Optimization of rotor profiles for energy efficiency by using chamber-based screw model

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Abstract

Based on the development of advanced manufacturing this paper highlights methods of analysis and improvement of rotor profiles to minimise energy costs. Therefor three different rotor profiles with the same number of lobes are analysed in view of actual requirements for lifetime and efficiency. The detailed analysis regards the possibilities of validations pertaining to geometric characteristics and influence of clearances for the profile selection for new screw compressor generations. This paper describes the modelling of screw compressor geometrics and the analysis with a one-dimensional simulation program. At the end the comparison of simulation results shows the influence of rotor profiles on overall efficiency and the development process for new rotor profile designs.

Papers about simulation of screw compressors often focus on one screw compressor and the quality of imaging. By using a chamber model generator and simulation tool this paper describes the methods of analysing rotor profiles for the development process of screw compressors. With the combination of geometrics and operating characteristics this paper goes one step further in reasoning to qualify the appropriate choice of rotor profiles.

1 Introduction

Due to growing energy costs and increasing climate and environment protection requirements, the development of screw compressors must be focused on improving the efficiency of the energy conversion process. The aim of optimizing the rotor profile using advanced production techniques is accompanied by the costs of the manufacturing process and the desire for a modular design. In the case of dry-running screw compressors, the largest influence on costs is attributed to the synchronizing gearbox and the rotors.

To optimize the construction of the compressor stage, the rotor profiles of screw compressors are considered during the development process to be crucial in terms of energy efficiency and possible modularity. The processes and stages of profile development and construction are therefore explained in detail, which gives an insight into the possible optimization potential.

2. Rotor profile development

There is a variety of different rotor profiles with different numbers and combinations of teeth, optimized for the relevant application or field of applications. The profile therefore characterizes the contours of the teeth and thus constitutes a characteristic independent of the number of teeth. On the other hand, the profile description also correlates with the number and combination of teeth. The underlying mechanical stresses on the screw compressor ultimately require a minimal tooth thickness, whereby sharp edges impede meshing in the pitch circle [2]. Existing rotor profiles thus have a limited field of application based on the specific application. This is reflected in the different profiles used in dry-running and oil-flooded screw compressors. The desire for a modular construction entails a time-consuming, repetitive development process. With it, development projects are sped up, and delivery times reduced with the help of a type of modular system. Based on the design, a distinction is made between two types of screw compressors: wet-running or fluid-injected (e.g. oil, water) and dry-running machines. Dry-running screw compressors are either single-stage or multi-stage depending on the volume flow to be transmitted through and the compression ratio, whereby the general construction differs depending on the type of screw compressor.

2.1 Profile comparison

The geometry of screw compressor rotors is normally described using the parameters profile, number of teeth, center distance, height and pitch. As the profile has a decisive influence on energy efficiency, this shall subsequently be examined in detail. The profile comparison takes into account three different rotor profiles with the same number of teeth on the male and female rotors as shown in Fig. 1 below. The different profiles include a profile taken from GHH (Fig. 1a), a reproduction of the N profile developed by Stosic of the City University of London (Fig. 1b) and an approximate Kaeser Sigma profile (Fig. 1c) [3]. The profiles shall hereinafter be compared with one another using the corresponding numbers (1), (2) and (3).

When discussing the profile and number of teeth (here: 5/6; 5 being the male rotor and 6 being the female rotor), all geometric measurements are given in terms of the cross-section as a multiple of the center distance. For operating performance, the length/diameter ratio (L/D), the male rotor wrap angle (φ_{MR}), and the theoretical scoop volumes and the gap variations dependent on them are decisive.

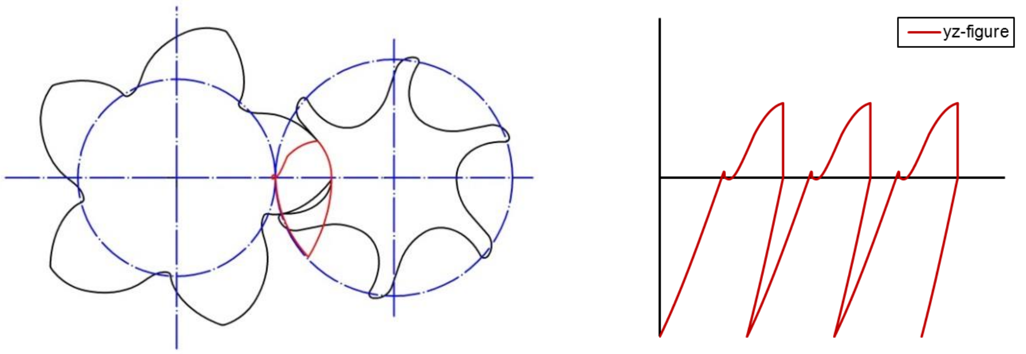

Fig. 1a: GHH 5/6 Profile (1)

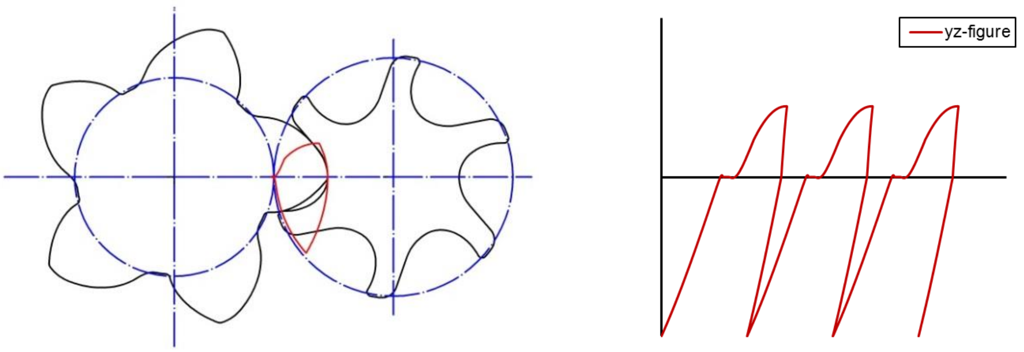

Fig. 1b: Based on Stosic N Profile (2)

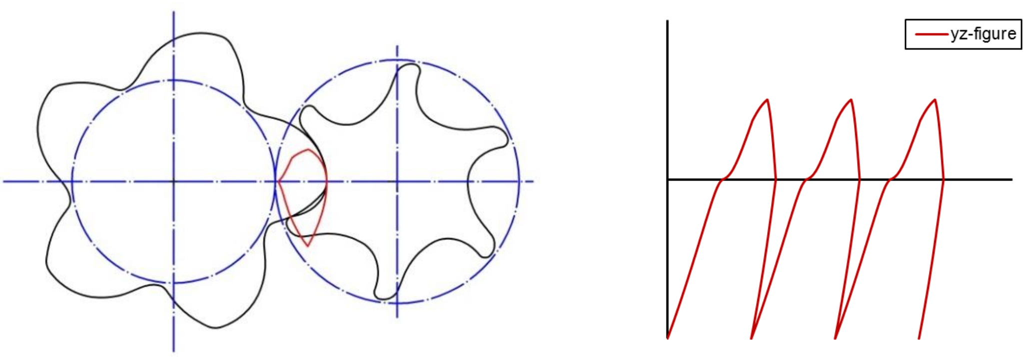

Fig. 1c: Based on Kaeser Sigma Profile (3)

2.2 Basic profile validation

To evaluate profiles, there is a variety of key figures that are calculated based on the geometric parameters. Besides the volume constants, the blow hole, the profile sealing line, the housing gap length and the construction volumes are used as key figures for profile optimization. With the help of a theoretical program to calculate the cross-section and a volume curve program, the key figures shown below can be derived (Fig. 2).

Profile type	Unit	Profile (1)	Profile (2)	Profile (3)
Male rotor wrap angle	°	300.00		
Length-to-diameter ratio	-	1.60		
Center distance	-	1.00		
Profile length	-	1.00	0.98	0.96
Theoretic volume (Male rotor tip speed = 80 m/s)	-	1.00	0.93	0.76
Construction volume	-	1.00	0.94	1.40
Sealing line length per lobe	-	1.00	0.97	0.80
Sealing line influence factor	-	1.00	1.05	1.08
Blow hole area	-	1.00	1.08	3.18

Fig. 2: Overview of rotor profile characteristics

A comparison of the three profile pairings being examined finds that, with the same center distance and the same geometric parameters, profile (1) exhibits the largest theoretical volume at a constant peripheral speed. In terms of construction volumes, the findings show a similarity between profiles (1) and (2) but an increase in profile (3). This is due to the large rotor diameter of the profile. When comparing profile (1) with (3), the length of the operation is constantly reduced whereby, when comparing the factor of the sealing line, profile (1) exhibits the lowest value due to the large theoretical volume. The factor of the sealing line describes the ratio of sealing line length by teeth, related to conveying area and axial pitch. The blow hole always behaves opposite to the path of contact, which profile (3) confirms. In the event of a variation in the geometric parameters, the dimensionless key figures and weighting factors are exhibited quickly. On the other hand, with the help of a simulation of the operating performance of the compressor stage, complex relationships together with their interactions can be demonstrated and examined.

2.3 Advanced profile validation

Below, investigations of variations are performed with the help of a simulation program to examine the influence of different geometric parameters in combination with various profile parameters. Thermodynamic operating performance is clearly in the foreground here. The reference point for the screw compressors to be examined is a peripheral speed of 80 m/s and final pressure of 3.5 bar (absolute), with which a minimum volume flow of 1100 m^3/h is realized. Accordingly, an internal compression ratio of $v_i = 2.19$ is set, which corresponds to pressure of $p_i = 3$.

3. Geometric modelling and configuration of the screw compressor

A chamber model based on Naujoks is the model serving as a basis for the simulation of the thermodynamic process [4]. For a specified increment, all thermodynamic sizes relevant to the working chamber are considered depending on the working cycle for every angular section. The geometry of a rotor is therefore determined by the profile, the number of teeth, the diameter, the length and the wrap angle.

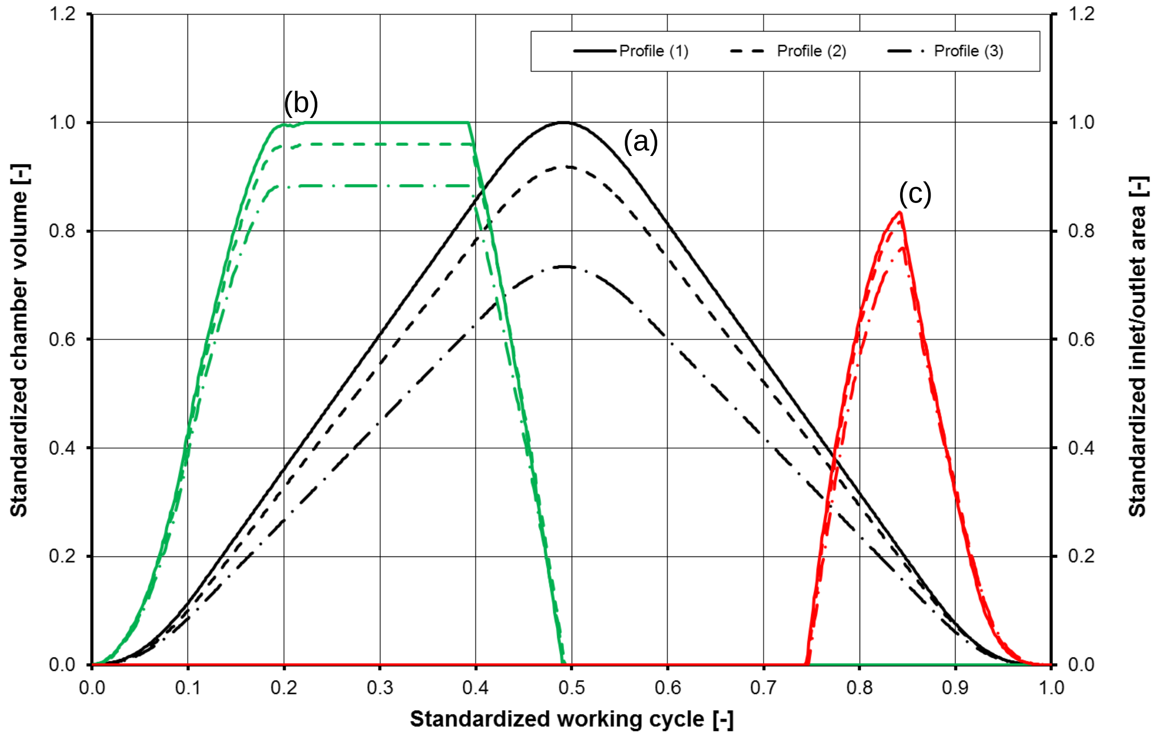

Fig. 3: Standardized volume (a), inlet (b) and outlet (c) curves at same center distance

The simulation model includes the volume curve, connections through the inlet and outlet areas and connections through the gaps (profile inlet, housing, transverse section and blow hole), which are calculated based on the geometric parameters. These sizes can be calculated based on the cross-section when the contour of the profile is given. Our own program developed for this provides the necessary processes over the work cycle. The standardized volume and areas are presented based on the maximum. Figure 3 shows the volume, inlet and outlet processes calculated with the same center distance for three different profiles with the same center distance. The different scoop volumes of the profiles already shown in Figure 2 are shown here graphically. It is clear that the relevant inlet and outlet areas in the machine change based on the different profiles. Figure 4 shows the volume and gap length processes over the standardized work cycle. Profile (1) exhibits a higher scoop volume with almost the same gap length processes in comparison with the other two profiles. Profile (3) has the lowest scoop volumes with reduced profile inlet gap lengths, however, caused by the configuration of the profile with a blow hole area some three times as large as profile (2) or (3).

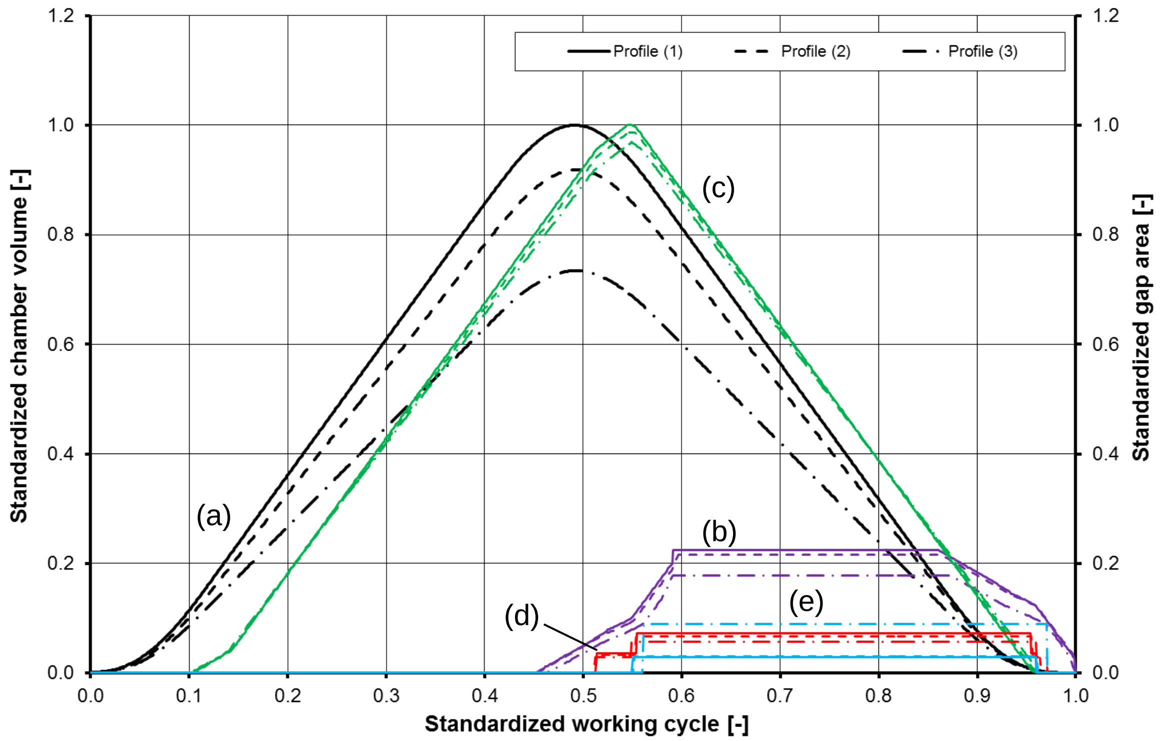

Fig. 4: Standardized volume and gap area curves at same center distance

<u>Reference values:</u> φ_{MR}=300°, L/D=1.6, volume curves (a), profile gaps (b), housing gaps (c), front gaps (d), blow hole (e)

In the chamber simulation model shown, there are connections between the working chambers themselves and also with the suction and pressure sides based on the rotor configuration. Due to the effects of various connections, a varied energy exchange is taken into account by the mass flows. Modelling is subject to the limitation of an adiabatic application of the chamber model, which simplifies the calculation by taking heat conduction into account.

4 Geometric analysis of rotor profiles

To evaluate the thermodynamic operating performance of various combinations of geometric parameters of the rotors, it is also necessary to take the compression ratio into account as a housing parameter, the rotational speed and the operating pressure as operating parameters and the gap heights as production parameters. To keep the number of parameters as low as possible, certain simplifications are made. Thus, for inlet pressure, solely the case of suction under ambient conditions is taken into account, as well as gap heights scaled by way of example with regard to existing machines of the same size. Therefor throttling losses and gap forms will also be considered based on existing machines and their experimental results. The focus of this work is on examining the influence of the L/D ratio and the wrap angle of the rotor pairing depending on the profiles.

4.1 Variation of length over diameter ratio

With a constant wrap angle and center distance, the variation of the length over diameter ratio initially causes a change in the scoop volume. For a constant scoop volume with fixed peripheral speed on the male rotor, the diameter or the center distance must be adjusted in the event of a decreasing L/D ratio. Under the general conditions stated, a variation of the L/D ratio from 1.0 to 2.0 is performed. In the case of the geometry stated, the length, pitch and machine speed increase proportionately in the event of an increasing L/D ratio. The volume, diameter, center distance and gap height decrease correspondingly. The areas of the housing gap and the profile inlet gap also increase in the event of an increasing L/D ratio.

Figure 5 shows the volume with regard to the maximum scoop volumes. The relative gap lengths relate to the maximum gap length, which here corresponds to the addition of the housing gap to the maximum gap length. The gap length per chamber volume decreases with an increasing L/D ratio. Figure 6 shows the internal specific performance declining with increasing flow volume, and volumetric efficiency increasing with increasing flow volume.

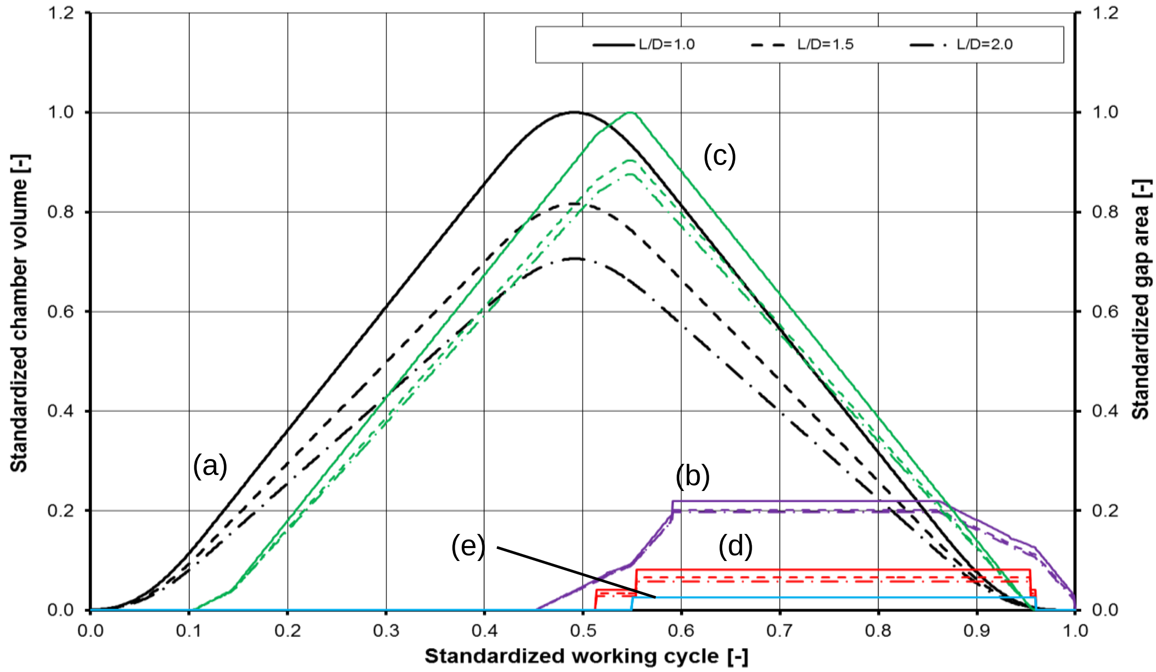

Fig. 5: Influence of length over diameter ratio from 1.0 – 2.0

<u>Reference values:</u> Profile (1), φ_{MR}=300°, Variation of length over diameter ratio, volume curves (a), profile gaps (b), housing gaps (c), front gaps (d), blow hole (e)

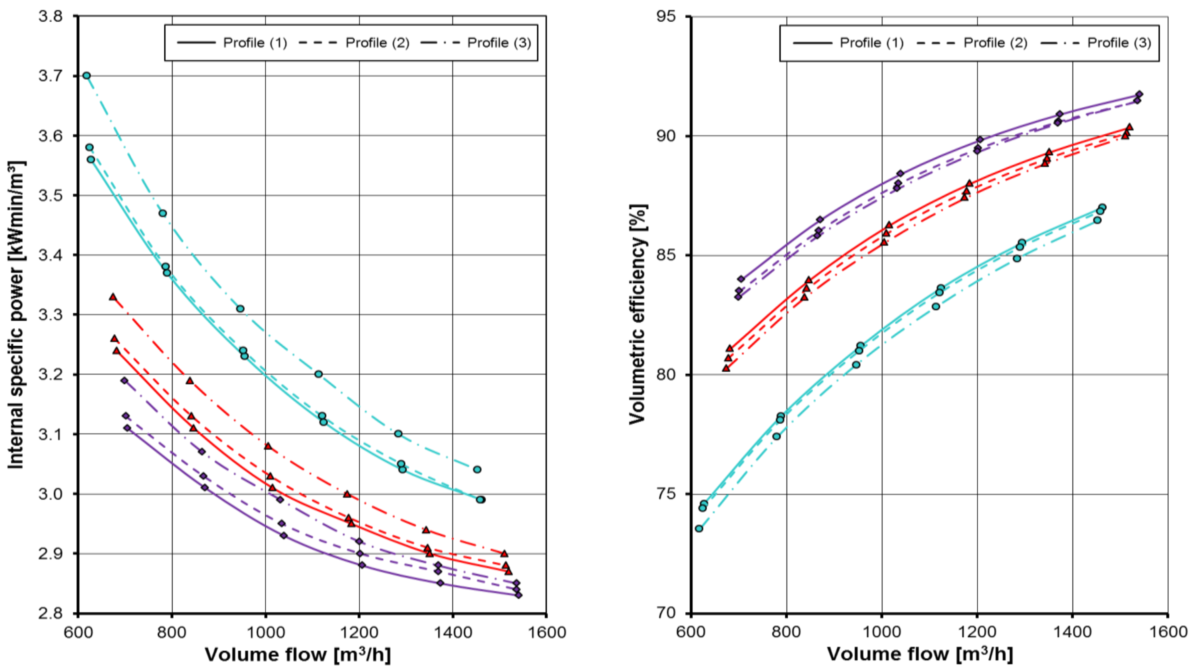

Fig. 6: Specific power and volumetric efficiency over volume flow at 3.5 bar (abs.)

<u>Reference values:</u> φ_{MR}=300°, Variation of length over diameter ratio,

L/D=1.0 (o), L/D=1.5 (△), L/D=2.0 (◇)

In the case of the volumetric efficiency stated, the known relationship between volumetric efficiency and peripheral speed ceases to apply, whereby volumetric efficiency generally increases with a higher L/D ratio. A greater L/D ratio with the general conditions stated effects a smaller rotor diameter, which leads to lower gap heights and higher rotor speed, and explains the processes depicted. Consequently, specific performance increases with a lower L/D ratio and lower peripheral speed. It is only possible to realize a higher peripheral speed with high L/D ratios to a certain extent due to deflection, the rotor dynamics and the useful life of the bearings [5].

4.2 Variation of male rotor wrap angle

In the event of a variation of the wrap angle, the chamber volumes change under the general conditions stated due to changing volume constants. With the desired constant theoretical flow volume with constant peripheral speed, the diameter increases. The most significant change, however, is the change in the length of the work cycle, which has a considerable influence on thermodynamic operating performance. In Figure 7, the fundamental geometric parameters in the event of a variation in the wrap angle in the region of 250 to 325 degrees and its influence on the simulation model are considered.

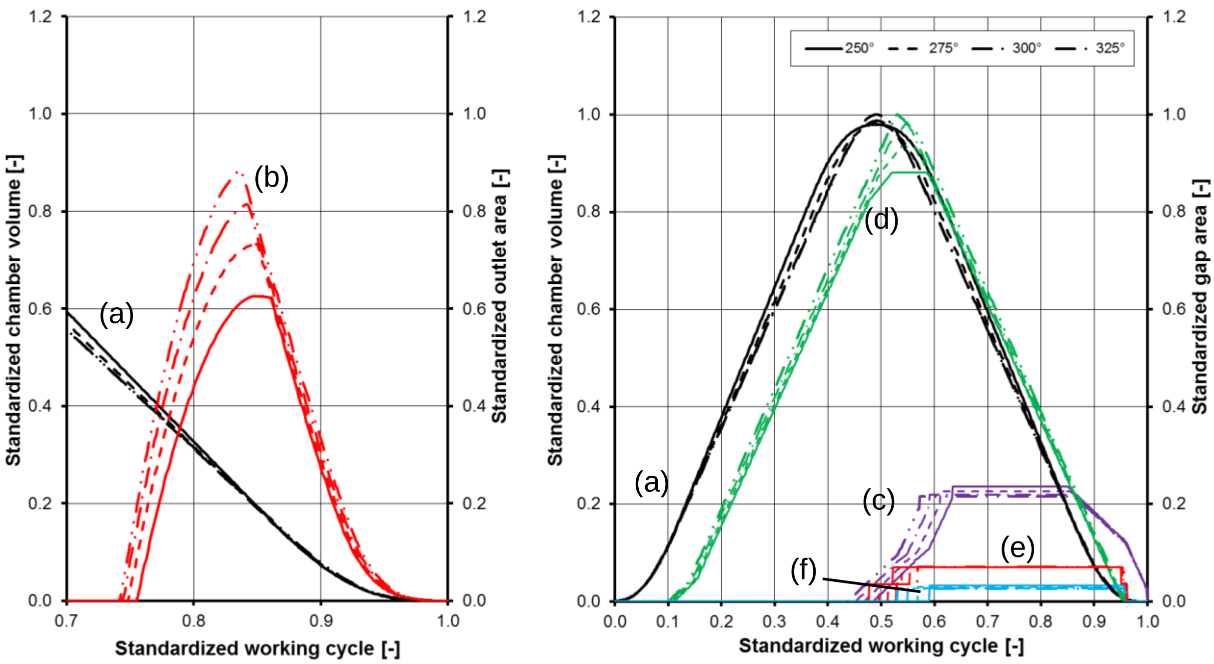

Fig. 7: Influence of male rotor wrap (φ_{MR}) angle from 250-325°

<u>Reference values:</u> Profile (1), L/D=1.6, volume curves (a), outlet area (b), profile gaps (c), housing gaps (d), front gaps (e), blow hole (f)

The volumes are depicted again in relation to the maximum scoop volumes; in turn, the relevant gap lengths relate to the maximum gap length. The outlet areas, diameter, gap lengths and the center distance increase with an increasing wrap angle due to the relationships described above. The outlet areas increasing with an increasing wrap angle also cause an increasing discharge period, as the drain angle increases. The discharge period thus has a positive effect on the thermodynamic operating behavior, as a longer discharge period generally has a reduced throttle effect on the outlet. On the other hand, the mass flow has a negative effect due to the profile inlet split, whereby the volumetric efficiency is reduced. The gaps are clinched in the event of an increase in the wrap angle in the direction of the rotor axis. This is particularly significant when considering the profile inlet gap. Due to the change in the sealing line length with the increasing wrap angle reduced, the profile inlet process increases. On the other hand, when the housing gap process is totalized, there is an increase in the process with an increasing wrap angle. In the case of higher wrap angles, the gaps thus exist over a longer period of time. In the case of lower wrap angles, the gaps turn out to be slightly larger in the discharge period.

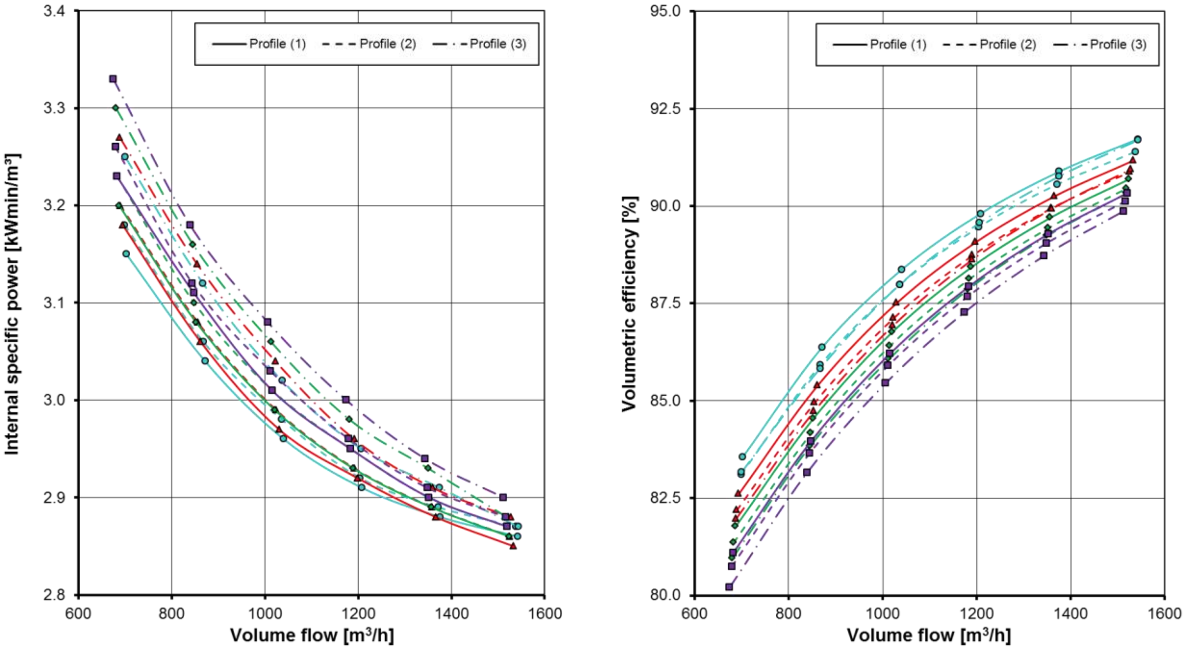

Fig. 8: Specific power and volumetric efficiency over volume flow at 3.5 bar (abs.)
Reference values: L/D=1.6, Variation of male rotor wrap angle, φ_{MR}=250° (○), φ_{MR}=275° (△), φ_{MR}=300° (◇), φ_{MR}=325° (□)

The geometric changes in the wrap angle variation are demonstrated in Figure 8 by the known depiction of the specific performance and volumetric efficiency for an outlet pressure of 3.5 bar (absolute). Due to the small changes in the rotor dimensions with constant volumes, there are similar gap heights. Thus, the growing sealing line length and housing gap length are mainly responsible for the reduction in volumetric efficiency with increasing wrap angles in all profiles. The increasing outlet area results in a minimal increase in specific performance with increasing wrap angles in all profiles. Here, especially in the lower and medium speed range, profile (1) exhibits low specific performance and, to a large extent, higher volumetric efficiency compared to profiles (2) and (3).

4.3 Combined variation of the geometric parameters

The result of the simultaneous variation of the L/D ratio and the wrap angle is that the changes in the geometric parameters interfere with one another. With a greater L/D ratio and smaller wrap angle, the diameter of the male rotor decreases. The length of the rotors arises from their diameter and the L/D ratio, whereby the relative change in length in the event of a variation in the wrap angle is proportionate to the relevant change in the diameter.

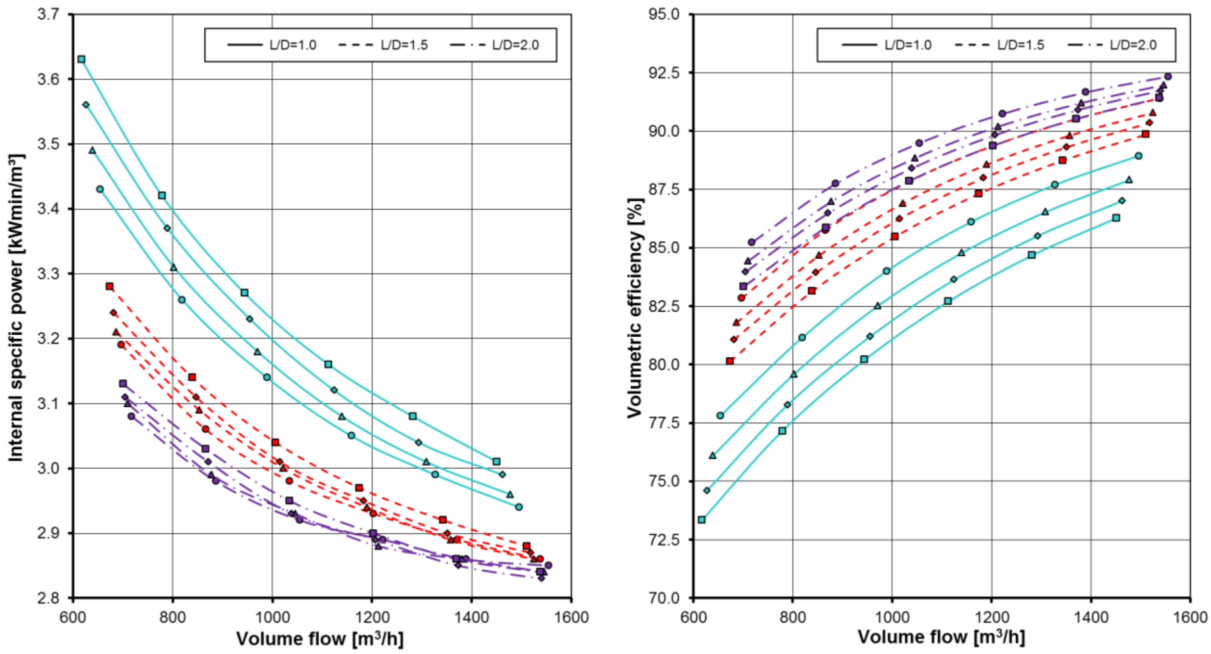

Fig. 9: Specific power and volumetric efficiency over volume flow at 3.5 bar (abs.)
<u>Reference values:</u> Profile (1), Variation of length over diameter ratio and male rotor wrap angle, φ_{MR}=250° (○), φ_{MR}=275° (△), φ_{MR}=300° (◇), φ_{MR}=325° (□)

As the behavior of the gaps in the profile inlet sit and the housing gap is sometimes opposite in the event of a variation in the wrap angle, the relationship between them also changes. In the event of a variation in the L/D ratio, this relationship only changes slightly due to the general conditions selected. In the event of greater wrap angles, the gap of the profile inlet gap turns out to be smaller relative to the housing gaps.

For clarity, Figure 9 only shows L/D ratios of 1.0, 1.5 and 2.0. The processes of volumetric efficiency shown as an example of a profile, here profile (1), demonstrate an increase in volumetric efficiency with an increasing L/D ratio. Lower wrap angles show, as a percentage, higher volumetric efficiencies than profiles with higher wrap angles on the male rotor. Thus machine geometries cause higher L/D ratios and lower wrap angles generally cause higher volumetric efficiencies in the profile examined. With higher rotation speeds, machine geometries with a medium L/D ratio and lower wrap angle achieve similar volumetric efficiencies as configurations with higher L/D ratios and higher wrap angles. Specific performance decreases with an increasing L/D ratio. The minimum of the specific performance shifts in the case of higher peripheral speed to smaller L/D ratios and slightly higher wrap angles. Lower wrap angles with lower rotation speeds and higher wrap angles from medium rotation speeds exhibit lower specific performances. With profiles (1) and (2), wrap angles from 300 degrees exhibit the lowest specific performances with higher rotation speeds.

5 Summary

This work describes the relationship between key geometric profile figures of screw compressors and the associated examination of thermodynamic operating performance to analyze rotor geometry. The geometric parameters wrap angle and L/D ratio with various profiles and a stipulated internal pressure ratio are considered in detail, to examine their influence on thermodynamic operating performance. With a variation in the geometric parameters with the same numbers of teeth, different profiles yield similar results. This is particularly clear when considering key geometric figures for profiles (1) and (2), which do not exhibit any great differences. However, the findings of the combined examination with the variation parameters wrap angle and L/D ratio do show a tendency for one profile. Profile (1), which corresponds to the GHH profile, exhibits the optimum configuration at higher tip speeds with a wrap angle of 300 degrees and an L/D ratio of 2.0.

In general, it is found that, in the case of profiles with the number and combination of teeth examined, machine geometries with high L/D ratios are generally desirable. The limitation of higher L/D ratios is given by requirements of mechanics like deflection and the bearing lifetime. When choosing a suitable wrap angle, profiles with a wrap angle of 300 degrees are recommended. Compared with other profiles, profile (1) - with almost the same scoop volumes - exhibits the lowest specific performances overall and, with regard to increasing the efficiency of screw compressors, is preferred over the other profiles. To conclude, it is also noted that the simulation tool can evaluate different profiles with regard to thermodynamic operating performance but, due to the number of other influences such as the thermal deformation of rotors and housing, deflection, etc., it is essential that this profile examination is validated experimentally.

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