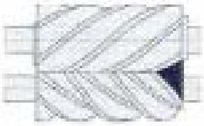

Research in rotary piston machines for compressible fluids

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Introduction

Rotary piston machines are a subgroup of the displacement machines, which are characterized by sealed rotating working chambers with volumes cyclically changing during a working cycle. This subgroup includes, for example, the roots blowers, scroll compressors, screw compressors or also the spindle vacuum pumps. For more than 100 years, rotary piston pumps have been used both in industry and in our immediate surroundings, e.g. in refrigerators mounted in motor vehicles. They represent the most commonly used type of compressor.

The idealized theoretical operating principle of the rotary piston machine is simple to describe and has been sufficiently known for some time. It consists of the phases "filling", "transporting" probably including a volume change of the closed working chamber as well as of "discharging". The theoretical pumping mass flow of the rotary displacement machine without exhaust chamber is determined by the product of the volume of a working chamber at the time of chamber closing, the number of working cycles per revolution, the fluid density at the suction side and the rotational speed. The minimum theoretical driving power results from the ring integral of the pressure-volume history of all working chambers, also called indicator or PV-diagrams, multiplied by the rotational speed. For the change of state of the fluid during the volume change in the closed working chamber, isentropics are frequently chosen, and one isobar each for "filling" and "discharging". Consequently, a doubling of the rotational speed idealized theoretically leads to a doubling of the pumping mass flow and the driving power.

In contrast to the ideal operating principle, the real operating behavior of the rotary displacement machines turns out to be complex and not adequately researched so far. Beginning with the unsteady "filling" of the working chambers, the flows through the design-dependent clearances at a gap geometry which, at the same time, due to thermal expansion depends on operating point and rotation angle up to multi-phase flows in the so-called wet-running machines or the sorption effects in vacuum pumps, today a number of basic issues are still unresolved. It is all the more surprising that the number of university research facilities continuously working in this field is very limited both in Germany and throughout the world.

The aim of the further report is to emphasize the significance and the necessity of research in rotary displacement pumps from the basics to the application. Due to the structure and application-dependent relevance of individual mechanisms of action, the current topics are discussed separately for three cases of application.

Rotary displacement machine for waste heat utilization

The conversion of waste-heat exergy into mechanical or electric energy will be the market of the future. To this regard, the screw-type engine as one type of the rotary displacement machine enjoys special prominence among the fluid energy machines [1–3]. In contrast to turbines or oscillating machines, a liquid fluid portion in a screw-type engine has a positive effect on the operating behavior due to the gap-sealing effect [4]. In particular because of the low temperature level in waste heat utilization, this characteristic is desired, since extensive immersion into the wet-steam region is possible. At the same time, a further reduction of gap losses can basically be achieved at a concurrent increase of the power density in connection with high operating speeds. These advantages need to be utilized in the design of screw-type engines for certain applications.

The engineer and scientist is thus faced with the by no means trivial

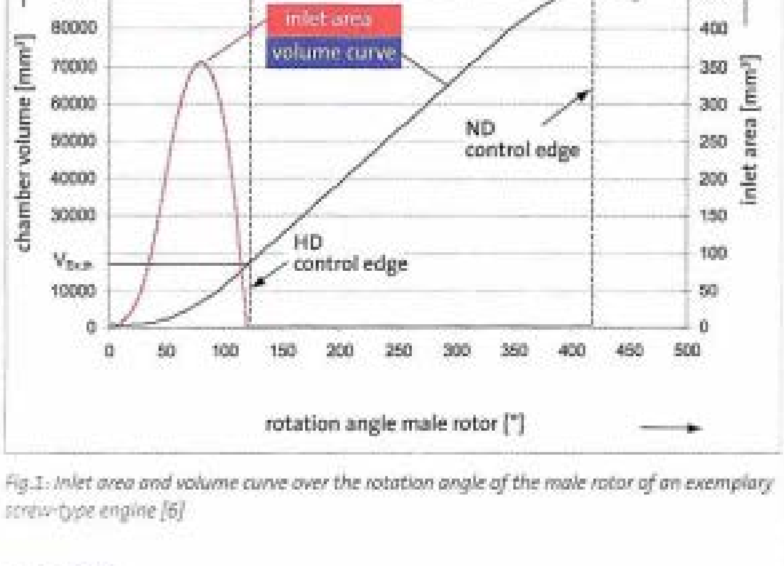


Fig.1: Inlet area and volume curve over the rotation angle of the male rotor of an exemplary screw-type engine [6]

question regarding the machine geometry, which, for thermodynamically determined limiting conditions of the surrounding plant enables the extensive conversion of waste heat exergy into mechanical or electric energy. This problem is pursued within the scope of a cooperative project, for which a screw-type engine for a water steam circuit was designed with the following conditions at the inlet-side

Mass flow: 360 kg/h
Inlet pressure: $7 \cdot 10^5$ Pa
Inlet temperature: 350 °C

Basic considerations on this have already shown that, in particular, the unsteady filling of the working chambers has a decisive effect on the inner quality of the engine [5]. For explanation purposes, Figure 1 shows a typical course of the chamber volume by way of the rotation angle of the male rotor [6]. At the same time, the position of the control edge at the high- and low pressure side (HP, LP) as well as the inlet area depending on the rotation angle are additionally marked.

It becomes clear that the chamber volume in the area of the rotation angle of the filling increases continuously. In particular in the range of a decreasing inlet area, the gradient of the chamber volume curve is high – in contrast to that of screw compressors – thus a complete filling of the chamber is hardly possible. Accordingly, the pressure in the working chamber falls below the predetermined inlet pressure already before the HP-control edge has been reached. The inevitable consequence is a loss of working area in the indicator diagram. This effect should be minimized by choosing the suitable wrap angle, the ratio of rotor length to rotor diameter as well as the position of the control edges.

Another influencing factor to be considered parallel to this is the already mentioned gap mass flow, which increases already during the filling of the chamber, from the chamber to be filled into the adjacent working chambers with low pressure levels. Apart from the pressure ratio at the respective gap, the gap mass flow is defined by the gap surface. The gap surface is calculated from the clearance length, which depends on time resp. rotation angle, and the clearance height, with the clearance height being a function of thermal expansion of the total machine (bearing seat, rotors and housing). Since due to the heat input, thermal expansion is impelled into the structure, which in turn depends on the change of state of the fluid in the machine and thus on the dissipation in the gaps – hence the gap flow – this influential factor can only be determined iteratively [7].

Within the framework of the research project mentioned, this coupled action mechanism is analyzed by means of a thermodynamic calculation of the screw engine in connection with the calculation of the thermal structural expansion. Here, the aim is to guarantee a clearance height in the range of 0.1–0.2 mm in the hot machine (inlet temperature 350 °C). Considering a change of the pumping mass flow by approx. 25 % at a change of clearance height of only 0.1 mm, the significance of the coupled calculation regarding the design of the screw-type engine is doubtlessly obvious.

Altogether, the theoretical considerations are aimed at the understanding of the basic physical interrelations and the development of a design tool for screw-type engines for given operating parameters, which does not exist so far. The examination of this tool urgently requires accompanying experiments. The engine designed within the scope of the project mentioned is on the verge of production. This project is a cooperation project between the research institution of the author and two renowned industrial enterprises.

The screw-type engine is expected to go into operation towards the middle of the year. First tests will then be carried out on a hot-air test bench. Here, the influence of the thermal expansion and thus of the gap flow depending on temperature as an operating parameter can be systematically analyzed. Afterwards, tests in a water steam circuit are planned. Based on a one-phase operation with superheated steam, the behavior of the screw-type engine in the wet-steam area with increasing liquid fraction can be researched. These experiments shall contribute to the development of a model concept and descriptive theory regarding the engine behavior in the two-phase area. Fundamental questions, for example regarding the time the steam requires for condensation in the hot rotating machine or the resulting droplet size and distribution within the working chambers and the gaps need to be answered. Moreover, the required measurement techniques, for example, for the determination of the steam content in the rotating chamber must be developed.

The consequent further development of this systematics leads to the two-phase engine with flash-evaporation (Fig. 2), in which directly overheated presswater is injected into the screw-type engine [8]. In this process, costly steam overheating is no longer required. In particular, the filling of the working chamber, unsteady in two respects, turns out to be a scientific challenge. The overheated presswater shall evaporate as spontaneously as possible at the moment injection occurs and, at the same time, shall completely fill the cyclically changing working chamber. Even the solution to this problem can only be found by means of basic experiments on the steam circuit in connection with the model concept and theories to be developed. This is an exciting challenge for many scientists and includes a great market potential, which may range from decentralized

Fig. 2: Oil-free, unsynchronized prototype of a two-phase screw-type engine for the examination of unsteady chamber filling in flash-evaporation [8].

utilization of waste heat up to power plant technology. However, it is still a long way to commercial realization.

Rotary displacement machines in vacuum applications

The types of positive displacement machines known from the application in the atmospheric pressure range are also used for the generation of vacuums [9]. Examples for this are the rotary vane pumps and the sliding valve pumps[2] as well as roots pumps and screw rotor vacuum pumps. Although the functional principle of these pumps does not change, the physical-technical mechanisms of action resulting in the characteristic curves important for vacuum applications are to a large extent fundamentally different compared to atmospheric applications. With the example of the roots vacuum pump industrially used for approx. 60 years, it will be explained why a merely theoretical forecast of the suction capacity (suction volume flow of the pump) as well as the possible compression ratio at zero delivery K_0 in the usual application range of this allegedly simple pump type has not been possible so far.

The maximum compression ratio of a roots vacuum pump is usually in the range of $K_{0,max}$ = 50 with a fore-vacuum pressure of approx. p_v = 100 Pa (1 mbar) (Fig. 3). This value is decisively determined by the gap flows and other mechanisms of gas return, for example, due to adsorption and desorption effects. The decrease of the compression ratio with increasing fore-vacuum pressure is caused by the increasing gap mass flows. However, the gas return beyond the gap flows gains increasing importance with decreasing fore-vacuum pressure and causes the decline of the characteristic curves, which can be observed here.

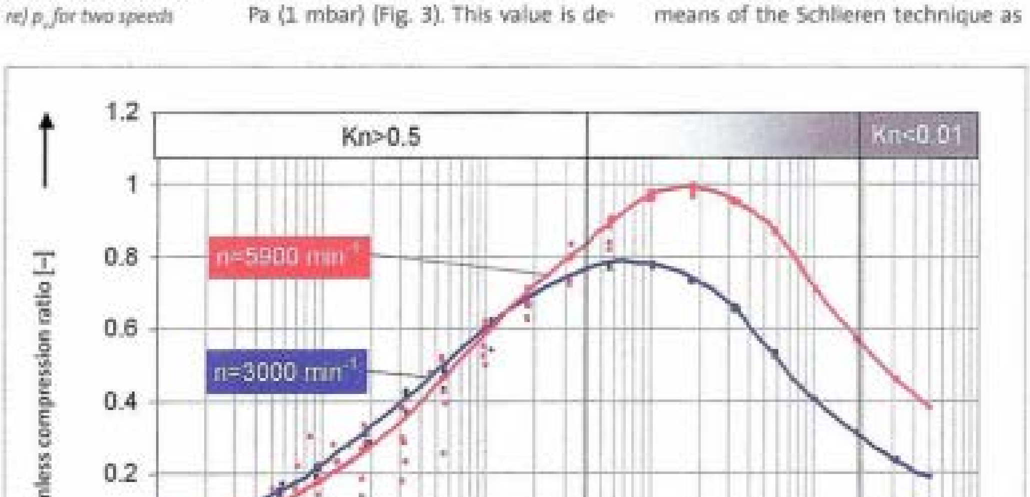
Already the calculation of the gap flow masses from rough vacuum up to fine vacuum (0.1–100 Pa) causes considerable problems. Responsible for this is the fundamental change of the type of flow in this pressure range, which is described by the ratio of the mean free path of the molecule to the characteristic length of the component (e.g. clearance height). The ratio is called the Knudsen-Number Kn and describes a continuum flow for values of Kn < 0.01, which applies to pressures above approx. 2000 Pa (20 mbar) in the gap, considering the usual clearance heights of approx. 0.1–0.2 mm. The calculation of the gap flow on the basis of the Navier-Stokes equation can provide reliable results only up to these pressures. However, already under the assumption of stationary gap flows the complex impact-boundary layer interactions analyzed in experiments by means of the Schlieren technique as well as the multiple shock reflection in the gap turn out to be quite demanding with regard to a numeric simulation [10, 11]. Moreover, a detailed calculation of the continuum flow in the moved gaps (housing gap, front clearance and profile meshing clearance) of a rotary roots vacuum pump could not be realized so far.

On the other hand, a molecular flow exists for the Knudsen number Kn > 0.5 (disjoining pressures below approx. 100 Pa), which is dominated by the collision of the gas particles with the gap bounds. Regarding the calculation, the Monte-Carlo-Simulation using the test particle method is suitable for this type of flow [12]. In satisfactory coincidence with the experiments, the results are gap mass flows of only a few percentage points of the value maximally possible in a continuum flow [13]. At the same time, more recent theoretical examinations show that, in particular in this range of the Knudsen-number, the relation time of the gap bounds has a major influence on the gap mass flow and thus on the achievable compression ratio at zero delivery K_0 as well as on the suction capacity of the vacuum pump [14].

Between these two kinds of flow, there is the transition range with Knudsen-numbers of 0.01 ≤ Kn ≤ 0.5. This range, also called Knudsen Flow is determined by both intermolecular impacts and collisions of gas particles with the gap bound. There are different models for the simulation of the Knudsen Flow, which consider the interaction of the gas particles with each other statistically or in more detail, for example on the basis of a power potential [12,15]. However, considering approx. 10^{13} particles per cubic centimeter already at a pressure of only 0.1 Pa, these models are computationally very intensive and have been limited to small volumes so far.

Thus considerable research, in particular in the vacuum application mentioned before, is required for a complete theoretical description of the gap flow in rotary displacement machines. Here, the application of an adapted model from the Knudsen Flow range is provided, initially for the numerical representation

Fig. 3: Measured dimensionless compression ratio K_0 of a roots vacuum pump at zero delivery over fore-vacuum pressure (pump pressure) p_v for two speeds



of a roots vacuum pump. In particular the combination of this approach with virtually rotating rotors, thus a realistic time-dependent geometry description seems to be promising. For example, first results indicate that the experimentally demonstrated decrease of the compression ratio K_0 at a determined fore-vacuum pressure above a certain speed in the range of molecular flows is caused by a progressive increase of the backflow through the profile meshing clearance. At the same time, this approach enables the integration and testing of different models for gas return e.g. due to adsorption and desorption effects or other loading and unloading mechanisms of the rotor and housing surface.

Parallel to the theoretical considerations, comprehensive experiments are required. For example, studies on gap flows with moved bounds are envisaged. Moreover, preliminary investigations for the measuring of heat flows introduced into the housing by means of Peltier elements are planned. In addition, the gas return mentioned should be researched beyond gap flows by means of basic pilot schemes, for example, by means of different gases at unchanged geometry.

Altogether, it can be said that research in rotary displacement machines in vacuum applications is still in its infancy compared to the state of knowledge regarding continuous flow machines for continuum flows.

Rotary displacement machines in compressed-air application

More than half of all running compressors are screw-type machines, which are to a great extent applied for compressed-air supply. The by far larger part of these machines (approx. 90 %) is „wet-running". For this purpose, oil is injected into the working chambers. This helps reduce the end temperature of the gas as well as the gap losses in the machine. At the same time, the male rotor can drive the auxiliary rotor directly – which means without the synchronization transmission. However, downstream of the machine the injected oil needs to be separated from the gas. In general, an undesired residual amount of oil remains in the gas depending on the quality of the separation [16].

What to a large extent is still unsettled in this regard is the theoretical correlation which indicates at which point with which direction and at which time the oil must be injected into the working chamber. Here, the aim is to minimize the amount of oil required with a maximum effect. Until now, manufacturers and scientists followed this question on the basis of comprehensive systematic experiments [17] and here, the practical experience achieved is accordingly high. However, a reliable model concept as well as a theory for the description of the droplet distribution or a liquid gush in a rotary machine is missing. A contribution to this scientific challenge could be the systematics regarding the survey of the machine behavior in the wet steam area already introduced in connection with the screw-type engine. Incidentally, further research work on this topic is still required.

Due to more stringent environmental regulations as well as increased demands on the process, manufacturers realize a rising demand for "dry-running" compressors. Accordingly, future developments are targeted at further avoidance of oil in the machines. The screw compressor of the future will run completely without oil at high peripheral speed (revs), either with water injection or completely dry – apart from an important residual humidity. Such a machine has already been developed and measured within the framework of various research projects at the university chair of the author (Fig. 4) [18,19]. It has no synchronization transmission. The male rotor directly drives the auxiliary rotor, and due to a novel rotor profile in connection with a special rotor coating this is, for the first time, possible in dry-running condition.

Also, with speeds of up to 30,000 l/min at a displaced volume of 286 cm³ per male rotor revolution, this supercharger, which has been developed for the charging of car engines, will break environmentally friendly and promising ground. However, the improvement of the operating life of the supercharger as well as the number of pieces required with regard to the application emphasize the continuous demand for cooperative research, in particular of material scientists, forming specialists as well as thermo- and fluid engineers.

Another sensitive topic of environmental and work protection is the avoidance of noise. Compressors cause unsteady flows as well as structural vibrations and thus emit media-borne, structure-borne and air-borne sounds [19–21]. Regarding the minimization of these negative concomitants, manufactures and research institutions are confronted with a number of unsolved questions. Today, primarily passive measures for the minimization of noise emissions are taken (e.g. mufflers, covers, noise protection housings), thus the demand for cost reduction as well as for energy efficiency calls more and more for alternatives. The starting point is the reduction of the stimulation of media-borne noise by increasing the uniformity of the volume conveyance. In case of screw compressors, an increase in the number of teeth or the ratio of rotor length to rotor diameter as well as, in particular, the selection of large wrap angles can be a solution. At the same time, the form of the time-dependent volume conveyance determines the spectral composition of the incitation of volume flow pulsations. Moreover, an adapted mode of operation helps minimizing the incitation of pressure pulsation.

Clearly more complicated is the answer to the question regarding the system response for the initiating media-borne sound. Apart from plane sound waves, acoustic transverse modes are often noticed in screw compressors. At the same time, the media-born sound as well as the synchronization transmission or the rolling motion of the rotors stimulate oscillations and mechanical resonances of the surrounding structure. Strong, often high-frequency fluid-structure interactions are the result, which are decisive for the noise emission to be minimized in the area of the screw-type engine. Due to these intertwined coherences, a forecast of the sound power emitted by screw compressors has not yet been possible with the necessary accuracy.

Conclusion

Rotary displacement machines have been used in industry as well as in our immediate surroundings for a number of decades. Although the basic physical-technical mechanisms of action have been known for long, the great success of this type of machine is to a large extent due to comprehensive, often empirically gained experience of the manufacturers. Accordingly, there is a number of interesting questions both for basic research and for application-oriented research. The goal of answering these questions would be furthered in particular by cooperative science, which unites the different approaches and disciplines as well as the experience of the manufacturers. From this pre-competitive research, manufacturers can derive targeted developments for their individual products from an ecological and economical point of view and thus fulfill the requirements of a continuously increasing demand for technological lead.

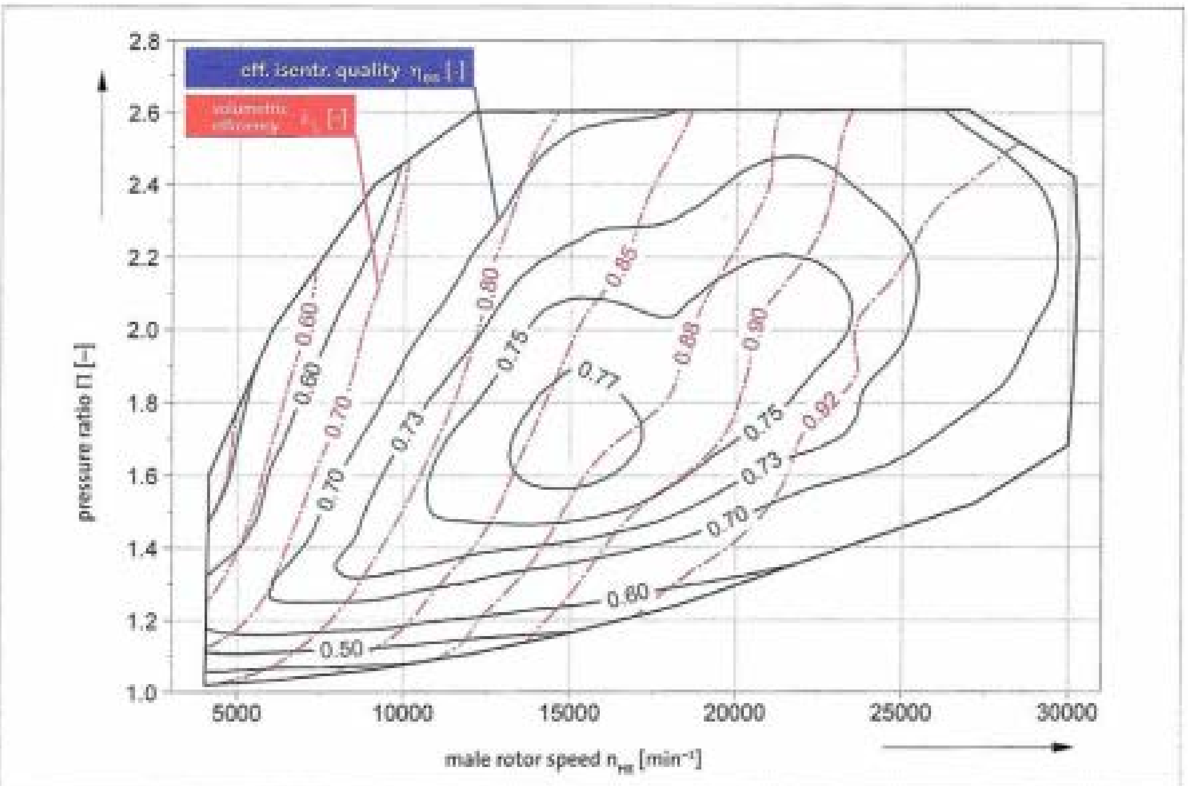


Fig. 4: Isolines of the real efficiency and the volumetric efficiency of an unsynchronized screw charger over the male rotor speed for different pressure conditions at a suction pressure of approx. 10^5 Pa (20 °C)

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[1] *Due to the compressible fluid as well as the usually high compression ratio, it should be talked about compressors – contrarily to the historical name „pump".*

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