below a suction level of 80mbar the model with optimal suction capacity to gradient switches to 150mm. At a further suction pressure reduction below 5mbar there is a switch to model s130, and below 3mbar the model s110 supposed to be superior.

During the design of screw spindle vacuum pumps it is possible to select the gradient, with knowledge of the final pressure to be expected, so that the suction capacity shows a maximum value. A similar suction pressure index can be constructed with the help of a simulation system in which the optimal gradient can be determined easily for each rotation speed.

8. Constructional consequences

The calculated tendencies show that at high suction levels a large gradient is advantageous. In these operating conditions the pressure ratio is relatively small above the rotor and the few gaps between suction and discharge suffice for sealing of the gaps. At low suction pressures the pressure ratio rises above the rotor. In the case of high gradients as the result of high pressure ratios above a gap high gap, mass flows continue into the suction area of the pump and reduce both the volumetric efficiency and the suction capacity.

9. Interpretation of the operating parameters

The design of the operating parameters determines the target volumetric efficiency λ_L, the final suction pressure and the suction capacity S. The connection between rotation speed and suction property is arrived at via the volumetric efficiency achieved and the chamber volume described (**eq. (4)**.

$$S = V_{AK} \cdot n \cdot \lambda_L \qquad \textbf{eq. (4)}$$

With the models and rotation speeds under consideration no significant suction restriction could be proved with the aid of the simulation. That can run out from its not resulting in an incomplete comb fulfilment ant any of the considered models that volumetric efficiency reducing affects. An increase in rotation speed leads therefore to higher efficiency by reducing the time during which the working fluid can flow from a chamber with high pressure into one with low pressure, i.e. efficiency improves as the rotation speed increases.

With all simulations a downward shift in volumetric efficiency characteristics depending on falling rotation speed can be observed in the index. At the critical point the suction level drops to values under 1mbar. At sufficiently high rotation speeds the volumetric efficiency achieved is more than 80% with all models, at a suction level of 1mbar or above. The suction level also has an influence on volumetric efficiency and consequently on the suction capacity of the pump. If a model reaches a pressure that leads, in combination with clearance heights, to Knudsen-flow in the gaps on the suction side, the volumetric efficiency goes up in the case of a further reduction in suction levels, since the clearance mass flow are reduced by the presence of cross-currents. During the design of screw-spindle vacuum pumps the required suction capacity and the final pressure are important. The selection of the geometry should be carried out as described above. Fundamentally a rotation speed as high as possible is to be aimed at since at high speeds the transfer flow shifts from viscous to molecular, with a corresponding rise in efficiency. Rotation speed is restricted by the bearing characteristics, the centrifugal forces that act on the rotor and potential thermal problems. The thermal problems arise particularly through the reverse currents of heated gas flowing from discharge area into the rotor chambers, there to be compressed because of the large quantities of gas occur in the screw-spindle vacuum pump.

To avoid thermal damage, therefore, it is inadvisable to choose a small machine running at high rotation speed. Heat exchange surfaces between gas and rotor surfaces clearly functions differently from heat dissipation in the machine itself, which is problematic in the case of small components. For further investigation of the thermal situation in screw-spindle vacuum pump it would be advisable to examine also the ratio of chamber volumes to possible heat exchange surfaces. This will be a component of further investigations.

For the reasons outlined above, the machine chosen for a particular suction capacity should always be as large as possible, and rotate fast enough to achieve a sufficiently high level of volumetric efficiency.

10. Literatur

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