Verification of a Program System for the Simulation of the Thermodynamic State Variables of a Gas in Positive Displacement Vacuum Pumps

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Abstract

The increasing demand for dry running vacuum pumps requires the further development of these machines. In order to improve displacement machines computer-aided simulation has emerged as a suitable tool. The depiction of vacuum pumps in the simulation requires a knowledge of clearance mass flows. In order to determine these mass flow rates for any desired gaps in vacuum pumps an extensive measurement data base has been created, which makes it possible to determine mass flow by interpolation within this base. The calculation module which results from this, is installed in the simulation program, the so-called chamber model, based on the conservation of mass and energy. In order to validate the program system the screw-type vacuum pump serves as an example. This machine's gaps are depicted as regular shapes, which can be calculated by means of the flow rate module. The simulated pressure curves which result are compared to experimentally determined pressure curves, measured on the real machine. Within the range of moderate suction pressures and rotor speeds a good degree of concordance can be claimed. Increasing the rotor speed or decreasing the suction pressure leads to conspicuous deviations and to the conclusion that the effect of vanishing vis inertia, which has not been taken into account in the simulation, is greater than in the high pressure range.

1 Introduction

The deployment of vacuum pumps in the manufacture of semiconductors, in metallurgy and in chemical engineering, but also in research and development, makes increasing demands both on the vacuum itself, and on the process for attaining it. In demanding applications purity of vacuum is required, which means that a least possible contamination of the conveyed gas has to be achieved /1/, /2/, /3/. This required purity can be equated with a relinquishment of auxiliary resources such as coolants and lubricants. It results in the demand for dry-running vacuum pumps, in which no vaporizable auxiliary fluids contact the conveyed gas.

Over the last few years this property of the machine has been gaining significance in the market. It is logical to supply the resulting need with dry-running positive displacement vacuum pumps, to be newly developed by the manufacturing side.

The relinquishment of direct cooling, sealing or lubricating fluids causes in the first instance unsatisfactory operation properties compared to wet-running vacuum pumps, which can either appear as an increase in the attainable suction pressure or as a decrease in the suction volume flow. More significant than losses in these operation parameters is a potentially inadmissible thermal load of the conveyed gas, which compromises of operational reliability, particularly with regard to the frequently-requested explosion protection on the one hand, and on the other hand the life-expectancy of the pump itself, because of the risk that the rotors may scratch the housing. A way must be found to guarantee the operational reliability of a thermally highly-loaded machine, while dispensing with coolants and lubricants.

Computer-aided simulation of thermodynamic process variables has proved capable of determining boundary conditions for the calculation of mechanical component deformation by the finite element method. The correct depiction of clearance flows is a basis for the close to reality computational simulation of vacuum pumps.

2 Chamber model

The simulation program for the thermodynamic depiction of a displacement pump is based on the so-called chamber model. During the whole working cycle on its way through the rotor each chamber is connected to pre- and post-working chambers via a number of clearances, which depend on the type of machine. The chamber model describes the network of chambers and connections which emerges in this way. The chamber model of a double-threaded screw-type vacuum pump is shown in **Fig. 1**.

Working from the suction side to the pressure side, the simulation program depicts the thermodynamic state of the gas inside a chamber along the angle of rotation one-dimensionally, in discrete time steps on the basis of the chamber model. The physical fundamentals of the model are the conservation of mass and energy. The process is subdivided into the three phases: suction, transport (incl. compression) and discharge. In order to depict the transfer of charge, simplified models are assumed. These are an ideal charge of the suction chamber at low pressure conditions and an isentropic charging of the discharge chamber.



Fig. 1: Chamber model of a double-threaded screw-type vacuum pump

3 Basis of measurement data

In order to depict the clearance mass flow rates during the transporting phase of the working cycle, flow rates on regular sections of gaps are measured. There are 21 varied shapes, e.g. differing in length and height of gap, as well as the intake and outtake pressure. The individual measurements of the extensive data pool which emerged in this way are decreased by continuous functions to a smaller amount of data. The shape parameter *I*, the height of gap (clearance) *s* and the intake and outtake pressure p_{in} and p_{out} are the four dimensions of the measurement data pool.

The application of the results as a module in a simulation program requires the continuous, closed depiction of the clearance flow behaviour for any desired parameter within the scope of the examined extreme values of the varied parameters. Concerning the pressure ratio $\gamma\gamma$ this demand is already achieved by the algebraic function of a mass flow measurement with constant intake pressure p_{un} and a continuously increasing output pressure p_{out} . Additional mass flow measurements support the interpolation of the other parameters. As the gas temperatures in the chamber model of the vacuum pump can differ significantly from the ambient temperature (which is the temperature of the measurement data pool), a physically reasonable assumption is needed to estimate the influence of this factor.

Influence of intake temperature

According to Sutherland /4/ a rising temperature is conterminous to the molecules' increasing mean free path \bar{l} . Consequently the flow-describing Knudsen number *Kn*, which represents the ratio between the mean free path and a characteristic clearance-describing value, also increases.

$$Kn = \bar{l} \cdot l_{char}^{-1} \qquad \qquad \Theta q. (7)$$

Assuming a similarity of the Knudsen number for vacuum flows at differing temperatures, a temperature variation as in the required application in the simulation program can be de_{p} picted by modulation of the intake pressure p_{in} and the temperature T_{in} to the required range of variables.

Interpolation of intake pressure

Inserting additional pressure sampling points into the measurement program, for instance single selected clearance contours, makes it possible to compare flow rates, which are measured here, with the interpolated flow rates of the adjacent pressure sampling points, which include the following flow types: molecular flow, Knudsen flow and viscous flow. Within the scope of the types of interpolation examined, the smallest average deviation in the measured value can be arrived at through a logarithmic interpolation.

Interpolation of clearance

Three additional clearances were set, using single clearance profiles. Their measurement results are compared with the interpolated results of the adjacent pressure sampling points

The logarithmic interpolation at constant intake pressure emerges as the most suitable procedure, producing the smallest average deviation.

Interpolation of shape parameter

The large number of examined clearance shapes allows a linear interpolation between the adjacent profile parameter sampling points to be made. It was decided not to produce additional profiles. **Fig. 2** illustrates an overview of the algorithm applied in the simulation program.



Fig. 2: The algorithm applied in the simulation program to determine the clearance flow rates for any desired gas intake temperature, pressure, pressure ratio, clearance and clearance profile parameter within the range of the examined parameters

4 Depiction of clearances in the simulation program

The clearances which actually occur in dry-running displacement vacuum pumps are not often present among the profiles examined in the experiments. With regard to these clearances, it is necessary to reduce their geometries to the investigated simple regular shape geometries, as has been demonstrated for the screw-type vacuum pump under examination. As illustrated in Fig. 1 the screw-type vacuum pump comprises four types of clearance which are typical for displacement machines: the housing gap, the profile meshing gap, the radial gap and the blow hole. These real clearances have to be depicted by means of the regular clearance profiles examined here.

The housing gap

The housing gap, **Fig. 3**, is formed by the rotor tooth head and the inner wall housing. Its height corresponds with the radius of the rotors in their housing. The width of this gap w_h arises by unwinding the screw line by an angle which results from a complete turn less the angle between the cusps and the center. Its length l_h (in direction of flow) is formed by turning the tooth head in the direction of the rotor pitch. This gap is depicted by regular gap, which owns the length l_h and width w_h in the simulation.



Fig. 3 Depiction of the housing gap in the chamber model of a screw-type vacuum pump

The radial gap

The radial gap is formed by the radial fetch between the outer diameter of the rotor and the inner diameter of the counter rotor. Its fetch correspond with the clearance height, the gear height, turned in the rotor pitch direction, with the clearances width. As the pitch direction of the inner circle is not exactly equal to the pitch direction of the outer circle a mean value is assumed. In the direction of flow of this gap, a nozzle-like gradient is formed by the inner and outer circle of the rotors. A double-sided curved gap like this can be depicted by a one-sided curved gap by bending the constructed mean flow thread of a double-curved gap and the clearance boundaries until one boundary becomes plane, as shown in **Fig. 4**. The resulting mean radius of the one-sided curved gap can be assumed to be the depiction of the pump's radial gap in the simulation program.



Depiction of the radial gap in the simulation program by bending the mean flow thread

Profile meshing gap

The profile meshing gap is formed by the axial fetch of the rotors' flanks. Its shape is formed by the leading flank of one rotor and the trailing flank of the other. As an analysis of the clearance is very costly, the minimum cross-section is regarded simply as the cross-section in the section where the axes are parallel, turned into the mean pitch direction of the rotors. The resulting geometry describes a shape which depends on the shape of the rotor flanks. In the case of a screw-type vacuum pump this geometry is reduced to a rectangle with the same area.

The simplified calculation of the cross-sections area and the course of the clearance is based on the assumption that the rotors have of symmetrical flanks, so that the shape of the leading flank is the same as that of the trailing one. These flanks are simplified by replacing them with plane surfaces. Based on these suppositions, the course of the gap along a mean flow thread, dashed line in Fig. 5, is calculable. The resulting course is convex and its mean radius is significantly larger than the largest radius of clearance profiles examined in the measurement data pool, so that this clearance can be depicted by two parallel surfaces, which are contained in the data pool.



Fig. 5 The profile meshing gap as a parallel, regular gap in the simulation program

The blow hole

The blow hole emerges in the cusp area when the rotors separate. It is bounded by the rotor flanks and the housing cusp. Enormously simplified, this geometry can be depicted by a trapezium, which is formed by the rotor flanks, the cusp and a fictitious boundary between the rolling circles. Because of the sharp-edged cusp, this gap is depicted in the simulation by a shape which resembles an orifice, and is part of the measurement data pool.



Fig. 6 The blow hole is depicted by a shape body, which resembles an orifice

The necessary information for the chamber model to calculate the interplay of mass flow rates and the thermodynamic state variables of the transported gas within the participating chambers at any desired point of operation, is a consequence of the depiction of these four clearance types in the simulation, using the measurement data pool and its interpolation algorithm. The results of these simulations are discussed in chapter 6 in connection with the pressure curves which result from the experiment.

5 Indicating pressure

The evaluation of the quality of the module for the clearance flow-rates simulation requires a comparison of the simulated pressure curves with experimentally measured pressure curves at different operating points, speed *n* and intake pressure p_{in} of the vacuum pump. The machine examined is a screw-type vacuum pump with a rather low pitch and consequently large wrap angle. The ascertainable degree of rotation angle of a pressure gauge is also a function of the pressure-sensitive area's size as a function of the chamber dimensions, and therefore a function of the rotor pitch. The limited height of the chamber decreases its proportion relative to the diameter of the sensor membrane, and that is how it restricts the indicated degree of rotation angle as shown by the pressure gauge. The combination of pressure gauges which is used here limits the ascertainable scope of rotation to a rotation angle of 64°. Due to the large wrap angle and the high number of pressure gauges needed to encompass the whole working cycle, it is limited to rotation angle 1304° (21 pressure gauges with 2° overlap). The sensor membranes should ideally end flush with the housing inside wall, to cause the least possible influence on the gas. For reasons of safety they are countersunk 1/10 mm.

The acquisition of measurement data, triggered axially above the intake, begins at the closure of the suction chamber. **Fig. 7** illustrates the positioning (a) of the pressure gauges and their correlation (b) to the angle of rotation.



Fig. 7 : Indicating pressure:

(a) Constructional layout and installation of the pressure gauges

(b) Disposition of the sensor measurement phases in the working cycle of the vacuum pump (membrane areas connected to the pre-working chamber, the height offset provider a better illustration)

Towards the end of the indicated scope of rotation angle four single closable pre-inlets are arranged, which serve to decrease the transporting phase to a scope covered by the measurement instrumentation (without the pre-inlets the transport phase requires rotation angle of

about 3000°). The pressure inside the chamber can consequently be raised to atmospheric pressure before the gas discharge actually begins. The arrangement of the pre-inlets in the simulation as well as in the real machine also helps to make the evaluation of simplifying assumptions about their flow characteristics (e.g. mass flow coefficient α =0.8) possible.

In order to guarantee the comparability of actual measurement results with the simulation results, the pressure has to be measured on the cool machine. The adjustment of the desired intake pressure takes less than one minute, so that changes in clearances don't have to be expected. The pressure data for a complete rotor turn are acquired. The scopes required are reassembled into a single pressure curve as shown in **Fig. 8**. This graph includes all points of operation are simulated.



Fig. 8 Acquisition of the 21 pressure signals and their assembling to a single pressure curve as a function of rotation angle φ.

6 Comparison of measurement and simulation

Fig. 9 to Fig. 12 illustrate first the vacuum pump without pre-inlets. The variation of intake pressure p_{in} and rotor speed *n* yields a concordance of measurement and simulation at high suction pressures and low rotor speeds, which is no longer resolvable within the scope of the measurement tolerances. This concordance disappears at decreasing intake pressures or higher rotor speeds. A possible reason for this seems to be the influence of the relative velocity of the gap boundaries, which has not yet been taken into account. Because it has the largest area, the housing gap seems to be the dominating clearance. The relative velocity of its boundary is orientated against the direction of gas flow, so that the existence of frictional influences, which retard the flow, can be assumed. This thesis is backed up by the fact that the gas density decreases at lower intake pressures and makes its flow more sensitive to frictional forces.



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The comparison of measurement and simulation of a vacuum pump which includes preinlets, **Fig. 13** to **Fig. 16**, delivers similar results in direction. A good concordance is also achieved here, which gets lost at higher speeds and lower pressures. In both cases, with and without pre-inlets, a tendency emerges for the real machine to be less leaky than the simulated one at these operating points. In the scope of the pre-inlets the emergence of dynamic effects cannot yet be depicted by the simulation. In addition to this a quantitative evaluation of the assumed behaviour of the flow is not possible without an additional measurement of the effective mass flow. Even though a necessary pressure profile of the whole working cycle is missing, a common tendency can be observed: pressure increases inside the chamber before the pre-inlets fully opens, but decreases before it totally closes. The reason for this effect is the reaction of the pump gaps. The gas mass which is lost through the pump gaps, is larger than the mass of gas which flows through the closing pre-inlet. The differing pressure quantities in the pre-inlets scope can be put down to the rather to large mass flow rates in the simulation, which cause a steep pressure rise in the post-working chamber. This decreases the pressure in the chamber, which is directly connected with the pre-inlet.



Fig. 17: Comparison of measured and simulated suction speed S for different rotor speed n

Fig. 17 compares the measured with the simulated pumping speed *S*. Within the scope of high pressures a good concordance is found. Even the reduction of the pumping speed at pressures near atmosphere is noted in both cases. At low pressure the simulated delivery rate λ differs conspicuously from the measured delivery rate, in spite of this the simulated machine achieving a better ultimate vacuum. In this case the gap flow rate cannot serve as an exclusive cause, since the real pump tends to be leaking especially at high rotor speeds. As a possible reason for this effect, a partial load of the suction chamber may be responsible. This effect has been observed in positive displacement machines in the high pressure range, but has not been examined in vacuum pumps so far.

7 References

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