Thermodynamical operation behaviour of screw vacuum pumps with Cycloid- and Quimby-toothed rotors

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1. Abstract

Within the scope of this article, a comparison of screw vacuum pumps with different profile shapes concerning the thermodynamical operation behaviour is carried out. To point out specific strengths and weaknesses an asymmetric quimby profile with one tooth and a symmetric cycloid profile with two teeth is chosen. Both profiles take advantage of a variable pitch curve along the rotation axis. After pointing out basic characteristics of the geometry of both profile shapes concerning the cross-section as well as clearances an investigation of the thermodynamical operation behaviour is carried out with the help of the simulation tool "*KaSim*". The suction speed and the inner power as a function of the inlet pressure of both profiles are presented varying the rotational speed. In the case of the cycloid profile the pitch curves along the rotation axis are optimised for a specific inlet pressure and rotational speed. Furthermore, the influence of the clearance height of both profiles on the suction speed is evaluated.

It can be shown that greater clearance areas of the cycloid profile require a higher rotational speed in comparison to the quimby profile. This can be overcome by using a two teeth cycloid profile, where a good mechanical balancing can be achieved. Regarding the suction speed and the required inner power of a two teeth cycloid profile with a higher rotational speed results similar to the quimby profile are observed. Also a smaller construction volume of the cycloid profile is required in this case.

2. Introduction

Screw vacuum pumps are widely used as fore-vacuum pumps in different applications. The main advantage compared to rotary vane pumps is the oil-free working chamber. Due to this and the contactless working principle of the two rotors, less maintenance is needed. The thermodynamical calculation for construction purposes has been a very important field of research in the past. For this, the knowledge of clearance gas flows is crucial. In screw vacuum pumps a wide pressure range has to be handled, which necessitates the consideration of clearance flows from very low to high Knudsen numbers. Besides the Knudsen number the dimensions of the clearances have an important impact on the operating behaviour of screw vacuum pumps. Increasing possibilities in production techniques have led to the use of rotors with a variable pitch to realize an internal compression. Used profile shapes are mainly grouped into symmetric and asymmetric profile shapes [1]. Screw vacuum pumps with asymmetric profile shapes commonly consist of a so-called quimby profile. On the other hand, screw vacuum pumps with a symmetric profile are based on a trapezoidal, involute or cycloid shape.

Table 1: Constant geometrical parameters

	quimby profile (geometry A)	**cycloid profile (geometries B and C)**
crown to root circle d_C/d_R	2.8	1.62
pitch circle diameter d_P	-	$(d_C+d_R)/4$
length to crown circle l/d_C	2.4	2.4
wrap-angle φ_W	1440°	1440°
inner volume ratio v_i	3.0 (variable pitch and end plate)	3.0 (variable pitch)

In the following, a comparison between an asymmetric quimby profile with one tooth and a symmetric cycloid profile with two teeth is carried out. In the case of the quimby-toothed rotors, a machine is investigated, which is already tried and proven in practice. Thus, no basic geometrical variation is carried out in this machine. For the cycloid toothed rotors, a variation of the geometry is carried out keeping some geometrical parameters constant. **Table 1** summarizes the important parameters of both profiles.

3. Cross-section of profiles

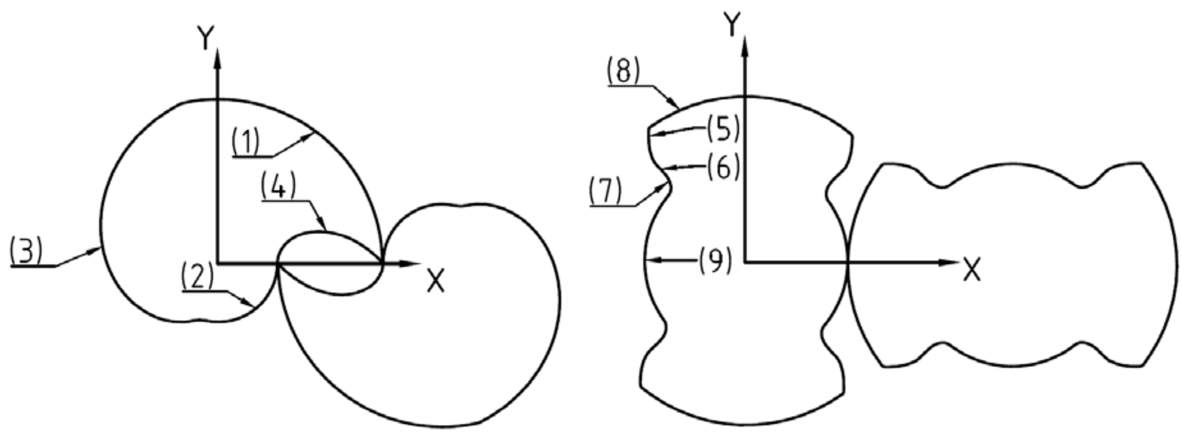

Fig. 1: Cross section of profiles with quimby- and cycloid-toothed rotors

Fig. 1 shows the cross-section of two profiles with cycloid and quimby toothing, as they are likely to be used in different applications. While the cycloid toothing has a symmetric profile cross-section, the quimby toothing has an asymmetric cross-section. Both profiles can be described completely with the help of subsegments. The quimby toothing, shown on the left side of Fig. 1, consists of an outer crown circle (1) and an inner root circle (2). Both segments are connected with an archimedean spiral (3) and a trochoid segment (4). The approach of constructing the quimby profile is taken from Kawamura et al. [2]. The first subsegment of the cycloid toothing (5), shown on the right side of Fig. 1, can be constructed by a pitch circle passed on the top of a roll circle. The resulting curve, called epicycloid, forms the upper part of the tooth flank. The complement of the epicycloid can be constructed by passing the pitch circle inside the roll circle and is called hypocycloid (6). To achieve a closed working chamber, the lower part of the tooth has to be modified (7). This is done by creating an envelope of the top point of the epicycloid. The approach of modifying the lower part of the tooth is chosen in compliance with the description of Rofall et al. [3]. The constructed flanks are connected with the help of the crown circle (8) and the root circle (9).

4. Comparison of different profile variants

The previous described methods of generating the cross-section can be done for profiles with a different number of teeth in principal. Several obvious characteristics of the cycloid and quimby profile as well as profiles with one or two teeth are presented in **Table 2**.

Table 2: Characteristics of cycloid or quimby profile with one or two teeth

cycloid profile ➔ one chamber each rotor side ➔ less undercut (manufacturing issue) ➔ possibility of two teeth-profile	**quimby profile** ➔ length of crown circle segment is a free parameter ➔ negligibly small blowhole at one side
one tooth (cycloid and quimby profile) ➔ clearances only connecting nearby chambers	**two teeth (cycloid profile only)** ➔ good mechanical balancing ➔ higher rotational speed possible ➔ more series connected chambers at constant wrap angle

The first main difference between both profile shapes is that one chamber is located on each rotor side of the cycloid profile. Contrary to the quimby profile, the symmetric cycloid profile has equal blowholes on the front and on the back side of the tooth flank. The quimby profile possesses a negligibly small blowhole at the front side and a large blowhole at the back side of the tooth flank, thus a chamber is located at both rotor sides. Moreover, the extended chamber prohibits the use of a two-tooth quimby profile. Another aspect of the cycloid-toothed rotor is the easy possibility of designing a rotor without undercut, which could be a manufacturing issue. The free parameter of the length of the crown circle segment is another advantage of the quimby profile. This parameter can be used to find an optimum between the displaced volume and the length of the housing clearance in flow direction to increase the throttling of this clearance [2].

When the profiles with one and two teeth are compared, the better possibility of mechanical balancing of rotors with two teeth is the most obvious difference. This directly leads to less vibration and the opportunity to realize higher rotational speeds. Additionally, an advantage of profiles with two teeth is the higher number of chambers connected in series and thus the expectation of a higher throttling of clearance backflows.

5. Chamber model

As described, an abstraction of the geometry for both profile shapes has to be carried out to investigate the thermodynamical operation behaviour. **Fig. 2** illustrates the chamber model of the cycloid profile with two teeth and the quimby profile with one tooth. In both diagrams all clearance connections from the viewpoint of chamber i are illustrated.

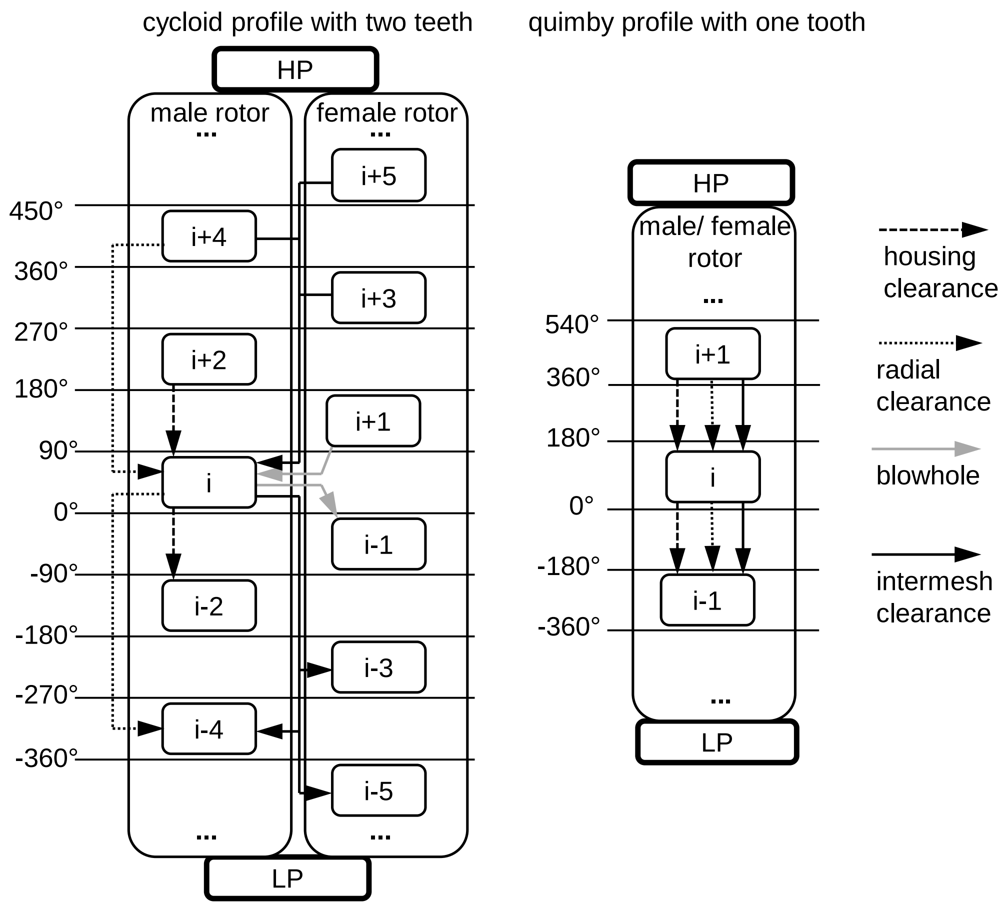

Fig. 2: Chamber diagram of the cycloid and quimby profiles

As it can be seen in Fig. 2, the main difference between both profiles is the modelling of only one chamber for the male and female rotor side in the case of the quimby profile. The large area of the blowhole at the back side of the tooth flank is caused by the asymmetric profile.

For the simulation, an integral connection is chosen, which means that in any time compensation in pressure is calculated by means of mass and energy conservation between both chambers. The second main difference can be seen in the radial and intermesh clearances. For the profile with two teeth, these clearances omit the adjacent chambers and connect chambers that have greater axial distances. The third difference is due to the missing blowhole in the quimby profile at the front side of the tooth flank. Because of the sharp-edged transition at the end of the crown circle segment and the trochoid segment, as it can be seen in Fig. 1, a negligible small blowhole is created. Thus, it is not considered in the simulations.

6. Comparison of clearance dimensions

To evaluate the thermodynamical operation behaviour of screw vacuum pumps, the knowledge of the geometrical characteristics is essential. For this purpose, the normalized clearance areas of exemplary chambers with cycloid and quimby profiles are depicted in **Fig. 3**. This points out the specific differences between them.

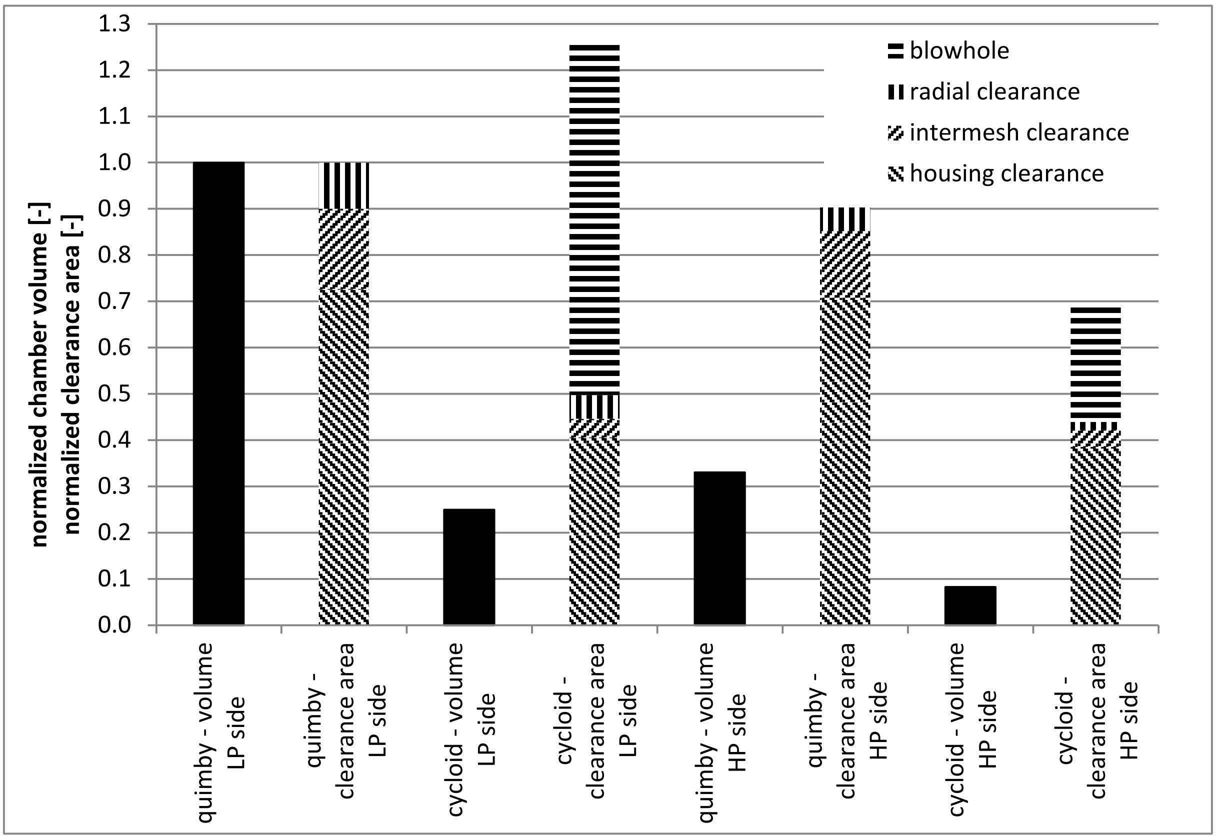

Fig. 3: Chamber volume and clearance areas of exemplary rotors with cycloid and quimby profiles and equal displaced volume normalized by the characteristics of the quimby profile on the low pressure side

To ensure comparability, two profiles with equal crown circles are considered. In addition, equal displaced volumes of the cycloid and the quimby profiles are chosen, such as they typically appear on the low-pressure (LP) and high-pressure (HP) sides. The chamber volume can be varied by adjusting the pitch of the profiles and as it can be seen in Fig. 3, despite an equal displaced volume, the chamber volume varies with the number of working cycles per revolution. This implies that in the case of the cycloid profile with two teeth, the chamber volume is traversed four times per revolution. In general, a greater clearance area for the radial, intermesh and housing clearance can be seen in the quimby profile because the chamber is located on both sides of the rotor. In combination with the lower number of teeth of the quimby profile, a greater chamber volume results. The remarkable clearance area of the blowhole causes a bigger overall clearance area in the case of the chamber on the low-pressure side of the cycloid profile. If the pitch is decreased, to obtain a smaller chamber volume, a distinct decrease in the clearance area of the blowhole is seen. Thus, a smaller overall clearance area for the cycloid profile is reached.

7. KaSim

The following theoretical analysis of both profiles is done by help of the simulation tool "*KaSim*", which enables the thermodynamical calculation of positive-displacement machines. For this purpose the positive-displacement machine is abstracted into a zero-dimensional multi-chamber model. Clearances are represented by means of connections, which enables the calculation of the interaction between the different chambers [4]. To simulate the operation behaviour of screw vacuum pumps, the determination of clearance mass flows is crucial. In particular, the wide pressure range and hence the wide range of Knudsen numbers in the clearance flow that have to be handled, is the main difficulty. For this reason, an experimental investigation of the flow for different clearance shapes was carried out and the results are used within the simulation process [5]. In addition, Stratmann [6] pointed out that the influence of a moving clearance boundary has a significant influence on the flow for higher Knudsen numbers in the Knudsen- and molecular-flow range. Therefore, a theoretical investigation of the influence of moving clearance boundaries was carried out by means of a test-particle method, which is valid in the molecular-flow range. The results of the theoretical investigation are also used in the simulation process of "*KaSim*". Stratmann was able to prove that there is a good agreement between simulation and experiment for screw vacuum pumps. Based on these findings, thermal effects are not considered in the investigation presented in this paper. The findings also support our confidence in the simulation presented in this paper.

8. Optimization of pitch curve

As it has been demonstrated in Fig. 3, the pitch of the profile does not only have an influence on the chamber volume, but on the clearance areas as well. In particular, the blowhole of the cycloid-toothed rotor has a great dependence on the pitch. Therefore, an optimized pitch-curve along the rotor axis has to be found, which represents a compromise between the volume and the clearance area of every chamber. Pfaller took up this problem and developed an evolutionary approach to optimize the pitch along the rotor axis of cycloid-toothed rotors [7]. The optimization criteria are represented by the sucked mass flow and the inner power of the machine. The input of an optimization calculation is represented by the cross-section of the profile as well as the clearance heights, the length of the rotors and the wrap angle. Additionally, an operating point, represented by a rotational speed and an inlet pressure has to be provided. The result of the calculation is a number of different pitch-curves, with each one representing an optimum between the sucked mass flow and the inner power of the machine.

9. Quimby and optimized cycloid with same rotational speed and dimension

To obtain the first comparison between both profile shapes, two machines with identical rotational speed, rotor crown circle and rotor length are investigated. The sizes of the crown circle and the length of the machine with the quimby profile are used as references. Additionally, the pitch along the rotation axis of the cycloid profile is optimized by using the approach developed by Pfaller [7].

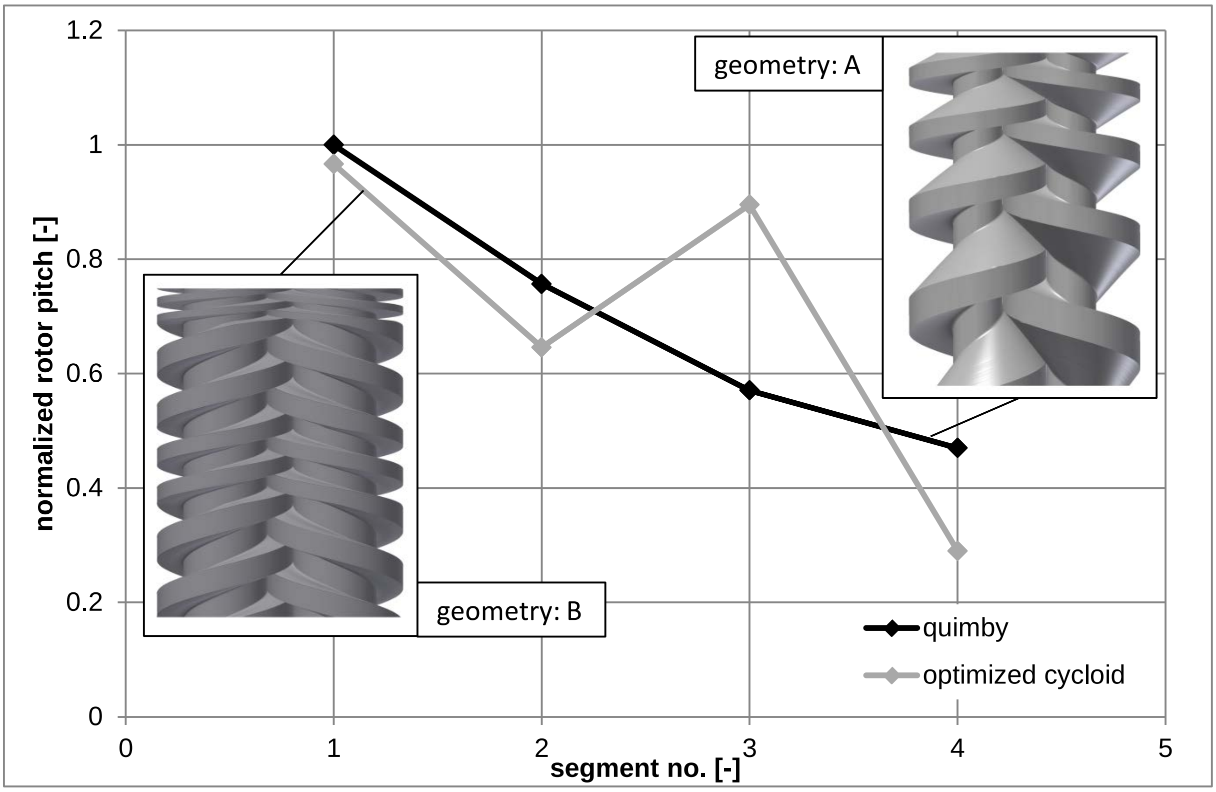

Fig. 4: Pitch-curve of the quimby profile (geometry A) and optimized pitch-curve of the cycloid profile for a rotational speed of 3000 min^{-1} and an inlet pressure of 10^4 Pa (geometry B) normalized by the maximum pitch of the quimby profile at the low pressure side

The operation point, which the pitch-curve is optimized for, is chosen at a suction pressure of 10^4 Pa and a rotational speed of 3000 min^{-1}. A pitch-curve with an inner volume ratio of $v_i = 3$ is chosen for the cycloid profile, which is similar to the quimby profile. Both profiles and their respective pitch curves, normalized by the maximum pitch of the quimby profile are shown in **Fig. 4**. The rotors are divided into four segments defined by a wrap angle of 360°. Each of these segments has a constant pitch. One important characteristic of the quimby profile is the use of an additional end plate. Beside the variable pitch the end plate is used to increase the inner volume ratio, which leads to the overall volume ratio of $v_i = 3$. **Fig. 5** shows the simulation results of suction speed and inner power of both machines as a function of the inlet pressure obtained by using "*KaSim*". The values are normalized by the simulation results of the quimby profile at an inlet pressure of 460 Pa.

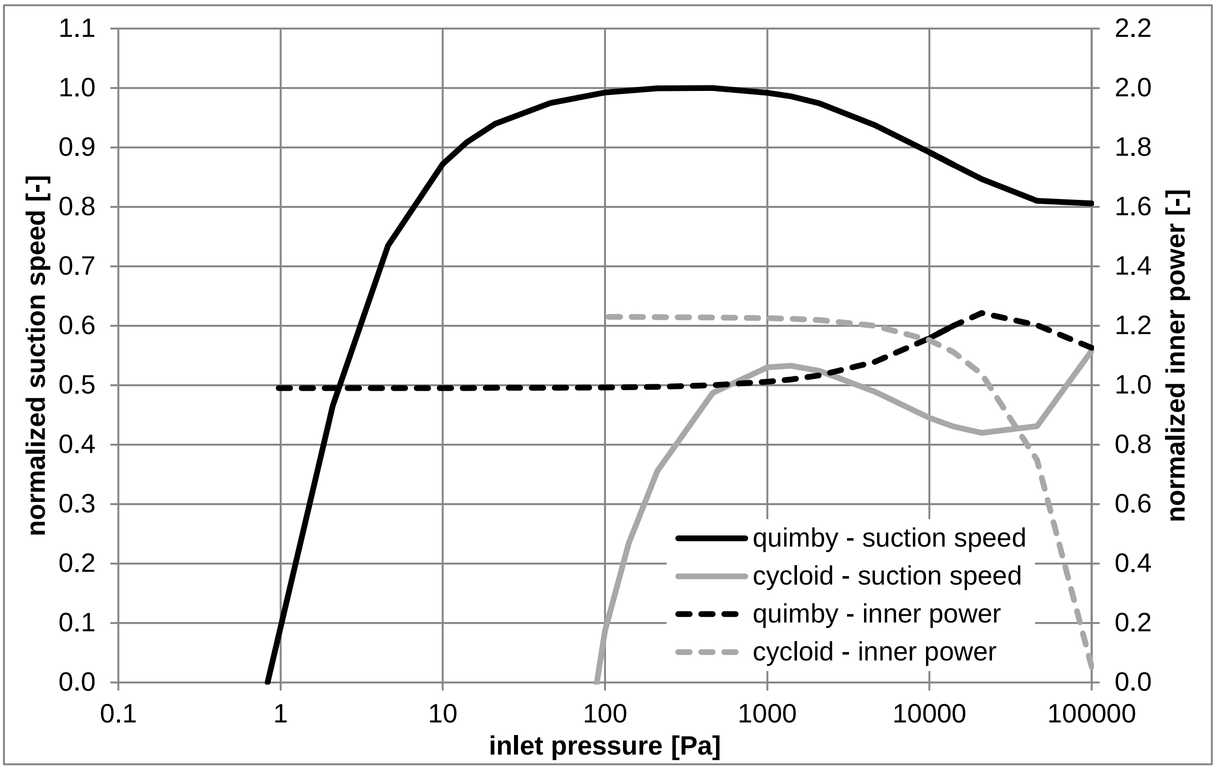

Fig. 5: Suction speed and inner power of rotors with quimby (geometry A) and cycloid profile (geometry B) with equal crown circle, rotor length and rotational speed, normalized by the values of the quimby profile for an inlet pressure of 460 Pa

With respect to the suction speed of the quimby profile, a typical pumping performance of a screw vacuum pump can be observed. For high inlet pressures, close to the atmospheric pressure, a lower suction speed is obtained as a result of over-compression and thus a high backflow through the clearances. For decreasing inlet pressures, higher suction speeds are achieved because of the missing overcompression as well as the increasing throttling effect of a rarefied gas flow in the clearances. For a low inlet pressure the clearance mass flow at the low pressure side is equal to the sucked mass flow of the working chamber and the ultimate pressure of the machine is reached. High power consumption can be observed with respect to the inner power of the machine with quimby profile for an inlet pressure close to 10^5 Pa. Again, this can be explained by an overcompression of the working fluid. With decreasing inlet pressure, the influence of the overcompression declines but despite this a higher inner power is needed because of the increasing pressure difference the machine has to overcome. When the inlet pressure is further decreased, the power consumption of the chambers generating the inner compression at the low pressure side become negligibly small due to very small mass flows, thus decreasing the overall power consumption to a constant level.

Comparing the machines with the quimby and the cycloid profiles, a greater suction speed can be observed for the quimby profile in the entire pressure range. A smaller displaced volume per revolution of the cycloid profile (about 0.71 times) in combination with the greater clearance areas creates the disadvantageous operation behaviour, despite the optimized pitch-curve of the cycloid profile. Considering the inner power of both profiles for an inlet pressure of 10^5 Pa, less power consumption of the cycloid profile is reported. Bigger clearance areas, which cause a smaller suction speed, indicate an advantage in the power consumption because of a reduction in the over-compression. With decreasing inlet pressure, the effect of over-compression disappears and a higher power consumption of the cycloid profile can be observed. For these operation points, higher clearance mass flows because of the greater clearance areas cause a greater inner power.

10. Quimby and scaled cycloid

It has been shown that big clearance areas of the cycloid profile lead to an inefficient operation behaviour at a low rotational speed when compared to the quimby profile. One big advantage of the cycloid profile with two teeth is the possibility of operating at a higher rotational speed. In the following, this advantage is taken into account and a rotational speed of 8000 min^{-1} is chosen for the cycloid profile. The pitch curve of the cycloid profile is optimized using the approach of Pfaller [7] for an operation point of 8000 min^{-1} and an inlet pressure of 10^4 Pa. As it can be seen in **Fig. 6**, an optimized pitch curve is chosen that leads to an inner volume ratio of v_i = 3.0. For better comparability, the cycloid-toothed profile is scaled down to the same maximum in suction speed related to the quimby-profile using "*KaSim*". The scaling is carried out taking into account full geometrical similarity and the scaling factor is calculated as f_{SC} = 0.74.

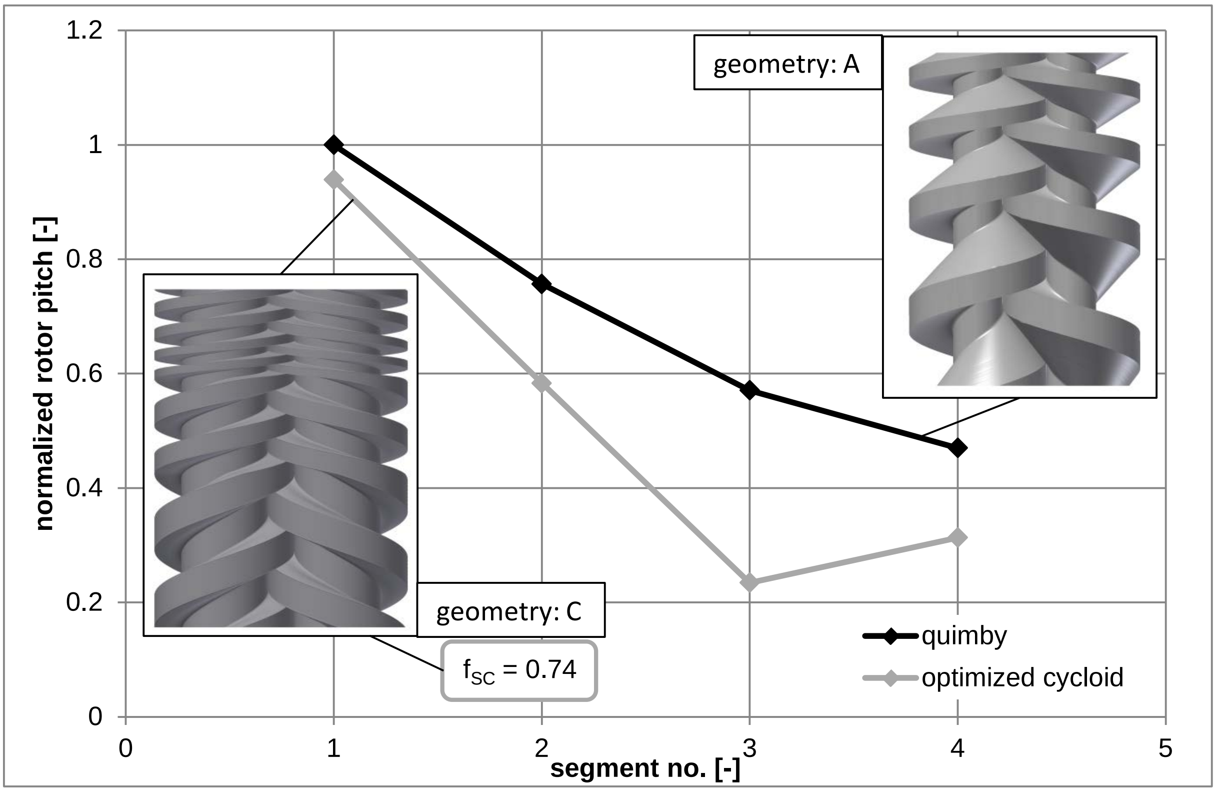

Fig. 6: Pitch-curve of the quimby profile (geometry A) and optimized pitch-curve of the cycloid profile for a rotational speed of 8000 min^{-1} and an inlet pressure of 10^4 Pa scaled to an equal suction speed (geometry C). Values are normalized by the maximum pitch of the quimby profile at the low pressure side

Fig. 7 shows the suction speed and the inner power of the machine with quimby profile and the scaled machine with cycloid profile. A slightly higher suction speed of the cycloid profile is seen for an inlet pressure greater than 10^3 Pa and a lower suction speed for an inlet pressure smaller than 10^3 Pa. A slightly greater theoretical suction speed of the cycloid profile is beneficial when operating at a higher inlet pressure. For a lower inlet pressure, the importance of a moving clearance boundary becomes more significant. As pointed out by Stratmann [6], the relative motion of the rotors has a non-negligible influence on the clearance mass flow at low gas densities. To illustrate this, the suction speed of the cycloid profile without considering the moving clearance boundaries (static boundary) is also shown in Fig. 7. The lower suction speed of the cycloid profile can thus be explained by the higher clearance boundary velocities due to the higher rotational speed increasing the clearance mass flow at the low-pressure side.

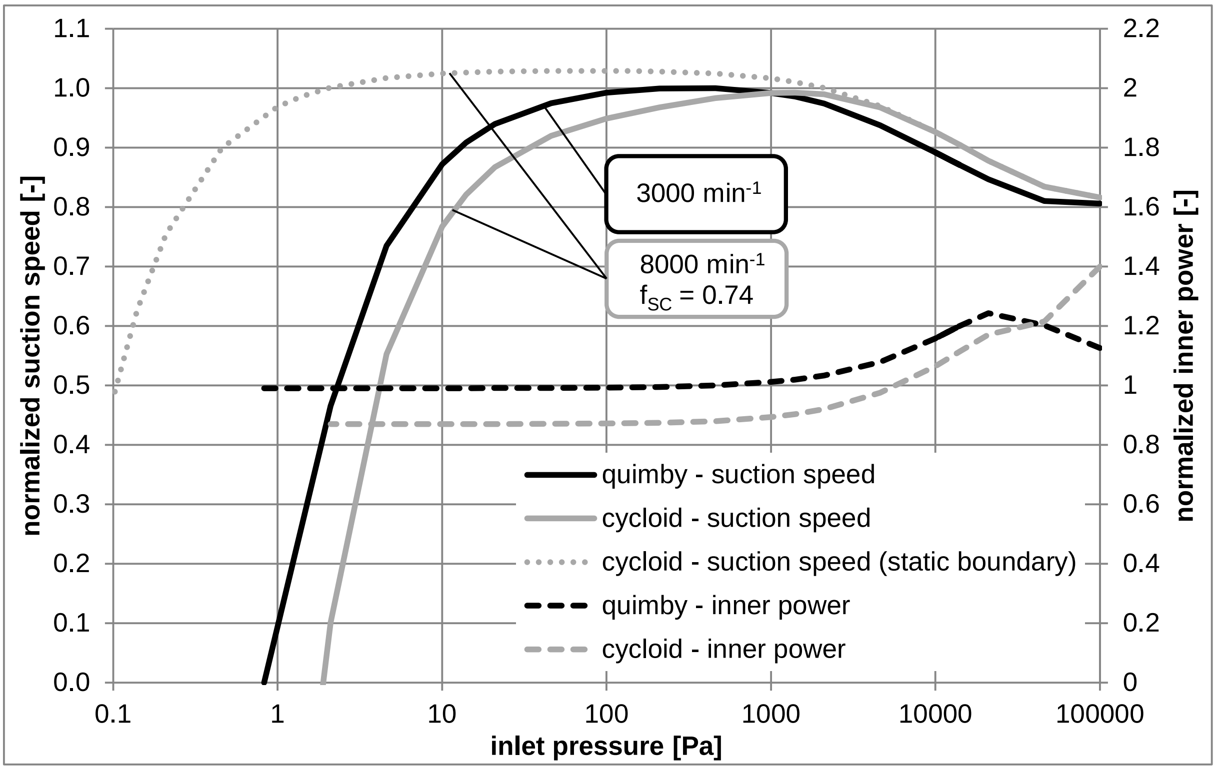

Fig. 7: Suction speed and inner power of rotors with quimby (geometry A) and scaled cycloid profile (geometry C) with moving boundary for different rotational speeds and without moving boundaries for the cycloid profile. Results are normalized by the values of the quimby profile at an inlet pressure of 460 Pa

For an inlet pressure near the atmospheric pressure, a higher inner power consumption of the cycloid profile is observed. Due to the higher rotational speed, the beneficial influence of the clearance mass flow decreases. Thus, a higher over-compression can be observed. With decreasing inlet pressure the effect of over-compression declines and because of the less overall clearance mass flow at the high-pressure side of the cycloid profile, a smaller inner power is needed compared to the quimby profile.

11.Influence of clearance height

In the design phase of a screw vacuum pump, the choice of the clearance height plays an important role. Thermal deformations as well as tolerances for manufacturing purposes have to be taken into account to guarantee a contact-free and save operation. Thus, variation of the clearance heights is carried out to estimate the influence on the operation behaviour. **Fig. 8** shows the suction speed of the cycloid profile and the quimby-profile normalized by the maximum suction speed of the quimby profile. The clearance height is varied in the range of +-10 percent.

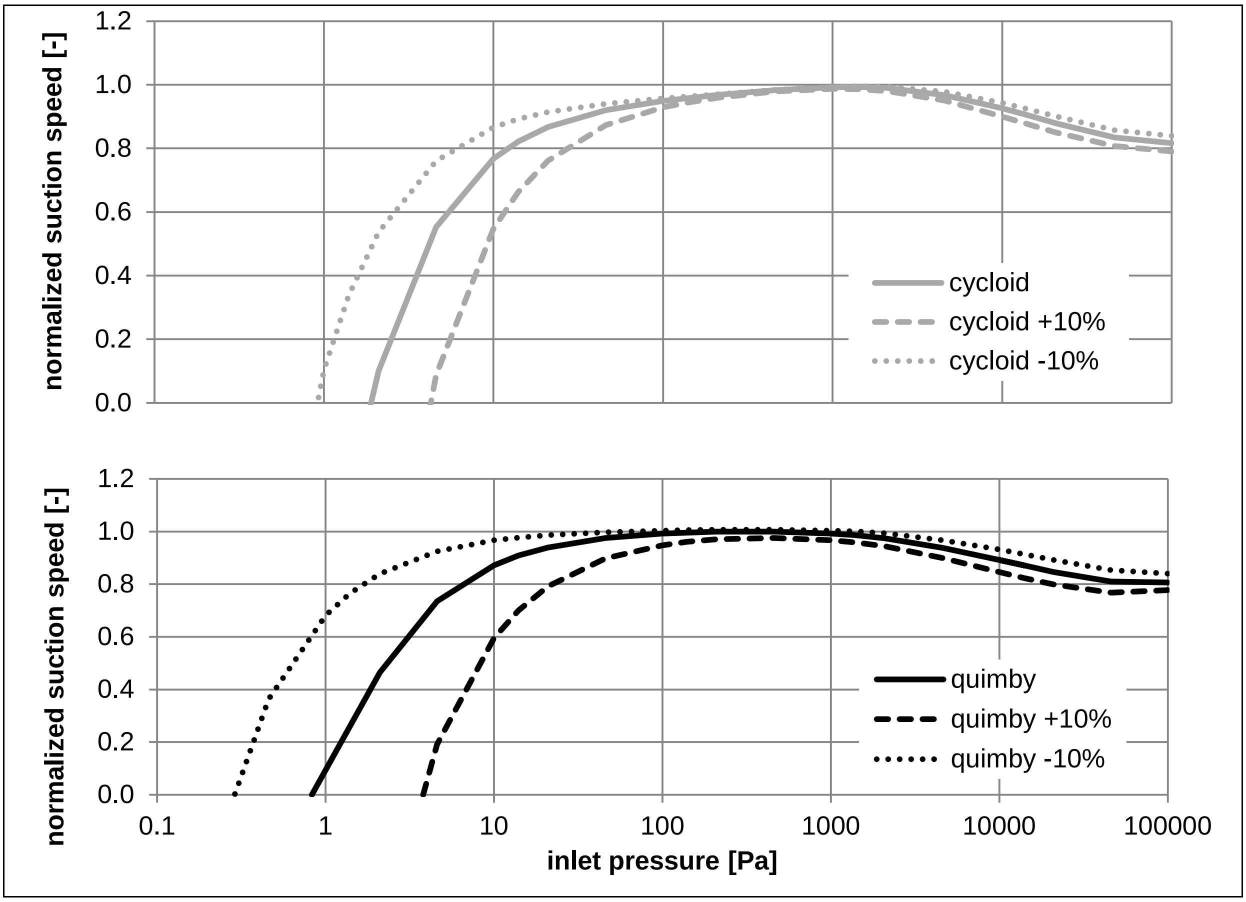

Fig. 8: Suction speed of quimby (geometry A) and cycloid profile (geometry C) under consideration of different clearance heights normalized by the suction speed of the quimby profile at an inlet pressure of 460 Pa

As can be seen, the variation of the clearance height has a small influence when operating at a high inlet pressure but a high influence in the calculation of the ultimate pressure. Because of the high throttling effect of the clearances for low gas densities, the variation of the clearance height has only a very small influence on the suction speed in the range of 10^2 – 10^3 Pa inlet pressure. When operating at an inlet pressure near the ultimate pressure, the clearance mass flow at the low pressure side is nearly equal to the suction mass flow of the machine. A smaller variation in the ultimate pressure of the cycloid profile can be seen due to the higher rotational speed in combination with the smaller overall dimension. Thus, it is easier to guarantee an ultimate pressure of a manufactured machine despite different thermal deformations and variations in the clearance height due to manufacturing inaccuracies.

12.Variation of rotational speed

Another possible application of a screw vacuum pump is maintaining a constant pressure of a vacuum chamber. For reasons of energy saving, this could be done by adjusting the rotational speed of the pump. To observe the behaviour of both profiles of this task, **Fig. 9** shows an investigation of the suction speed and the inner power as a function of the rotational speed.

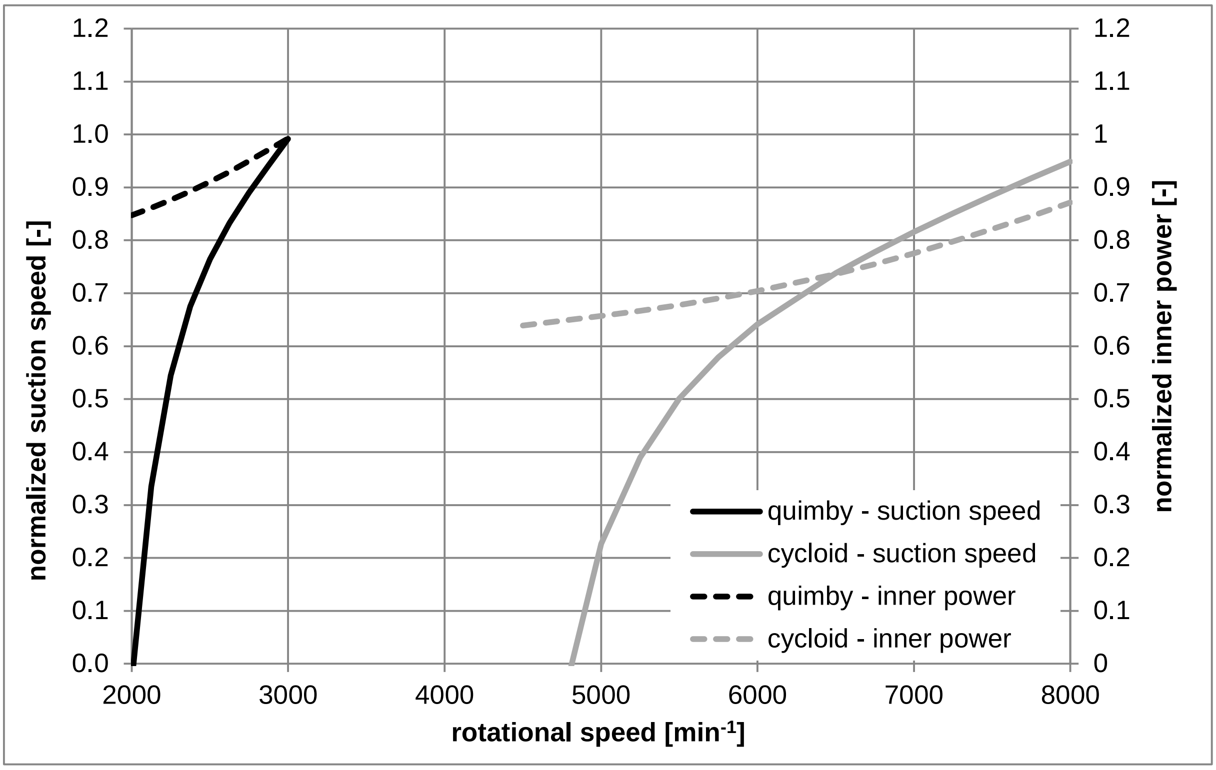

Fig. 9: Suction speed and inner power as a function of the rotational speed for an inlet pressure of 10^2 Pa of the quimby (geometry A) and the cycloid profile (geometry C) normalized by the suction speed of the quimby profile at an inlet pressure of 460 Pa

For this comparison an inlet pressure of 10^2 Pa is chosen. In the case of the quimby profile a higher gradient of the suction speed is observed. This means that a small change in the rotational speed leads to a higher change in the suction speed compared to the cycloid profile. Also important is the amount of energy that is saved by adjusting the rotational speed compared to an adjustment done by using a throttle at the inlet of the pump. Assuming a suction speed of zero in order to maintain a pressure of 10^2 Pa in the vacuum chamber, a saving of power consumption of 22 % is obtained in the case of the cycloid profile. For the quimby profile, only a saving of power consumption of about 15 % is achieved.

13.Conclusions

Within the framework of this article, a comparison of two commonly used profile shapes of screw vacuum pumps has been described. The article points out the basic differences between the quimby and the cycloid profiles with respect to one or two teeth. Comparing geometrical features like the clearance areas of both profile shapes, the advantages of the quimby profile are identified, especially for the blowhole at the front side of the tooth flank. Thermodynamical simulations are carried out with the help of the simulation tool "*KaSim*".

The first comparison of two machines with the cycloid and the quimby profiles at the same rotational speed shows several advantages for the suction speed and the inner power in the case of the quimby profile. Taking advantage of the easy possibility of mechanical balancing of the two-tooth cycloid profile, it is shown that better results of the inner power can be achieved at lower inlet pressures of this profile shape. Simulations indicate a less dependence of the ultimate pressure on the variation of the clearance height in the case of the cycloid profile. Moreover, it is shown, that using a cycloid profile offers more potential in energy savings when operating at variable rotational speeds. As possible future work, experimental validation of the simulation results can be done.

14.References:

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