Advanced analysis of twin screw compressors with variable rotor pitch using one-dimensional thermodynamic simulation

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Abstract

This paper presents methods for detailed analysis of the twin screw compressor's operation characteristics using a one-dimensional simulation approach. These methods are used for reasoning the advantages and disadvantages of screw rotors with variable pitch. The one-dimensional model includes next to the working chamber also several geometric and operation characteristics. These characteristics are combined to develop key figures to understand the effects of rotors with variable pitch on the operating characteristics and their influence. The calculated results are discussed in detail and a final outlook about the challenge of the manufacturing process of rotors with variable pitch is explained.

1. One-dimensional thermodynamic simulation of screw compressors

A one-dimensional, thermodynamic simulation of the operating characteristics of screw compressors represents a proven and established tool for the calculation and design thereof. Even in times of extensive three dimensional simulations, a one-dimensional simulation still offers adequate accuracy for the evaluation of the performance and analysis of the compression process. Furthermore, the one-dimensional simulation offers the big advantage of being relatively clear when being analysed. Parameters that evaluate the compression process as a whole and parameters that are formed for each period of time used for this.

For screw compressors, you call a thermodynamic simulation as one-dimensional, when the fluid in the work chambers is expected to be homogeneous within the respective chamber. Thus, the fluid state can be described by one-dimensional variables, which stand for the average condition within the respective chamber.

Initial applications of the mentioned approach for screw compressors were executed in a simplified form in the 80s and constantly developed further in the following decades [1, 2, 3, 4, 5, 6]. This paper uses the approach described in [6].

In contrast to a few other displacement machines, like piston compressors, the required clearances between the rotating components do not exclusively affect the clearance flows leaving the working chamber. By screw compressors, most clearances represent internal connections between working chambers, which are found in the different phases of the working cycle. Thus, not only the effect exists that the fluid escapes the working chambers, but rather also that the fluid flows from compressing and discharging chambers into suctioning and compressing chambers of a lower pressure.

These connections between the chambers themselves and to the suction and pressure sides represent the heart of the one-dimensional thermodynamic simulation of screw compressors. There the fluid mass and energy is exchanged between the working chambers and run through the work cycle step by step until a convergence criteria is met with the help of an iterative calculation process. This simulation model normally includes effects in regards to the flow characteristics in the clearances and in the inlet and outlet channels, heat exchange between components and fluid, as well as different power losses.

In terms of clarity, by the analysis of the thermodynamic operating characteristics, this paper is limited to the fluid flows between working chambers, which are determined through internal clearances between the components. In order to be able to ignore the remaining variables of influence, rotor geometry variants are defined in a way that these effects can be viewed as constant in the first approximation. All variants have the same theoretical volume flow at the same rotor speed.

2. Twin screw compressors with variable pitch

The motivation of using rotor profile with variable pitch is based on the desire to develop energetically more effective profiles. The rotor profile from screw compressors are designed in a way, that a profile cross section is extruded along the rotor axis and twisted. The wrap angle indicates the twisting of the profile from one side to the other. The technical more relevant variable for the production is the pitch of the profile. It specifies how far from a front side a movement in an axial direction must be taken, in order to receive an imaginary complete rotation of the profile cross section.

Since the rotor profiles of screw compressors are normally produced by grinding, the pitch describes the ratio between the rotor rotation and axial path of the grinding tool. The pitch, however, depends on the size of the rotors. So the size-independent wrap angles are frequently used to describe the profile shape. In order to define a profile with a variable pitch, there is the question which of these two geometry parameters should be used.

The "variable pitch" term describes a pitch, which does not run constantly along the rotor axis. In the following, the pitch should be described as the path along this. The rotor profile is divided in an axial direction in infinitesimal thin discs. The local wrap angle describes the wrap angle for each of these discs that a rotor profile would have if the entire rotor profile would have this pitch. In order to keep this procedure simple, a linear, non-constant path is aimed for. The profile cross section (SRM 4+6 profile with 90mm centre distance), profile length (L/D-ratio 1.65) and effective wrap angle (300°) remain unchanged.

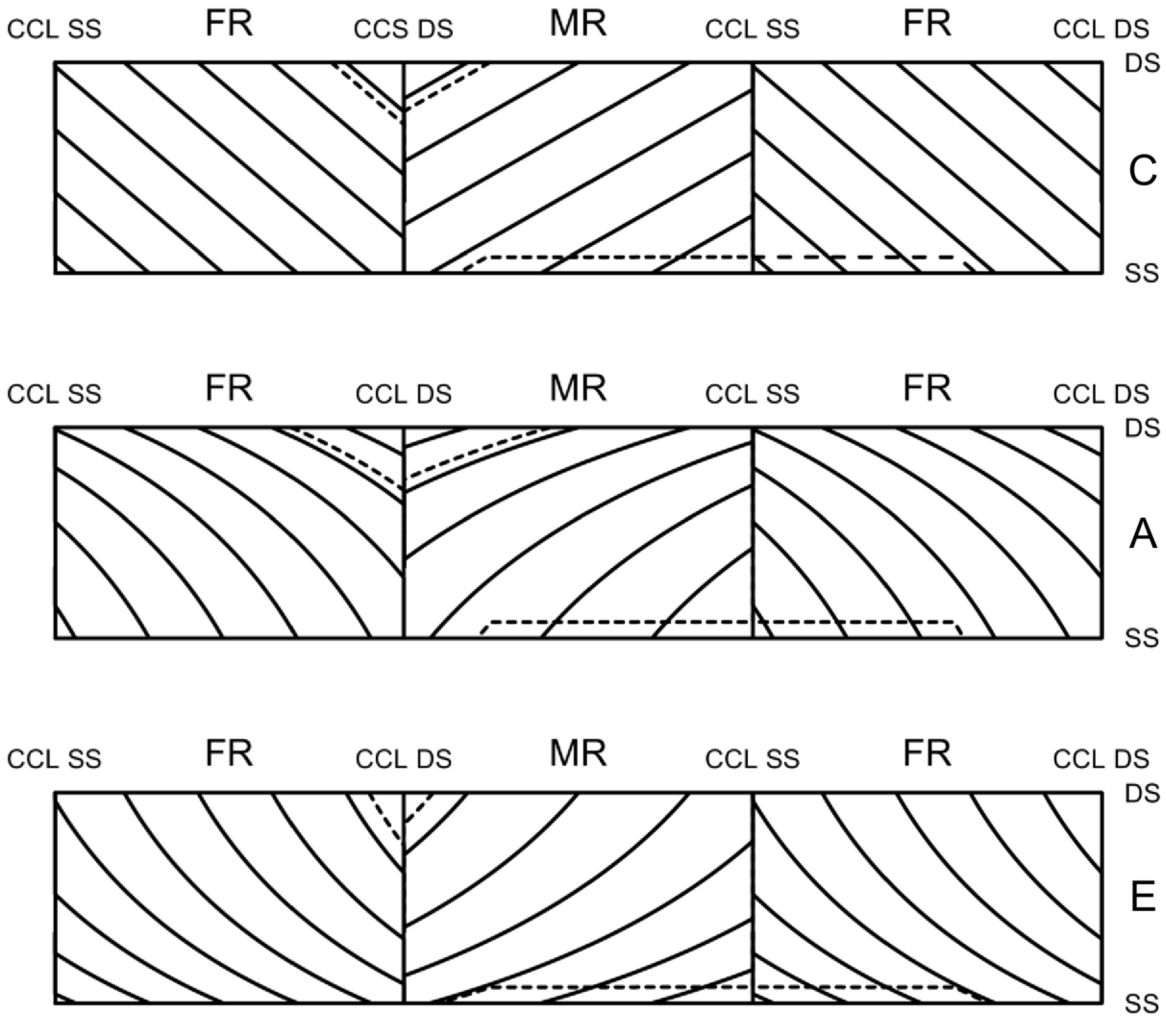

Fig. 1: Flat projection of rotor-profiles with different kinds of variable pitch

Variants A, C, E MR: male rotor FR: female rotor

DS: discharge side SS: suction side CCL: case cut line

Due to the basic requirements of a gear tooth system for screw compressors, there is a non-linear path by the female rotor with a linear path of the local wrap angle. Since when using the pitch as a descriptive parameter there is a linear path both by the male and female rotor, this will be used in the following.

In order to analyse the thermodynamic operating characteristics of such specific rotor profiles, five geometric variants are examined. One variant with constant pitch (c) and two variants each with different strongly decreasing (A, B) and increasing (D, E) pitch along the rotor length from the suction to pressure side (Figure 1). As already described, the remaining rotor dimensions are identical and are based on series machines. In this way all variants have the same theoretical volume flow per revolution.

3. Geometric characteristics

In order to be able to examine the influence of this geometric variation on the operating characteristics, the geometry must first be described with the help of geometric parameters. In the first step, the volume curve is used. It describes the chamber volume through the working cycle. Two major effects can be observed (figure 2).

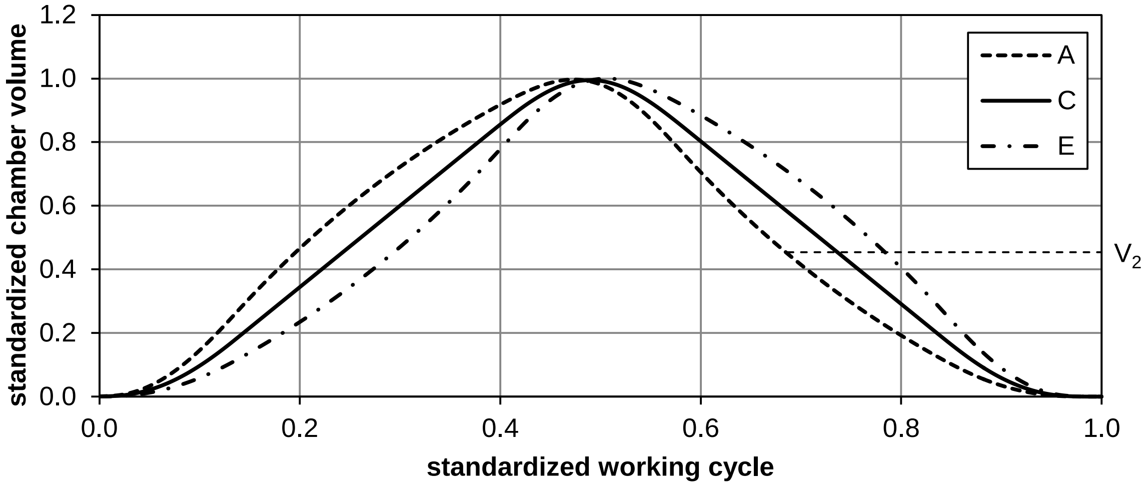

Fig. 2: Volume curves of rotor-profiles with different kinds of variable pitch (variant A, C, E)

V_2: chamber volume at begin of discharging

Chamber volume standardized to maximum of all variants

On the one hand, the variable pitch moves the position of the maximum chamber volume slightly towards the suction (variant A) or towards the discharge (variant E), whereby the value for the maximum volume remains constant. However, the different path of the chamber volume during suction and during the compression and discharging is significantly more different. Thus, the compression does not only work inconsistently in regards to time (figure 3), but also the opening angle of the pressure side changes dramatically for a defined volume ratio (figure 2 and 4). With variant A, the latter led to the fluid being discharged over a longer period of time and provides a larger outlet opening. This should lead to lower throttle effects during the discharging, because the average theoretical discharge speed is less (figure 5). Therefore, this results in lower required power consumption. The large theoretical discharge speeds at the beginning and end of the exiting are only of a theoretical nature. At the beginning, the fluid is compressed in the chamber a little bit longer, because the fluid cannot exit the chamber as intended. At the end, a majority of the fluid escapes through the clearances, because the ratio between the clearance area of the profile clearance and the outlet area increases while the remaining chamber volume is already very small.

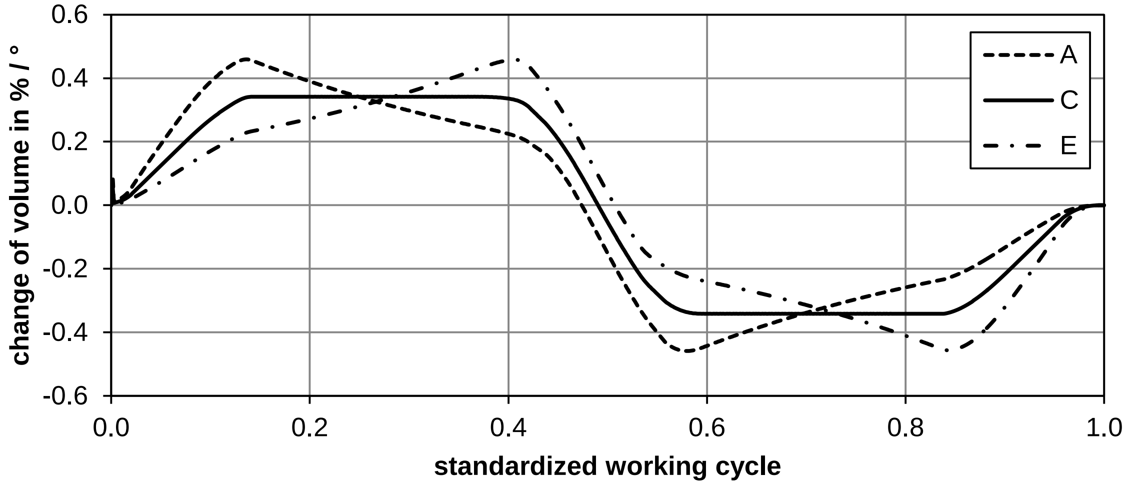

Fig. 3: Change of chamber volume per degrees of rotor rotation

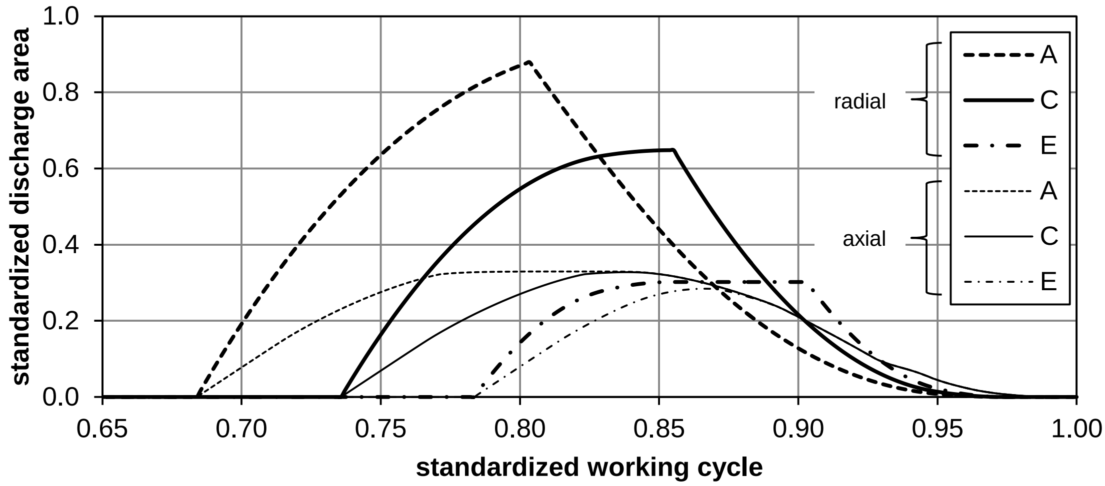

Fig. 4: Axial and radial discharge area of rotor-profiles with different kinds of variable pitch

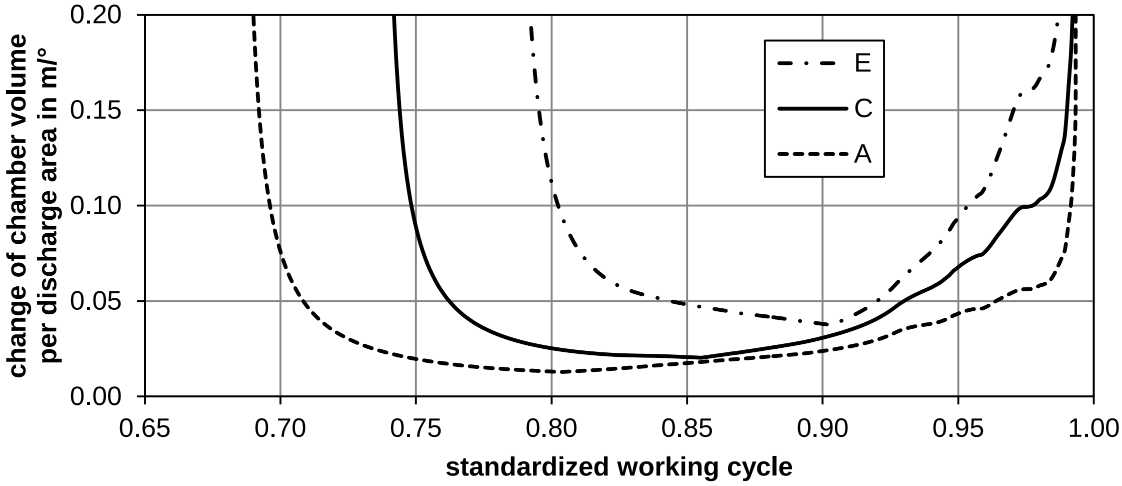

Fig. 5: Theoretical discharge velocity as change of chamber volume per discharge area

With the observation of the clearance area paths, it stands out that these are also influenced by the variable pitch. The trailing clearance area from the sum of the casing and front clearances as well as the blowhole has a slightly lower clearance area during the compression and discharging with variant A (figure 6). This is due to the significantly lower pitch on the pressure side, which influences the length of the casing clearance. For the clearance heights, typical values from series machines were used.
The path of the profile clearance area shows that the clearance area is less during the compression with variant E, in contrast when discharging by variant A (figure 7).

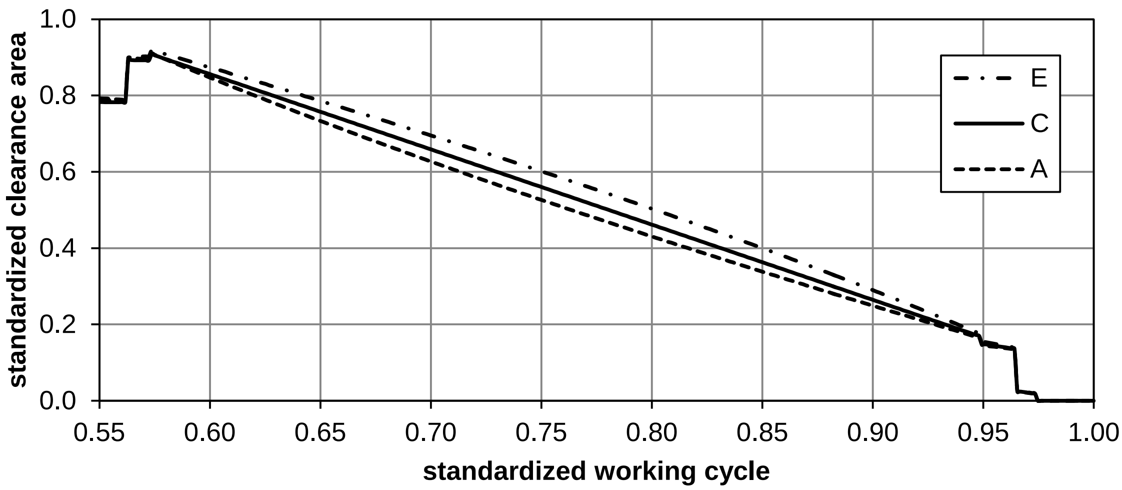

Fig. 6: Summed trailing clearance area of blowhole, casing- and front-clearances

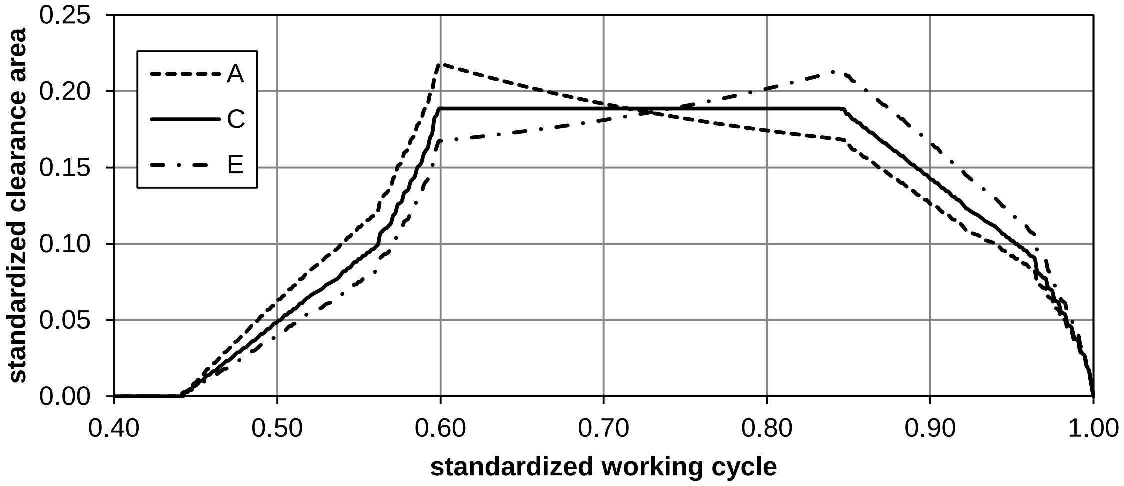

Fig. 7: Summed trailing clearance area of profile-interaction-clearances

Based on this geometric analysis, the variant A should have better operating characteristics than the other variants. Fewer throttle effects are to be expected on the outlet. And when discharging, the casing and profile clearance areas are smaller and during compression, the casing clearance area are smaller. Only the profile clearance area is slightly larger during compression.

4. Simulated operating characteristics

The one-dimension thermodynamic simulation of the described geometry variations serves for the purpose of deepening the analysis of the operating characteristics. For example, in the following the simulation, results of a dry running screw compressor are presented by an average speed common for this machine size as well as discharge pressure.

The compression through the working cycle is faster with variant A than with the other variants, as the volume curve already showed (figure 8). It stands out that the over-compression at the beginning of the discharging is comparatively small with variant A and C, while the described throttle effect can be clearly recognized with variant E. When observing the pressure curve in the indicator diagram, it stands out that variant A is more optimal during the discharging, but not as optimal at the beginning of the suction and during compression (figure 9). This characteristic at the beginning of the suction is reasoned by the faster increase in volume (figure 3). The slightly larger increase in pressure during compression is surprising at first, because this is due to a larger clearance fluid flow, but the casing clearance is less with variant A (figure 6). With an exact observation of the pressure flow over the working cycle (figure 8) it is clear that the faster compression by variant A causes a higher pressure over a longer period of time. This leads to a higher pressure difference on the clearances and to a higher clearance flow. This reverses the effect of the slightly smaller casing clearance area and worsens the operating characteristics.

This shows that a purely geometric analysis can be inadequate for the analysis of the operating characteristics. The thermodynamic simulation, and even if it is just one-dimensional, offers a significantly more detailed analysis, because of pressure curves and mass flows.

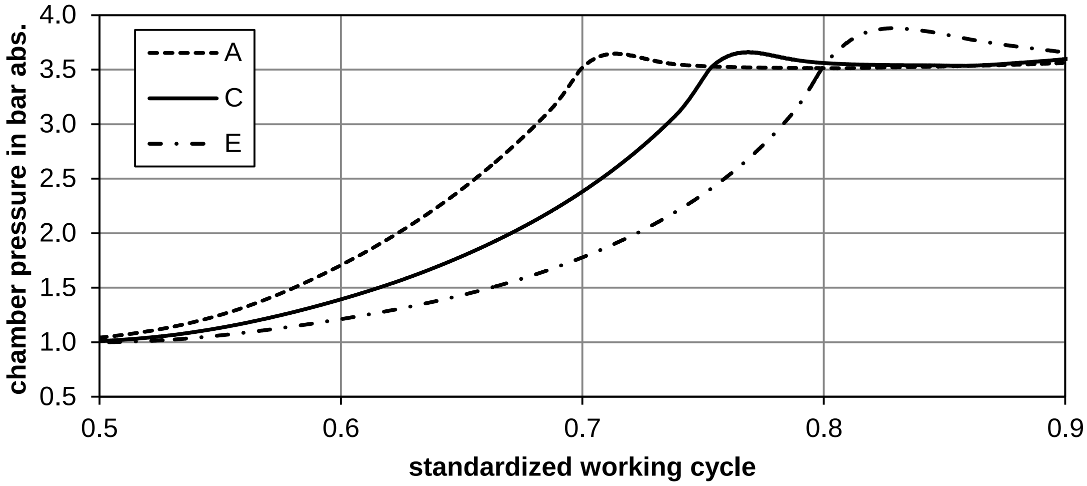

Fig. 8: Chamber pressure as a function of the working cycle (proportionate to time)

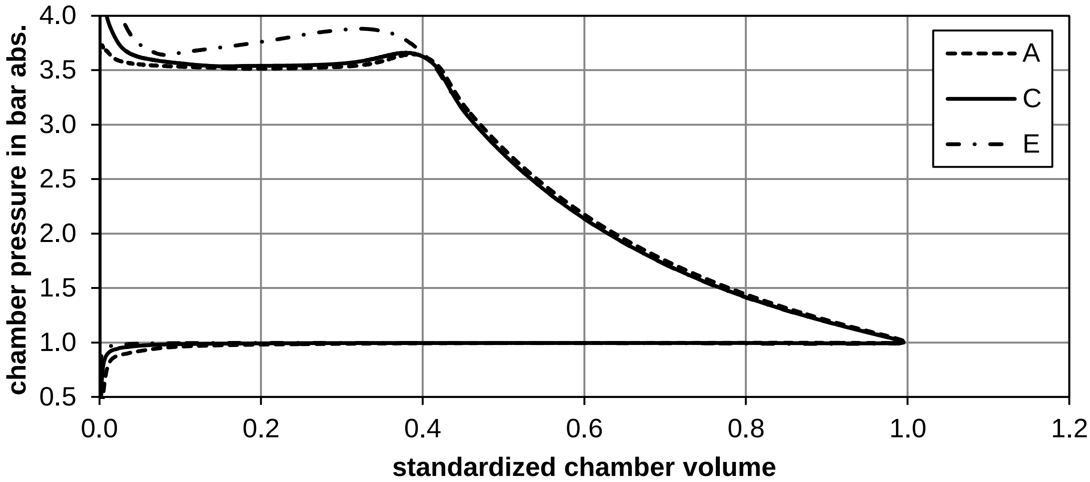

Fig. 9: Indicator diagram of rotor-profiles with different kinds of variable pitch

For the variants analysed here, it shows that the volumetric efficiency from variant A to variant E increases (figure 10). On the one hand, this is due to the mentioned effect at the beginning of the suction, and on the other side due to the higher clearance flows with variant

A. This causes more fluid mass to be transported in circuit in the machine and thus, less fluid mass is suctioned and discharged. The power consumption has a minimum for the base variant with the constant pitch. This is because the discharging is less optimal at the variants D and E and the intake and compression is less optimal with by the variants A and B. Altogether, the specific performance, for this case, has a minimum for the constant pitch.

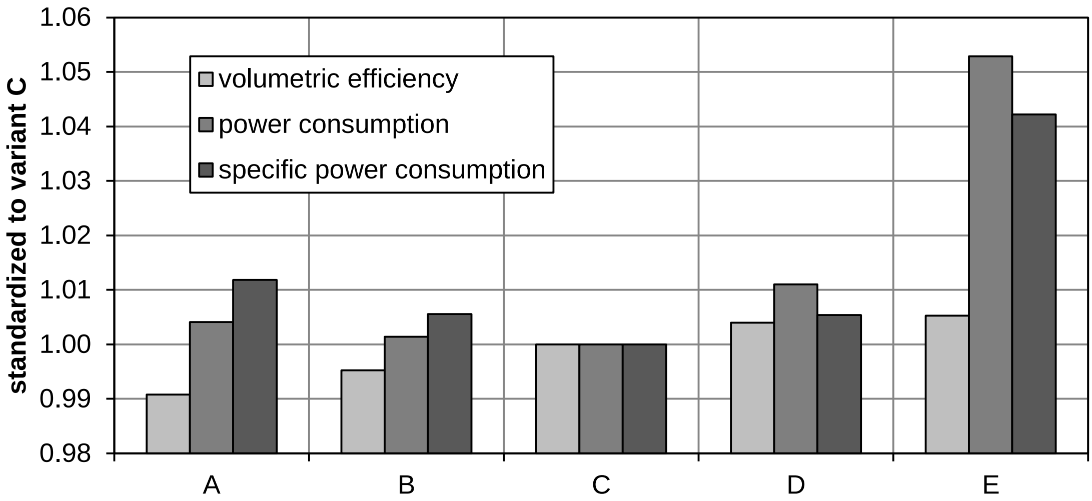

Fig. 10: Performance characteristics of rotor-profiles with different kinds of variable pitch
Variant B is the geometric mean of variant A and C. D is mean of C and E

The minimum of the specific power consumption is determined on the one hand by the volumetric efficiency (clearance flows) and on the other hand through the power consumption (pressure through the working cycle). Thus, it is not surprising that standard rotor geometry is already designed for this optimum and a radical change like a variable pitch does not result for any improvement to the operating characteristics, at least for this case.

However, it is to be assumed that a variable pitch may cause improvements if other geometry parameters deviate from the proven standards or the ratio of the clearance and throttle effect moves significantly. This may be the case with an auxiliary fluid injection or the compression of other fluids than air.

5. Challenges for manufacturing rotors with variable pitch

As already mentioned, the rotor profiles from screw machines are normally produced by grinding. The grinding wheel is set up diagonal to the rotor axis corresponding with the angle of pitch and move longitudinal along the rotor axis while the rotor slowly moves around its axis. This works, because the distance from a profile tooth to the next is constant in any level throughout the length of the rotor. With a rotor profile with variable pitch, this is only the case in the profile cross section level due to the basic requirements of a gear tooth system. In a longitudinal direction and in the cross-section of the profile pitch, this distance is not constant (figure 1). Consequentially, this type of machining cannot be realized with traditional grinding wheels.

A possibility would be to divide the profile part into a limited number of sections with a local, constant pitch and could be produced with a separate tool for every section. Another possibility would be to use free-form-procedures for production.

6. Conclusion

This paper shows that a geometric analysis of displacement machines allows for a lot of conclusions about the compression process, but may not be adequate for screw machines with their internal clearances, if the compression process and therefore the pressure progression are significantly different. Through the use of a variable pitch, the speed of the compression can be influenced. This may be desirable if a faster or slower compression is desired. Furthermore, the size of the outlet can be influenced. This can be useful if an application suffers from throttle effects on the outlet (solution variant A). This would also allow for compression ratios, which would not be able to be realized under normal conditions with screw machines (i.e. for screw expanders). For the case that the outlet is adequately large, the internal clearance flows can be reduced through an initially slower compression (solution variant E).

However, the production represents a challenge, because it would be more cost intensive and probably would not have the same production quality as through conventional machining. In the following, the first market ready application of a variable pitch is not to be expected by conventional compressed air technology, but rather in special application areas.

Definitions

standardized working cycle:	standardized to the male rotor rotation from the formation till the disappearing of the chamber volume
standardized chamber volume:	standardized to the maximum of the chamber volume of the variant with constant pitch
standardized clearance area:	standardized to the maximum of the trailing clearance area of the variant with constant pitch

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