Experimental and theoretical investigation of screw machines as vacuum blowers

A Nikolov, K Nadler, A Brümmer
TU Dortmund, Chair of Fluidics, Germany

ABSTRACT

To attain lower pressures in the fine and adjacent high vacuum roots vacuum pumps are used in conjunction with a forevacuum pump in almost any application in the so-called "blower" operating range. At this point, the already advanced screw compressor technology may be applied to "blower" applications too.

Systematic experimental investigations as well as numerical simulations (Kasim) of the application potential of the screw machine as a blower vacuum pump have been realized. Geometric and operating parameters have been varied. Both the maximum achievable compression ratio and the pumping speed have been investigated. Finally the theoretical and experimental results have been compared and a basic understanding of the effective physical-technical mechanisms has been presented.

SYMBOLS

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Unit</th>
<th>Meaning</th>
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<tbody>
<tr>
<td>d</td>
<td>[m]</td>
<td>lateral expansion of the gap cross-section</td>
</tr>
<tr>
<td>Ko</td>
<td>[-]</td>
<td>compression ratio at zero delivery</td>
</tr>
<tr>
<td>Ko&quot;</td>
<td>[-]</td>
<td>relative compression ratio at zero delivery</td>
</tr>
<tr>
<td>Kn</td>
<td>[-]</td>
<td>Knudsen number</td>
</tr>
<tr>
<td>i</td>
<td>[m]</td>
<td>mean free path length of a gas particle</td>
</tr>
<tr>
<td>M</td>
<td>[Nm]</td>
<td>drive torque</td>
</tr>
<tr>
<td>n</td>
<td>[min⁻¹]</td>
<td>rotational speed</td>
</tr>
<tr>
<td>pₚ</td>
<td>[bar]</td>
<td>gap pressure</td>
</tr>
<tr>
<td>pₑ</td>
<td>[bar]</td>
<td>atmospheric pressure</td>
</tr>
<tr>
<td>pₑₚ</td>
<td>[bar]</td>
<td>(gap) inlet pressure</td>
</tr>
<tr>
<td>pᵥ</td>
<td>[bar]</td>
<td>forevacuum pressure</td>
</tr>
<tr>
<td>qₙ</td>
<td>[mbar·l·s⁻¹]</td>
<td>leak rate</td>
</tr>
<tr>
<td>S</td>
<td>[m³·s⁻¹]</td>
<td>suction speed, clearance volume flow</td>
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<tr>
<td>Sₑₚ</td>
<td>[m³·s⁻¹]</td>
<td>theoretical suction speed</td>
</tr>
<tr>
<td>tₚ</td>
<td>[s]</td>
<td>operating time</td>
</tr>
<tr>
<td>T</td>
<td>[K]</td>
<td>temperature</td>
</tr>
<tr>
<td>vᵢ</td>
<td>[-]</td>
<td>inner volume ratio</td>
</tr>
<tr>
<td>λₑ</td>
<td>[-]</td>
<td>delivery rate</td>
</tr>
<tr>
<td>φ₀</td>
<td>[°]</td>
<td>wrap angle</td>
</tr>
<tr>
<td>φₑₚ</td>
<td>[°]</td>
<td>wrap angle of the male rotor</td>
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1 INTRODUCTION

In vacuum technology, very different pump designs are successful; the field of application of different types of vacuum pumps is primarily determined by their achievable ultimate pressure and their suction speed. Combinations of pumps are used for the generation of lower pressures in the fine vacuum and the adjacent high vacuum while high pumping speed is provided at the same time. In connection with a backing pump, in most cases a screw pump, Roots vacuum pumps in the so-called "blower"-operation are used as standard in almost all fields of applications (1).

At this point, the already advanced screw compressor technology can be applied in "blower"-operation in vacuum technology. Due to the energetically more favourable process control, advantageous operation behaviour of the screw machine at low gas densities is to be expected. In addition, the small design volume of the screw vacuum pump due to higher machine speeds is expected to result in cost advantages. In comparison to the Roots vacuum pump applied so far, the availability offers the prospect of three significant advantages: reduction of the number of pumps in the unit, of the pumping speed of the backing pump and of the design volume of the entire pump system.

Against this background, the design of the screw machine’s geometry as well as its operating parameters gains increasing importance. Optimal design of geometry and thermodynamics based on experimental investigations is not expedient with regard to the high number of experimental vehicles required and the high expenditure related to it. Instead, the design of the geometrical parameters and of the thermodynamic boundary conditions by means of numeric simulation (KaSim) is to be preferred. A prerequisite for this is a validation of the numeric simulation based on the exemplary comparison with experimental results of a reduced number of test machines.

2 THE SCREW VACUUM PUMP

A screw machine that fulfills the requirements of the most flexible geometry with regard to the desired experimental investigations of the machine variations is the screw-type supercharger GL 51.2 developed at the chair of fluidics (2), (3). In this respect, the screw-type supercharger GL 51.2 provides the following advantages:

- constructionally simple, modular design (simple variation of component geometries);
- reduction of intermesh clearance height compared to conventional screw machines (non-synchronized rotor operation);
- dry-running operation ensured due to wear-protective coating of the rotor surfaces.

This type of machine is particularly interesting for the consideration of the rotational speed, since speeds of up to 30,000 min⁻¹ have already been realized in charging mode. Furthermore, the screw rotors rolling off against each other show a clearly lower profile meshing clearance height than conventional screw machines with synchronized rotor drive. Accordingly, a positive influence on the operating behaviour of the machine is to be expected with regard to the reduction of the clearance mass flow through the profile clearance. The rotor wear progressing as a

1 GL(GearLess) 51.2 – This is a dry-running non-synchronized supercharger with an axis-center distance of 51.2 mm, developed, built and tested at the chair of fluidics of the TU Dortmund.
function of the operating time shall be less due to the reduced pressure in the vacuum application.

Due to its modular structure, the screw-type supercharger had to be modified in view of the optimization of the tightness between the inside of the charger and the environment for use in vacuum, Figure 1. Within the framework of the redesign, for the vacuum variant of the screw-type supercharger \textit{SVB 51.2 (Screw Vacuum Blower 51.2)}, first O-ring seals were inserted between all housing modules. This design measure was accompanied by a direct increase in housing wall thickness. Furthermore, the rotor housing originally consisting of two parts (rotor housing and bearing module) was redesigned to one single module. As a result, the disadvantage of a three-dimensional clearance in the area of the outlet area on the pressure side could be compensated. On the other hand, this design change limits the possibility of an increase in the internal volume ratio of the machine, since, for manufacturing reasons, exclusively radial outlet areas can be manufactured. The identifiable axial portion of the outlet area merely provides a connection of the rest of the gas volume remaining in the chamber to the pressure port after the closure of the radial outlet area.

The sealing of the driving shaft out of the screw-type supercharger represents another potential leak. Therefore, in contrast to the prototype version of the screw-type supercharger, the machine was equipped with two radial shaft seals instead of one, which seal against atmospheric pressure. The consideration of a pronounced wear-related time dependence of the leakage has not been included in the investigation so far. However, it probably represents an influential factor which should not be underestimated, in particular with regard to longer operating times at high speeds. Furthermore, in order to reduce leakages, the passage of the female rotor was closed at the pressure side opposite the coupling chamber.

\begin{figure}[h]
\centering
\includegraphics[width=\textwidth]{screw_type_supercharger}
\caption{Exploded view of the screw-type vacuum pump \textit{SVB 51.2}}
\end{figure}
Based on the design measures taken, the overall situation regarding the tightness of the newly developed vacuum variant SVB 51.2 of the screw-type supercharger could be improved in comparison to the prototype version GL 51.2. The leak rate of the test machine could be reduced to \( q_0 = 1.61 \times 10^{-2} \text{ mbar} \cdot \text{l} \cdot \text{s}^{-1} \). With this, the advantages of a flexible geometry for the variation of geometric parameters were combined with the safety of reliable results for the vacuum application of the screw-type machine.

3 EXPERIMENTAL INVESTIGATION

In the following, the experimental investigation of the screw-type vacuum pump will be discussed. Within this context, a short overview of the test setup, the measuring equipment and the boundary conditions of the experiments made is given. The analysis and interpretation of the experimental results are first carried out based on the compression ratio \( K_0 \) achieved by the SVB 51.2 at zero delivery. Among others, also the temperature-dependent operating behaviour of the screw machine with regard to the parameter \( K_0 \) will be discussed. This is followed by the representation and discussion of the delivery behaviour of the pump based on the volumetric efficiency of the machine \( \lambda_s \).

3.1 Test setup and plan of measuring points

Figure 2 gives an overview of the test setup and schematically shows the measurement instrumentation applied. A screw spindle vacuum pump with constant rotor lead\(^2\) was used as forevacuum pump, which enables forevacuum pressures of down to about 0.03 mbar to be reached. By means of throttling valves on the suction side of the screw-type vacuum pump SVB 51.2, the volume flow resp. the pump inlet pressure can be adjusted. Moreover, the test stand is equipped with a control and measurement computer.

![Figure 2: Test setup and measuring equipment](image)

\(^2\) Screw-type vacuum pumps are dry-running rotary displacement pumps for inlet pressures in the low and fine vacuum range. The screw spindle can have both constant and variable lead, nowadays however the variable lead is state of the art. (1)
For the measuring of the pressures in the suction resp. pressure port, capacitive pressure transducers, arranged in cascades with five pressure transducers each, for different measuring ranges – up to 1000 mbar, 100 mbar, 10 mbar, 1 mbar and 0.1 mbar – are applied. Furthermore, the suction pipe is equipped with flow monitors which help measuring the air volume sucked in from the atmosphere for the determination of the suction speed. In the suction and pressure port of the screw-type vacuum pump, thermocouples are installed for the measuring of the gas temperature. For reasons of operating safety as well as for the measuring of the temperature distribution along the rotors in the machine housing, thermocouples were installed in the area of the bearing on the suction side and in the rotor housing. For the monitoring of the bearing, the SVB 51.2 is equipped with measuring instruments for vibration monitoring, which enable the early detection of damaged bearings and the protection of the screw-type supercharger against further damages during operation. Furthermore, the drive torque and the machine speed are measured immediately in front of the drive shaft of the supercharger and the results are further used for the calculation of the driving power of the charger.

3.2 Boundary conditions of the experimental investigation

For the investigation of the operating behaviour of the screw-type vacuum pump SVB 51.2 in "blower" operation as well as for the determination of the suction speed, basically general operating parameters such as rotational speed, forevacuum pressure and inlet pressure in ultimate pressure operation are systematically varied, independent of the machine geometry. At this point, the lowest achievable forevacuum pressure is limited by the operating properties of the forevacuum pump applied.

On the part of the machine geometry, the main dimensions of the rotors – length and diameter – are kept constant in the experimental investigation, thus enabling the effort for design and remodeling as well as the costs to be kept low. The focus of the experimental investigation is on the variation of the internal volume ratio $v_i = 1.1, 1.47, 2.0$.

The machine variant with $v_i = 1.47$ and a wrap angle of the male rotor of $\varphi_m = 200^\circ$ was adopted from the charger (GL 51.2) and represents the basic variant for the experimental investigation (see Table 1). Regarding the selection of the remaining two volume ratios, the position of the control edge on the suction side is determinant. An internal volume ratio of $v_i = 2.0$ provides a still sufficiently radial outlet area, while from the manufacturing point of view, it is realistically feasible down to $v_i \approx 1.1$ due to its strongly increasing dimension.

### Table 1: Geometrical data of the basic variant of SVB 51.2 (3)

<table>
<thead>
<tr>
<th>Description</th>
<th>Unit</th>
<th>Male rotor (MR)</th>
<th>Female rotor (FR)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of lobes</td>
<td>[-]</td>
<td>3</td>
<td>5</td>
</tr>
<tr>
<td>Length of the rotor profile part</td>
<td>[mm]</td>
<td></td>
<td>101</td>
</tr>
<tr>
<td>Axis-center distance</td>
<td>[mm]</td>
<td>51.2</td>
<td></td>
</tr>
<tr>
<td>Diameter</td>
<td>[mm]</td>
<td>71.8</td>
<td>67.5</td>
</tr>
<tr>
<td>Wrap angle $\varphi$ (direction)</td>
<td>[°]</td>
<td>200 (left)</td>
<td>120 (right)</td>
</tr>
<tr>
<td>Lead</td>
<td>[mm]</td>
<td>181.8</td>
<td>-303</td>
</tr>
<tr>
<td>Internal volume ratio $v_i$</td>
<td>[-]</td>
<td>1.47</td>
<td></td>
</tr>
<tr>
<td>Displaced volume per MR-rotation</td>
<td>[cm$^3$]</td>
<td></td>
<td>286</td>
</tr>
</tbody>
</table>

3.3 Compression ratio $K_o$ at zero delivery

A relevant parameter to characterize the operating behaviour of the screw-type vacuum pump in "blower" operation is the compression ratio $K_o$ at zero delivery of the machine (1). The pump parameter $K_o$ is measured by sealing the suction side of the screw-type vacuum pump. Here, with the machine running, an ultimate pressure arises at the suction side, which depends on the given operating
parameters. The compression ratio at zero delivery \( K_0 \) can be defined as the ratio of forevacuum to inlet pressure:

\[
K_0 = \frac{P_f}{P_E}
\]

Eq. 1.

Below, the experimental investigations regarding the compression ratio \( K_0 \) are shown as a function of the operating parameters, i.e. forevacuum pressure \( P_f \) and rotational speed \( n \) of the machine as well as of one of the selected geometrical machine parameters – the internal volume ratio \( \nu_i \).

Prior to the closer analysis of the experimental data, priority is given to the temperature-dependent operating behaviour of the screw-type vacuum pump under consideration of the compression ratio applied to the machine. This turns out to be necessary with regard to the subsequent interpretation of the measured data and the comparison of the experimental results with those of the simulation.

**Figure 3: Compression ratio \( K_0^* \) as a function of time \( t \) for the basic variant of SVB 51.2**

The \( K_0^* \)-curve\(^3\) as a function of the operating time \( t \) for \( n = 6000 \text{ min}^{-1} \) and \( n = 15,000 \text{ min}^{-1} \) and at a forevacuum pressure of \( P_f = 1 \text{ mbar} \) and \( P_f = 5.6 \text{ mbar} \) is shown in Figure 3. Irrespective of the given operating conditions, an increase in \( K_0^* \) can be observed with increasing operating time. The higher compression ratios are attributable to the fact that, both in the steady and unsteady case, the screw rotors are subjected to stronger thermo-mechanical deformation than the enclosing housing. The rotors show stronger expansion than the housing, and the gap heights (mainly at the housing and the front gap), and thus the lossy gap flows, decrease.

\(^3\) The results of the experimental and theoretical investigations regarding the compression ratio \( K_0 \) are illustrated, for secrecy reasons, by the standardized compression ratio \( K_0^* \). At this point, it is defined as \( K_0^* = \frac{K_0}{K_{0,\text{max},6000}} \), with \( K_{0,\text{max},6000} \) corresponding to the maximum of the determined \( K_0 \)-value for the examined machine variant with an internal volume ratio of \( \nu_i = 1.47 \) for a rotational speed of \( n = 6000 \text{ min}^{-1} \).
This causes a reduction of the machine's suction pressure $p_e$ and with this an increase in the compression ratio. The fact that the relative increase in $K_0^*$ decreases with decreasing forevacuum pressure at constant rotational speed $n = 15000 \text{ min}^{-1}$ is due to the decreasing gas density and the related decrease in the conductance of the gap (1), (4).

Since the comparison of the simulation with the experiment in chapter 4 must take place under the same boundary conditions and the gap situations in the screw-type vacuum pump are known in their "cold state"", it is indispensable to determine the experimental data for a (quasi) "cold" machine, too. Thus, the result of the measurements of the $K_0^*$-increase as a function of the operating time $t$ were permissible measuring times, which vary according to machine speed and forevacuum pressure from $t_{m} = 5 \text{ min}$ at higher speeds and in the continuum range up to measuring times of $t_{m} > 20 \text{ min}$ for lower speeds and mainly molecular flow conditions. Decisive for the determination of the permissible measuring time is a relative change of $K_0^*$ of maximal 3%.

As already mentioned, the screw-type machine in "blower" operation shows a behaviour which is strongly depending on the given flow conditions, which is clearly visible in Figure 4 and Figure 5 on the basis of the compression ratio $K_0^*$ as a function of the forevacuum pressure $p_v$ for all three machine variants examined and the variable speeds. Characteristic for the type of flow is the so-called Knudsen number $Kn$, which is defined as follows:

$$Kn = \frac{\bar{l}}{d}$$

Eq. 2.

In this context, $\bar{l}$ is the mean free path of the gas particles related to the lateral expansion of the gap cross-section $d$ (here: the gap height). All three flow ranges - continuum for $Kn < 0.01$, Knudsen for $0.01 < Kn < 0.5$ up to molecular for $Kn > 0.5$ - are characterized by the transition pressures in Figure 4 and Figure 5.

\[\text{Figure 4: Compression ratio } K_0^* \text{ as a function of the forevacuum pressure } p_v \text{ and the rotational speed } n (v_r = 1.1 \text{ und } v_r = 1.47)\]

*In "cold state", the screw-type vacuum pump shows an almost identical gap situation as it did prior to commissioning.*
Here, the qualitative curve of $K_0^*$ is characterized by two typical areas and, as can be taken from literature (1), is similar to a bell-shaped function. In range (I), a continuous increase in the compression ratio for constant rotational speeds with decreasing pressure is to be observed for both machine variants. From a certain forevacuum pressure however, at which a maximum of $K_0^*$ can be measured, a decrease in the compression ratios achieved by further decrease in forevacuum pressure occurs in the range of low gas densities (II). Depending on the flow range, the rotational speed, similar to the forevacuum pressure, has variable effects on the machine behaviour with regard to $K_0^*$. At higher forevacuum pressures (I) and in particular within the flow range of the continuum and in the transition to the Knudsen flow range within the gap, a significant influence of the rotational speed on the compression ratio occurs. In contrast to this, there is no significant dependence of the $K_0$-value on the rotational speed noticeable in the range of low forevacuum pressures (II). Moreover, an increase in the rotational speed is associated with the shifting of the $K_0^*$-maxima towards higher forevacuum pressures. With regard to the internal volume ratio, a considerable increase in the metrologically determined compression ratios is to be observed, as expected, with increasing duration of the internal compression – thus increasing $v_1$ – both in Figure 4 and Figure 5. This tendency turns out to be qualitatively relatively independent of the given type of flow. Furthermore, a shifting of the $K_0^*$-maxima first towards lower and then back towards higher forevacuum pressures with increasing $v_1$ is to be noticed with regard to the $v_1$-variation.

Figure 5: Compression ratio $K_0^*$ as a function of the forevacuum pressure $p_v$ and the rotational speed $n$ ($v_1 = 1.47$ und $v_1 = 2.0$)

The knowledge about the physical-technical mechanisms in the corresponding pressure and flow range, respectively, provides the explanation of the dependencies determined so far. For the transition and continuum flow within the gap, the gap mass flows play a decisive role as a loss mechanism, with their influence on the operating behaviour of the machine becoming less with decreasing gas density and diminishing conductance of the gap. The speed-dependent $K_0^*$-increase is decisively determined by the shortened working cycle and the resulting relative decrease in the gap mass flows in relation to the conveyed mass. This also explains the shift of the maximum values for $K_0^*$ towards higher vacuum pressures, since as a result, the negative influence of the gap flow on the operating behaviour of the screw-type vacuum pump is reduced. In the pressure range of molecular flow conditions within the gap, mainly other loss mechanisms described in literature are responsible for
the drop of the compression ratio, such as sorption/desorption effects, leakages, moving gap boundaries as well as the drag effect of moving components (1).

The direct comparison of the $K_{0\text{-}}$-maxima for $v_1 = 1.1$ and $v_1 = 1.47$ in Figure 4 shows an offset of the $K_{0\text{-}}$-peaks towards higher forevacuum pressures for an internal volume ratio of $v_1 = 1.1$. This suggests that for $v_1 = 1.1$, the loss mechanisms characteristic for the molecular flow range become dominant already at higher pressures compared to the machine variant with $v_1 = 1.47$. In Figure 5, the maximum of the $K_{0\text{-}}$-values for $v_1 = 2.0$ is to be found again at higher forevacuum pressures, against all expectations and against the trend of Figure 4. An explanation for this can be the higher pressure in the working chamber during the working cycle, which, in connection with the larger contact surface - in contrast to the variant with $v_1 = 1.47$ - of the contained gas molecules with the machine components, promotes the loss mechanisms of the sorption/desorption or other kinds of gas return. Moreover, in case of the machine variant with $v_1 = 2.0$, the fluid gets into contact with the compression flange at a later time; the forevacuum pressure $p_v$ in this flange is significantly higher than that in the chamber prior to passing the control edge of the outlet. This leads to a smaller pressure difference to the preceding chamber over a larger phase angle and in a reduction of the loss effect of the clearance mass flows (see also Chapter 4.3, Figure 4b).

### 3.4 Volumetric efficiency $\lambda_L$

In the following, the analysis of the pumping behaviour of the screw-type vacuum pump will be carried out based on the volumetric efficiency $\lambda_L$ of the machine. The volumetric efficiency is defined as the effective pumping speed $S$ related to the theoretical pumping speed $S_{th}$ of the pump:

$$\lambda_L = \frac{S}{S_{th}}$$

Eq. 3.

The volumetric efficiency integrally characterizes the pumping behaviour of the pump and takes the value one in case of an "ideally sealed" machine. Since this parameter is less important for the screw-type machine in "blower"-operation than the compression ratio at zero delivery, at this point only general coherences will be discussed in more detail.

First the volumetric efficiency of the experimentally examined screw-type vacuum pump is graphically represented in Figure 6 for the basic variant with $v_1 = 1.47$ as a function of the operating parameters of rotational speed $n$ and forevacuum pressure $p_v$ as well as of the inlet pressure $p_E$. For higher forevacuum pressures, this decrease shows a higher gradient and initially significantly lower values for small pressure differences applied to the machine. Moreover, a pronounced dependence of the volumetric efficiency on the rotational speed $n$ is to be noticed only for a forevacuum pressure of $p_v = 100$ mbar. This decreases with lower forevacuum pressures and can almost be neglected for $p_v = 1$ mbar, in particular at an inlet pressure of the machine of $p_E > 0.1$ mbar.

The relatively low volumetric efficiency of $\lambda_L = 0.85\ldots0.9$ for a pressure of $p_e = p_v = 100$ mbar at this point can be attributed to the high gap mass flows as a result of over-compression. Here, the leakage flows in the gaps decrease under the influence of shorter working cycles with increasing rotational speed, which results in the increase of the volume flow effectively delivered through the screw-vacuum pump.

367
The volumetric efficiency of the three machine variants of different internal volume ratios is illustrated in Figure 7 for a rotational speed of \( n = 6000 \text{ min}^{-1} \) and a forevacuum pressure of \( p_v = 10 \text{ mbar} \). While for maximal inlet pressures \( \lambda_k \) depends only slightly on the internal volume ratio, an increase in \( v_i \) results in a direct increase of the delivered gas amount at lower inlet pressures \( p_e \). Here again, similar to the compression ratio, the differences in the progression of the volumetric efficiency for the \( v_i \)-variation, can be explained by the influence of the decreasing
gap flow with a shift of the control edges at the pressure side towards greater phase angles. Consequently, the pressure level in the closed working chamber and the pressure difference, respectively the gap mass flow, to the previous chamber remain relatively low, so that the gas amount pumped through the machine can effectively increase.

4 THEORETICAL INVESTIGATIONS

In addition to the experimental investigations, theoretical investigations based on a chamber model were carried out. The simulation calculations are being compared with the experimental results already discussed to enable the assessment of the modeling accuracy and the future utilization of the simulation for the interpretation and development of screw-type vacuum pumps.

4.1 KaSim

The simulation of rotary displacement machines on the basis of a chamber model represents a recognized method for their analysis and further development. The chamber model method is primarily based on the common characteristic of all displacement machines, on one or more cyclically changing working chambers. For the purpose of simplicity, it is assumed that spatial gradients in the intensive state variables are negligibly small and the fluid condition within a working chamber can thus be regarded as homogeneous. Based on this principle, the fluid condition is described by the extensive state variables of energy and mass and the related working chamber volume. A thermodynamic change of the fluid state can occur e.g. by means of volume change, heat convections or mass and energy flows. The calculation by the time-stepping method is made under consideration of the law of conservation of mass and energy. (3)

The simulation program KaSim represents an implementation of the chamber model method and combines this with the advantages of object oriented programming. A fundamental distinction is made between capacities and connections. Capacities represent a storage for different physical manifestations, whereas connections enable an exchange between the capacities and, in their state, they are at any time defined by the related capacities. The working chambers of the rotary displacement machine are an example for a finite fluid capacity with time-variant volume; the gaps are an example of a connection for the exchange of mass and enthalpy. (5)

The calculation of essential physical-technical mechanisms for vacuum pumps in general and in particular for blowers is carried out on the basis of physical models in KaSim, which have been developed within the framework of previous research projects. These are models for gap flows with moving boundaries in vacuum (4), as well as models for the calculation of a detrimental gas return via component surfaces and via sorption/desorption.

4.2 Modeling of the SVB 51.2

The simulation of a rotary displacement machine by means of KaSim requires the geometric abstraction of the real geometry and the transfer to a chamber model. In addition to the geometric modeling, physical boundary conditions are required within the framework of the simulation, which concern e.g. the heat transfer between working fluid and component.

4.2.1 Geometric modeling

The geometric part of the chamber model basically consists of scalar information with regard to the volume of the working chamber or the area of the gap through which the fluid flows, both depending on the angle of rotation. The geometric modeling takes place with the help of an automated method which extracts the working chambers and the gap junctions from the user-defined geometry of the
twin-shaft rotary displacement machine and transfers it to a format readable for Kasim (7). Figure 8a shows the abstract chamber model with all connections except heat transfers and front gaps, whereas Figure 8b shows the scalar values for the chamber volume and the inlet and outlet area in dependence on the abstract angle of rotation, the rotor phase. Being the decisive factors that influence the operating behaviour, the gap heights of the simulation are adapted to those measured in cold state. The profile meshing clearance height is not constant along the intermesh line, due to the rotors rolling off against each other. Based on the measured data of the rotor profile, an arithmetic mean is calculated for the respective gap section along the intermesh clearance line.

Figure 8a: Chamber model of the SVB 51.2 without front gaps

Figure 8b: Curve of inlet and outlet areas in comparison to the working chamber volume of the screw-type supercharger SVB 51.2, plotted against the rotor phase (3)

4.2.2 Physical modeling
For the model of a screw-type vacuum pump, physical modeling comprises the representation of the gap flow, of the gas transport over component surfaces and the heat transfer between fluid and components. The gap flow is depicted by an approach which combines measured data for the continuum range and simulation data on the basis of a free molecular flow for the molecular flow range with moving gap boundaries for different shapes of gap contours (6). The gas return is depicted with the help of a virtual dead space, whose fluid content is determined according to the Freundlich isotherm (6). The last boundary conditions concern the heat transfer. Due to the high compression ratios at zero delivery, extremely high temperatures are to be expected for an adiabatic model, which could not be confirmed in experimental investigations. However, since Nussell-equations are not known for the fine and high vacuum, the simulation assumes a complete heat transfer and thus an isothermal change of state in the working chamber.

4.3 Simulation of the maximum compression ratio
Analogously to the experimental investigations, the simulation of the maximum compression ratio is carried out by varying the forevacuum pressure pv and the rotational speed n of the male rotor. Then, in the second step, the internal volume ratio vi is varied.
Figure 9 shows the comparison of the measured and the calculated maximum compression ratio $K_0^*$ for different forevacuum pressures $p_v$ at a constant rotational speed of $n = 6000 \text{ min}^{-1}$. It is to be noticed that the qualitative trends in dependence on the forevacuum are accurately reflected. On the one hand, this concerns the increase in the $K_0^*$-value with decreasing forevacuum pressure up to a maximum value. In this characteristic point, the absolute compression ratio in the simulation is higher than the measured value, however the forevacuum pressure related to the maximum $K_0^*$-value is well calculated. Now with decreasing forevacuum pressure $p_v$, the compression ratio $K_0^*$ decreases from the maximum $K_0^*$-value due to the increasing gas return through the virtual dead space. Since the deviation in absolute figures increases with decreasing forevacuum pressure $p_v$ but, in relative terms, is always constant, it can be assumed that the models for the depiction of gap flows have principally been the right choice.

![Figure 9: Calculated and measured maximum compression ratio $K_0^*$ as a function of the forevacuum pressure $p_v$ with varying rotational speed $n$ of the male rotor](image)

Figure 9 shows a reflection of the calculated $K_0^*$-values with varying rotational speed. Again, the qualitative trends are well illustrated, but even with varying rotational speeds the absolute values are higher by an almost constant factor, independent of the examined forevacuum pressure. This very systematic deviation becomes clear in the pure examination of the gradients. Figure 10a shows the relative change of the $K_0^*$-values as a function of rotational speed in comparison to experimentally determined data. Here, good conformity with the experimental results is to be noticed. A possible cause for the discrepancy between relative and absolute modeling accuracy could thus be the geometrical modeling of the clearance areas, since based on the qualitative conformity of the $K_0^*$-values, insufficient physical modeling can presumably be excluded.

For this reason, the integral clearance volume flow $S$ was examined in different determined rotor positions. An exemplary comparison of experimental and calculated results is shown in Figure 10b. Here, the clearance volume flow $S$ for two different inlet pressures $p_e$ with varied outlet pressure $p_a$ was measured and calculated. A generally good concordance is to be noticed, which shows a deviation of only a few percentage points at the inlet pressure of $p_e = 100 \text{ mbar}$. A reduction of the inlet pressure $p_e$ results in a decrease in the gap volume flow, with the calculated volume flows being generally lower than those experimentally.
determined. This deviation can be explained by the physical modeling of the clearance flows, however obviously neither the geometrical nor the physical modeling of the clearance flows are decisive for the deviations of the $K_0^*$-value in experiment and simulation (cf. Figure 9).

A variation of the internal volume ratio $v_i$ is realized in the simulation, analogously to the experimental investigation, by a modification of the outlet surfaces. A comparison for three selected internal volume ratios $v_i$ is depicted in Figure 11a.

Again, the qualitative influence of the volume ratio is correctly reflected, however, regarding the absolute $K_0^*$-values and the internal volume ratio $v_i = 2.0$, a deviation from the previous pattern is to be noticed, since in this case, the relative deviation decreases. Nevertheless, the simulation can help deepen the comprehension for the influence of the internal volume flow on the thermodynamic process. For an exemplary operating point, the working chamber pressures $p$ as a function of the working chamber volume $V$ are shown for three different internal volume ratios (Figure 11b). Although even the highest volume ratio of $v_i = 2.0$ does not result in adjusted operation, significant work-saving can be achieved. This work-saving results from two essential factors, which also contribute to the increase in the maximal compression ratio. The almost sudden pressure rise caused by
undercompression is retarded by the time extension of the compression phase and thus the clearance flows are reduced by flatter compression curves. As a decisive consequence of this, the indicated suction volume increases and a lower ultimate pressure can be achieved.

4.4 Simulation of suction speed

The simulation of the pumping speed is carried out analogously to the experimental investigations. Figure 12 shows the simulated volumetric efficiency $\lambda_v$ as a function of the inlet pressure $p_e$ for a determined forevacuum pressure $p_v$ and different rotational speeds in comparison to the experimental data.

![Figure 12: Calculated and measured volumetric efficiency as a function of the inlet pressure $p_e$ for different rotational speeds $n$.](image)

Starting at the point of the lowest inlet pressure $p_e$, a deviation between experiment and simulation is to be noticed for all rotational speeds, which is synonymous with deviations of the $K_0$-values already discussed. With increasing inlet pressure $p_e$, the calculated volumetric efficiency increases as expected. Here, it can be observed that, before the inlet pressure $p_e$ reaches the forevacuum pressure $p_v$, the calculated volumetric efficiencies are always higher than those experimentally determined. When the forevacuum pressure $p_v$ is reached, the calculated volumetric efficiencies approximate the theoretical maximum value, which is not reached due to the internal compression.

5 SUMMARY

Within the framework of this article, the successful transfer of a machine concept from the field of screw-type superchargers to the field of vacuum technology has been described. In comparison to industrially applied Roots blowers, a first test machine has obtained good results regarding the achievable ultimate pressures and volumetric efficiencies. A great advantage compared to the Roots blowers used so far is the reduction of the machine size at a concurrently increased maximal compression ratio $K_0$ per theoretical suction speed.

The investigation of the geometric parameter of internal volume ratio shows that a gradual diminution of the outlet surfaces and the related increase in the internal volume ratio has a positive effect on the achievable compression ratio. Up to a
volume ratio of 2.0, the increase is even progressive, which indicates further potential. An investigation of further internal volume ratios is desirable in order to evaluate, in particular, the possible influence of an increasing throttling and the related excess pressure during the expulsion.

A comparative analysis of experiment and simulation shows that the simulation software KaSim is able to calculate essential characteristics of the machine in a qualitatively correct way so that in future, a further optimization of the geometrical parameters can be done purely theoretically. According to current knowledge, the deviations shown between experiment and simulation are not completely explainable, since in particular for high forevacuum pressures, geometric and physical modeling should be sufficiently precise, as the investigation of the integral gap volume flows indicate. Further experimental investigations are planned in order to clarify this discrepancy.

REFERENCES