Improving Chamber Filling in Screw–Type Engines

Verbesserung der Kammerfüllung in Schraubenmotoren

Estimation of different rotor geometries by the use of key values

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Abstract

This article deals with the quantification of two processes which take place during chamber filling and which are important for energy conversion in screw-type engines. These are on the one hand gas leakage through the clearances during the filling process and on the other hand pressure loss at the inlet site of the engine. Key values for analysing these processes are presented. These key values are built by means of geometrical rotor parameters. In the following, geometrical variations of the rotor geometries are estimated via the key values (presented). (The variations of male rotor wrap angle, length / diameter ratio, number of lobes on male and female rotor and the inner volume ratio are calculated for an asymmetrical SRM-B-Profile). Especially for a large inner volume ratio, which is interesting for running the machine as an expander, no optimal geometrical conditions can be found. A geometrical reduction of the operative clearance width, which should reduce the gas leakage too, always causes increased pressure loss at the inlet. Reducing the pressure loss causes increased gas leakage. Therefore two options for modification of the inlet area are investigated. These are on the one hand an inlet control mechanism and on the other hand the use of conical rotors. Based on the calculations for both variants a positive effect on chamber filling may be expected. The inlet control mechanism mainly reduces gas leakage whereas conical rotors have a greater effect on inlet pressure loss.

Zusammenfassung

Dieser Beitrag beschäftigt sich mit der quantifizierten Bewertung von Verlustquellen, die für eine Beurteilung der Energiewandlung in Schraubenmotoren wesentlich sind. Dies sind zum einen die Spaltverluste während der Kammerfüllung und zum anderen die Drosselung am Eintritt in den Motor. Es werden Kennzahlen zur Beurteilung der Verlustquellen vorgestellt, die einen Vergleich unterschiedlicher Rotorgeometrien anhand von Geometriegrößen erlauben. Anschließend erfolgt eine vergleichende Bewertung der Geometrievariationen von Umschlingungswinkel, Längen-/Durchmesserverhältnis, Haupt- und Nebenrotor-Zähnezahl sowie des inneres Volumenverhältnis für das asymmetrische SRM-B-Profil anhand der Kennzahlen.

Speziell bei den für den Motorbetrieb interessanten großen inneren Volumenverhältnissen sind anhand der Kennzahlen keine eindeutigen Optima zu erkennen. Eine Verringerung der Spaltverluste führt immer auch zu einer Vergrößerung der Einlassdrosselung und umgekehrt. Deshalb werden alternativ auch zwei Varianten betrachtet, die von der herkömmlichen Maschinengestaltung abweichen. Dies sind zum einen eine Maschine mit Einlasssteuerscheibe auf dem Hauptrotor und zum anderen ein Rotorpaar mit konischem Hauptrotor. Beide Varianten lassen auf grund der dargestellten Kennzahlen eine Verbesserung der Kammerfüllung gegenüber der üblichen zylindrischen Rotorgeometrie erwarten. Während die Verbesserung bei der Steuerscheibe auf einer Verringerung der Spaltwirkung beruht, ist bei der konischen Variante mit einer deutlichen Verringerung der Einlassdrosselung zu rechnen.

1 Introduction

The presently predicted field of applications for an economical operation of screw-type engines in steam cycles has a middle or low temperature range of less than 350°C and a pressure ration of about 15 – 50. At present the design of screw-type engines is based on the design of compressors. Therefore the pressure ratio at a particular engine stage is mainly determined by the permissible pressure difference at that stage. This means that an expansion in more than one stage has a much higher pressure ratio at the low pressure stage than at the high pressure stage. From this it follows that an engine with single stage expansion as well as the low pressure stage of a multi stage expansion engine needs a high inner volume ratio (vi > 5).

The inner volume ratio of the hot gas screw-type engine (GASSCREW) is normally considerably lower than this. The ideal pressure ratio at the GASSCREW is in a range of about 4 – 8. It is determined by the interaction of compressor and engine and it rises as a result of an increase in the high pressure temperature /3/. Consequently the inner volume ratio can be expected in a range of 2,5 – 4,5.

Former investigations about the filling process of screw-type engines have shown a significantly low rate of chamber filling at the start of expansion in comparison to the delivery rate. This fact seems to be important for the energy conversion of the engine. Two processes can be seen as the main reasons for the mass reduction in the working chamber. These are on the one hand gas leakage through the clearances during the filling process and on the other hand pressure loss at the inlet side of the engine /2/. The gas flows through the clearances at this early stage in the expansion phase are for the most part lost for energy conversion in the engine. Therefore it seems to be essential for an assessment of rotor geometries to take a closer look at these processes.

Both processes are significant in connection with higher inner volume ratios. Hence especially for high inner volume ratios there is a need to improve rotor geometry. But also for the lower inner volume ratios of the GASSCREW it is important to minimize the influence of pressure loss and gas leakage, because it will reduce the power input at the compressor and increase the efficiency of the cycle.

2 Chamber filling

The term chamber filling characterises the mass content of one working chamber during an expansion cycle. It results from the mass flow which streams through the inlet during the filling process and it is continuously changing because of mass flows through the clearances of the chamber.

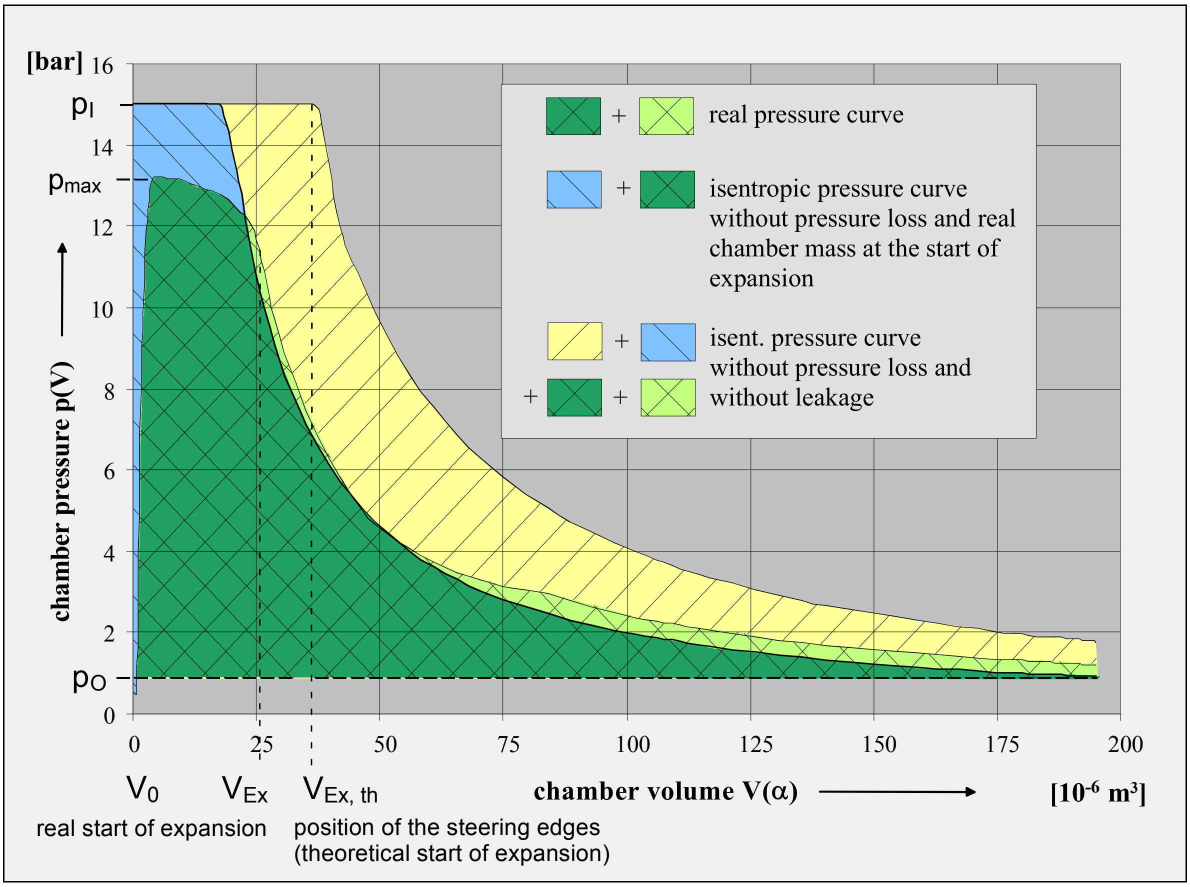

Fig. 1: ***Comparison of isentropic and real pressure curve history in the working area***

Bild 1: ***Vergleich des isentropen mit dem realen Kammerdruckverlauf abhängig vom Kammervolumen***

The law of ideal gases already shows the considerable influence of chamber filling on pressure and temperature in the chamber and therefore for the energy conversion of the expansion process

$$p \cdot V(\alpha_{HR}) = m \cdot R \cdot T \qquad \text{eq. (1).}$$

In order to estimate chamber filling it seems to be useful to take a closer look at the characteristics of a typical expansion process. These can taken from results of former measurements on several engines in different cycles /2/,/4/,/5/. Characteristics which are found in all measurements can be described by the example of a typical pressure curve pictured in ***Fig. 1***.

These are:

- a difference in pressure between the inlet and the chamber
- the real expansion already starts at the chamber volume V_{Ex} 15° up to almost 25° of male rotor rotation before reaching the theoretical point of expansion $V_{Ex,\ th}$
- in the first half of the expansion phase ($V < 75{\cdot}10^{-6}\ m^3$) the real pressure curve decreases faster than the pressure curve of an isentropic expansion
- in the second half ($V > 75{\cdot}10^{-6}\ m^3$) of the expansion phase the isentropic pressure curve decreases faster

For a calculation of the chamber mass during the expansion it is necessary to know the temperature in the chamber. A reliable measurement of the gas temperature in the chamber is difficult, but by making reasonable assumptions about the temperature it is possible to estimate the temperature at two significant points namely at the start and at the end of expansion. To reduce the influence of heat transfer between the gas on the one hand and the surface of the casing and the rotors on the other hand, it is useful to align the temperature of the gas at the inlet with the surface temperature. With this precondition we can make the assumption that the temperature at the real start of expansion is near the inlet temperature. If the pressure in the chamber at the end of expansion is concordant with the pressure at the outlet, the temperatures should be also correspond. With these assumptions about the temperatures the chamber mass can be estimated. We come to the following conclusions that:

- For engines at optimal speed chamber filling at the theoretical start of expansion is normally less than 85 % of the incoming mass flow. For engines without injection of a liquid phase it is even less.
- Filling at the end of expansion is often slightly less than at the start (5 % - 10 %). But this value is highly influenced by the pressure ratio in the engine.

By looking at the pressure curve of the real expansion ***Fig. 1*** with a fast decrease at the start and a slow decrease at the end of expansion, chamber filling during expansion should be no greater than at the start. This means that generally 15 % or more of the incoming mass flow already leaves the chamber through the clearances during the filling process and can hardly contribute to energy conversion /2/.

The pressure loss at the inlet is also caused by a decrease of mass flow, which reduces the area of technical work $\oint p \cdot dV$ (green area in ***Fig. 1***). This is accompanied by an increase of entropy because the part $\int_{V_0}^{V_{Ex}} \Delta p(V) \cdot dV$ (blue and yellow area from V_0 up to V_{Ex} in ***Fig. 1***) of the streaming energy $p_E \cdot (V_{Ex} - V_0)$ will be dissipated.

3 Key values for clearances and pressure loss

3.1 Clearances

The purpose of this study is to discover ways of optimising the geometry of the rotor properties of a screw-type engine in order to improve the filling process. Therefore it is helpful to have a method of comparison for engines with different rotor geometries at the same thermo-dynamical operating conditions by considering only their geometrical properties. This can be achieved by the use of key values. In his studies of the screw compressor Professor Rinder has defined a key value for estimation of the clearances, for which he uses the ratio of the integrated clearances *F(α)* and the chamber volume at the start of compression $V(\alpha_{VB})$ /1/

$$K = \frac{1}{V(\alpha_{VB})} \cdot \int_{\alpha}^{\alpha_{VB}} (F_S(\alpha) + F_B(\alpha)) d\alpha \qquad \text{eq. (2).}$$

The higher the value of this ratio is, the greater the influence of the clearances. According to this model a key value for the screw-type engine can be described, which considers the clearances *b(α)* and the chamber volume only during the filling process $V(\alpha_{vi})$

$$\frac{\dfrac{1}{\alpha_{vi} - \alpha_{Füllbeginn}} \cdot \displaystyle\int_{\alpha_{Füllbeginn}}^{\alpha_{vi}} \left(b_{St}(\alpha) + b_{PE}(\alpha) + b_{Ge}(\alpha) + \frac{A_{Kopf.}(\alpha)}{\bar{z}_f} \right) d\alpha}{V(\alpha_{vi})} \qquad \text{eq. (3).}$$

The clearance in this case refers to the geometrical dimension of the gap vertical to the flow direction (width in ***Fig. 2***). By modification of this key value we can evolve methods for comparison of rotor geometries in different running conditions:

For the assessment of engines with the same expansion volume flow and the same rotational speed we get the dimension less key value

$$\Pi_{Sp,V,n_{HR}} = \frac{\displaystyle\int_{\alpha_{Füllbeginn}}^{\alpha_{vi}} \left(b_{St}(\alpha) + b_{PE}(\alpha) + b_{Ge}(\alpha) + \frac{A_{Kopf.}(\alpha)}{\bar{z}_f} \right) d\alpha}{V(\alpha_{vi}) \cdot 360°} \cdot V_{Ex}^{\;2/3} \qquad \text{eq. (4).}$$

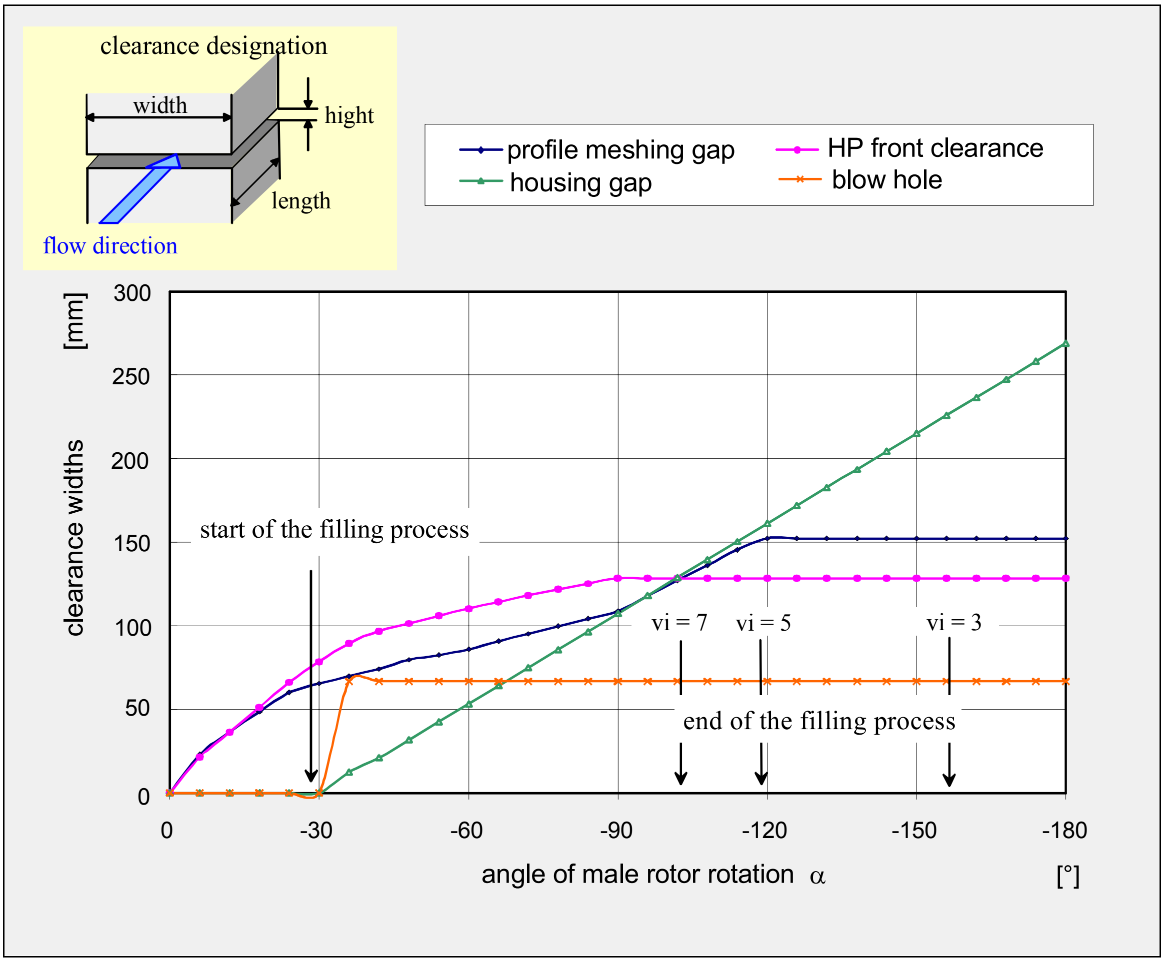

Fig. 2: ***Changes in clearances depending on the angle of male rotor rotation (calculated for an asymmetrical SRM-B-Profile 4 + 6; L/D = 1,5; Φ_{MR} = 300°)***

Bild 2: ***Spaltbreitenverläufe als Funktion des Hauptrotordrehwinkels (berechnet für das Profil: asym. SRM-B 4 + 6; L/D = 1,5; Φ_{HR} = 300°***

And for the estimation of engines with the same expansion volume flow and the same circular velocity the key value can be calculated

$$\Pi_{Sp,V,u_{HR}} = \frac{\int\limits_{\alpha_{Füllbeginn}}^{\alpha_{vi}} \left(b_{St}(\alpha) + b_{PE}(\alpha) + b_{Ge}(\alpha) + \frac{A_{Kopf.}(\alpha)}{\bar{z}_f} \right) d\alpha}{V(\alpha_{vi}) \cdot 360°} \cdot V_H^{\,2/3} \cdot \sqrt{\left(\frac{D_{HR}}{D_{HR11}}\right)^3 \cdot \frac{V_{Ex11}}{V_{Ex}}} \qquad \text{eq. (5).}$$

This equation contains values for an engine which is used as a basic engine for comparison and is marked by the indices 11. The engine has the geometrical properties:

It is also possible to compare engines of different sizes at different speeds but with the same theoretical expansion volume flow. Therefore we have to define another key value

$$\Pi_{Sp,V,\dot{V}} = \frac{\int_{\alpha_{Füllbeginn}}^{\alpha_{vi}} \left(b_{St}(\alpha) + b_{PE}(\alpha) + b_{Ge}(\alpha) + \frac{A_{Kopf.}(\alpha)}{\bar{z}_f} \right) d\alpha}{V_{Kammer}(\alpha_{vi}) \cdot 360°} \cdot \frac{V_{Ex}}{V_{Ex11}^{1/3}}$$

$$\Leftrightarrow \Pi_{Sp,V,\dot{V}} = \frac{1}{360°} \cdot \int_{\alpha_{Füllbeginn}}^{\alpha_{vi}} \left(b_{St}(\alpha) + b_{PE}(\alpha) + b_{Ge}(\alpha) + \frac{A_{Kopf.}(\alpha)}{\bar{z}_f} \right) d\alpha \cdot \frac{v_i(\alpha_{vi}) \cdot z_{HR}}{V_{Ex11}^{1/3}} \qquad \text{eq. (6).}$$

	male rotor	female rotor
number of lobes [-]	$z_{HR11} = 4$	$z_{NR11} = 6$
outer diameter [mm]	$D_{HR11} = 102$	$D_{NR11} = 102$
shaft distance [mm]	$a_{11} = 80$	
rotor length [mm]	$L_{11} = 150$	
L/D [-]	L/D = 1,47	
MR-wrap angle [°]	$\Phi_{HR11} = 300$	
expansion volume [mm³]	$V_{Ex11} = 752000$	

Table 1: ***Data for the basic engine 11***

Tabelle 1: ***Daten der gewählten Einheitsmaschine 11***

If we only vary the size of the engine and leave the geometrical properties unchanged, the key value is only influenced by the clearances. With increasing size the clearances and the key value increase too. This means that a screw-type engine should be kept as small as possible, and the rotational speed should be as high as possible to keep the influence of the clearances small.

3.2 Pressure loss

A calculation of the pressure loss at the inlet of a real engine is very difficult. This can be seen by the model of a transient one-dimensional gas flow /5/. For our intention to find ways of achieving geometrical improvement in chamber filling it seems to be sufficient to simplify the assessment process and to consider only parameters of major influence. Investigations of the filling process have shown that the virtual inlet velocity $c_{E,f}(t) = \frac{\dot{V}_K(t)}{A(t)}$, which is described by Kauder /6/ and Huster /5/, can be a useful parameter /7/. For a simplified model of an ideal impervious chamber we can find a relationship between the delivery rate and the median virtual inlet velocity

$$\lambda'_L \sim \frac{1}{\frac{1}{t_{vi}-t_{Füllbeginn}} \cdot \int_{t_{Füllbeginn}}^{t_{vi}} c_{E,f}(t)\, dt} = \frac{1}{\overline{c}_{E,f}} \qquad \text{eq. (7).}$$

Based on this model, key values for evaluation of the pressure loss at the inlet can be defined in a similar way to the key values for evaluation of clearances.

With the substitution of the expansion time $t = \frac{\alpha_{HR}}{\omega}$ for the ratio of male rotor rotation angle and angular speed

$$\overline{c}_{E,f} = \frac{\dot{V}_K(t)}{A(t)} = \omega \cdot \frac{dV_K(\alpha_{HR})}{d\alpha_{HR}} \cdot \frac{1}{A(\alpha_{HR})} \qquad \text{eq. (8)}$$

we get the dimension less key value for engines with the same expansion volume flow and the same rotational speed

$$\Pi_{E,\lambda_L,n_{HR}} = \frac{V_{Ex}^{\ 1/3}}{\frac{1}{\alpha_{vi}-\alpha_{Füllbeginn}} \cdot \int_{\alpha_{Füllbeginn}}^{\alpha_{vi}} \frac{dV_K(\alpha_{HR})}{d\alpha_{HR}} \cdot \frac{1}{A(\alpha_{HR})}\, d\alpha_{HR}} \qquad \text{eq. (9).}$$

For the estimation of engines with the same expansion volume flow and the same circular velocity the key value has the term

$$\Pi_{E,\lambda_L,u_{HR}} = \frac{V_{Ex}^{\ 1/3} \cdot \sqrt{\left(\frac{D_{HR}}{D_{HR11}}\right)^3 \cdot \frac{V_{Ex11}}{V_{Ex}}}}{\frac{1}{\alpha_{vi}-\alpha_{Füllbeginn}} \cdot \int_{\alpha_{Füllbeginn}}^{\alpha_{vi}} \frac{dV_K(\alpha_{HR})}{d\alpha_{HR}} \cdot \frac{1}{A(\alpha_{HR})}\, d\alpha_{HR}} \qquad \text{eq. (10).}$$

The significance of these key values is diametrically opposed to the key values for the clearances. A high value means a high delivery rate and therefore a small pressure loss. So that in this case high values are positive, while for the clearance key values small values are positive because of small operative clearances.

4 Comparison of the geometrical variations

For an estimation of the geometrical variations, both effects on the filling process which are explained above have to be considered. Therefore in the following the influences of the geometrical variations on the key values for the delivery rate and the clearances are compared. Only the key values for the comparison of engines with the same expansion volume flow and the same rotational speed are depicted, because the results of the comparison of engines with the same circular velocity are quite similar.

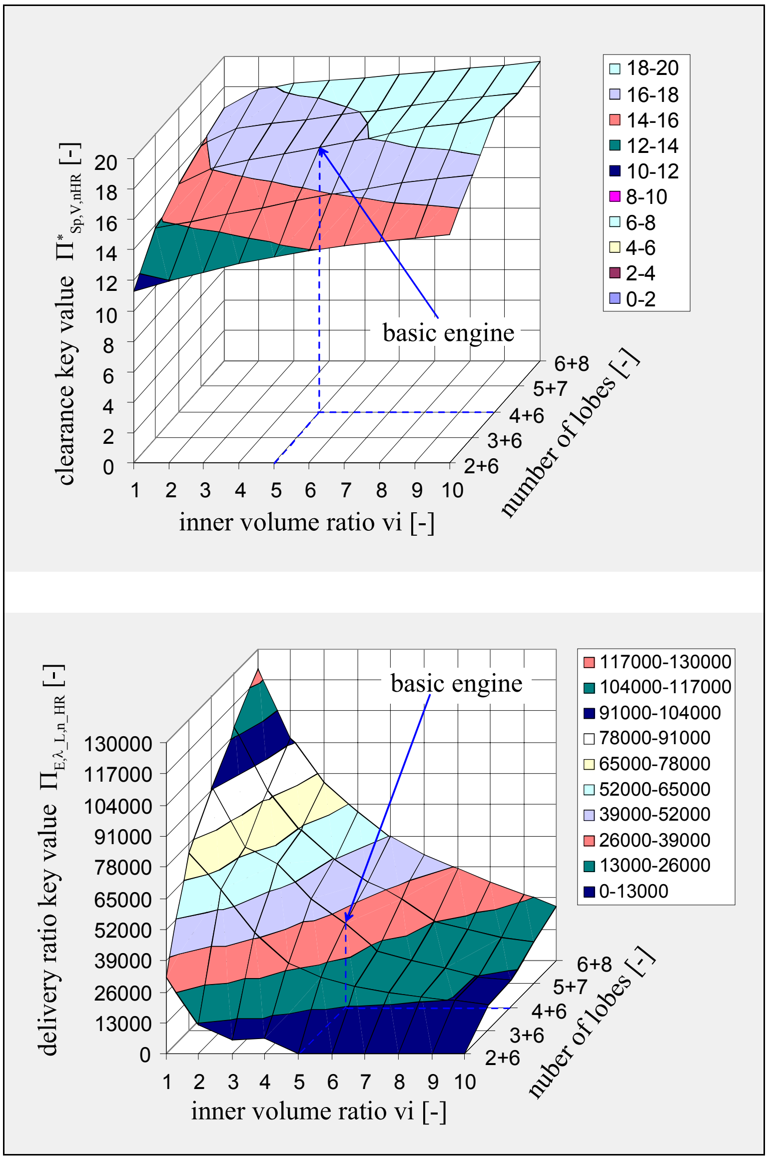

Fig. 3: ***Comparison of engines with equal theoretical volume flow and rotation speed by the use of key values for evaluation of clearances and pressure loss at the inlet (calculated for an asym. SRM profile; L/D = 1,5; Φ_{MR} = 300°)***

Bild 3: ***Vergleich der Kennzahlverläufe für Spaltbreiten und Einlassdrucksenkung von Maschinen mit gleichem theoretischen Expansionsvolumenstrom und gleicher Drehzahl (berechnet für das Profil: asym. SRM; L/D = 1,5; Φ_{HR} = 300°)***

The numerical values which are pictured in the diagrams for geometrical variations are only valid for an inner volume ratio of about *vi* = 5. Depending on the inner volume ratio the influence of clearances and pressure loss on energy conversion changes. With increasing inner volume ratio the influence of pressure loss at the inlet increases too. This can be seen by the decreasing key value of the delivery rate $\Pi_{E,\lambda_L,n_{HR}}$ in ***Fig. 3.*** The increase in pressure loss could be compensated for by increasing the number of lobes, but this has a negative effect on the clearances, as is shown by an increase in the key value $\Pi^*_{Sp,V,n_{HR}}$.

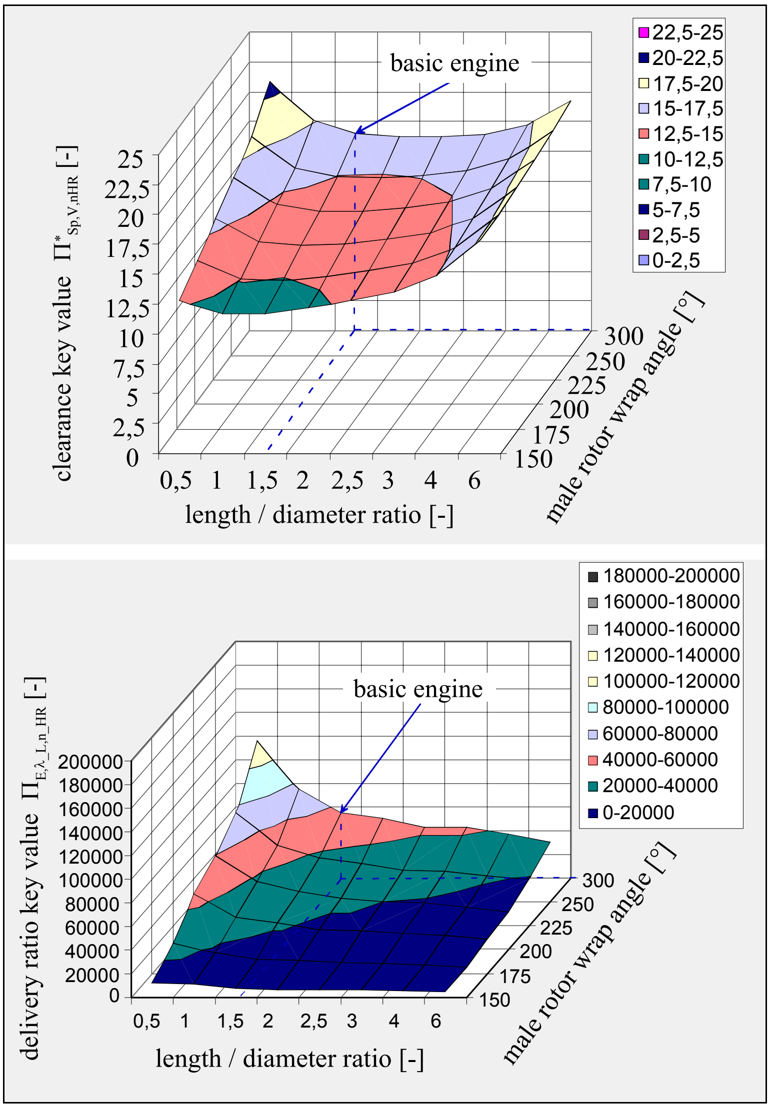

Fig. 4: ***Comparison of engines with equal theoretical volume flow and rotation speed by the use of key values for evaluation of clearances and pressure loss at the inlet (calculated for an asym. SRM profile 4+6; vi = 5)***

Bild 4: ***Vergleich der Kennzahlverläufe für Spaltbreiten und Einlassdrucksenkung von Maschinen mit gleichem theoretischen Fördervolumenstrom und gleicher Drehzahl (berechnet für das Profil: asym. SRM 4+6; vi = 5).***

For an assessment of the geometrical variations we have to consider that the term, basic engine, refers to an engine of average performance with potential for optimisation of both pressure loss and clearances.

The variation of length/diameter ratio and male rotor wrap angle in ***Fig. 4*** produces no corresponding tendencies in the key figures quoted abouve. A decrease in the male rotor wrap angle, which reduces the operating clearances, causes an increase in pressure loss. But the gradient of the delivery rate key figure $\Pi_{E,\lambda_L,n_{HR}}$ is significantly higher than the corresponding clearance key value, so that it seems to be more important to reduce the pressure loss by choosing large male rotor wrap angles in a range of about 225° to 300°.

The variation of the length/diameter ration delivers similar results with an expected optimum in the range of 1 to 2.

A variation of the number of lobes in ***Fig. 5*** shows no great influence on pressure loss for the female rotor, as long as the number of lobes is greater than on the male rotor. But to minimise the influence of the clearances, the number of lobes should be not less than 6. The key values for the number of lobes on the male rotor show opposite tendencies. While a decrease in the number of lobes reduces the operating gap widths, because of an increasing chamber volume, the inlet area decreases too. This causes a significant increase in pressure loss. Therefore in this case the number of lobes on the male rotor should be not less than 4.

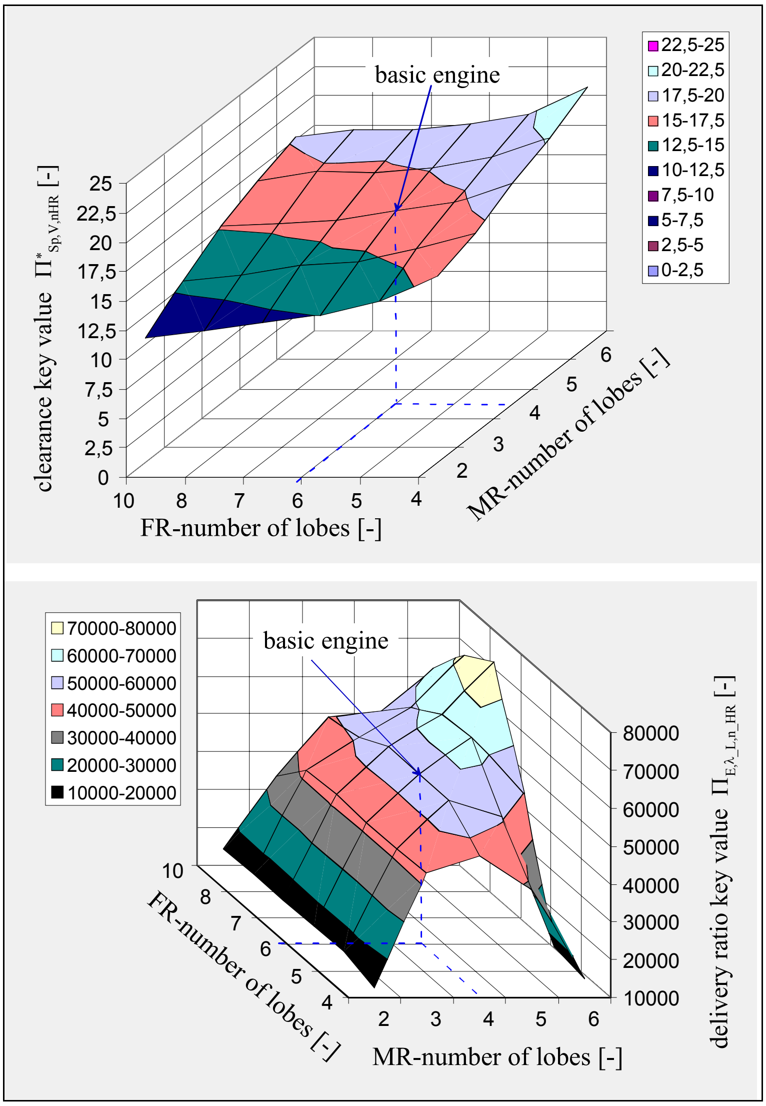

Fig. 5: ***Comparison of engines with equal theoretical volume flow and rotation speed by the use of key values for evaluation of clearances and pressure loss at the inlet (calculated for an asym. SRM profile; Φ_{MR} = 300°; L/D = 1,5; vi = 5)***

Bild 5: ***Vergleich der Kennzahlverläufe für Spaltbreiten und Einlassdrucksenkung von Maschinen mit gleichem theoretischen Expansionsvolumenstrom und gleicher Drehzahl (berechnet für das Profil: asym. SRM; Φ_{HR} = 300°; L/D = 1,5; vi = 5).***

The variations of the male rotor wrap angle and the number of lobes, depicted in ***Fig.*** 6, already contain the results of the former variations. For this reason the number of lobes on the female rotor is never less than 6 and it has always 2 lobes more than the male rotor. The key values for these variations also show opposite tendencies and the gradient of the delivery rate key figure in this case is significantly higher too, so that for best results this value should be considered first. But these variations show only a small potential for optimisation. In a region with a constant delivery rate key value (e. g. *4 + 6 and* φ_{MR} *= 300° to 6 + 8 and* φ_{MR} *= 250°)*, the clearance key value is constant too.

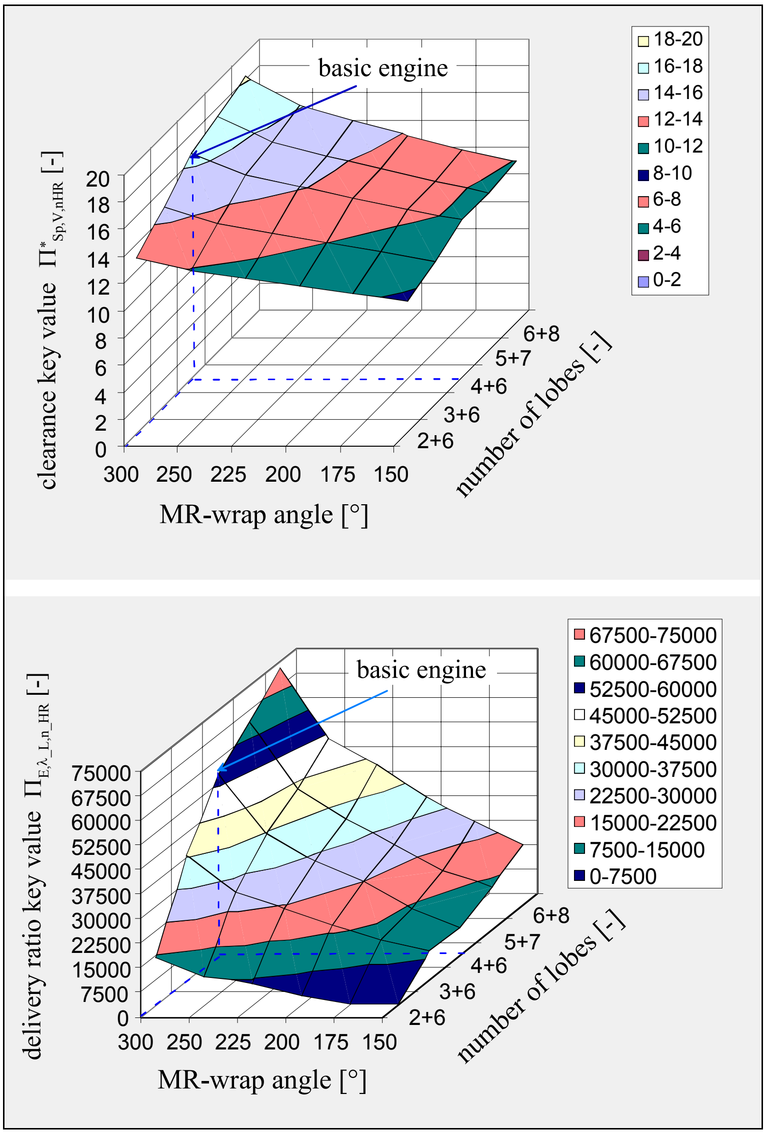

Fig. 6: ***Comparison of engines with equal theoretical volume flow and rotation speed by the use of key values for evaluation of clearances and pressure loss at the inlet (calculated for an asym. SRM profile; L/D = 1,5; vi = 5)***

Bild 6: ***Vergleich der Kennzahlverläufe für Spaltbreiten und Einlassdrucksenkung von Maschinen mit gleichem theoretischen Expansionsvolumenstrom und gleicher Drehzahl (berechnet für das Profil: asym. SRM; L/D = 1,5; vi = 5).***

5 Modification of the inlet area

The usual geometrical variations, which are discussed above, show only a small potential for an improvement in chamber filling, because of their opposing effects on gas leakage and on pressure loss at the inlet. For a geometrical improvement of screw-type engines, therefore, it seem to be necessary to take a look at inlet areas which are different from the usual geometry. Two options are subsequently discussed, a control disc on the male rotor and conical rotors.

5.1 Inlet control disc

The inlet opening of a screw-type engine consists of a fixed opening in the housing and the areas between the rotor lobes, which move periodically along the opening in the housing. Thus the values of these two opening parts can be very different. This applies especially for a large inner volume ratio in connection with a small number of lobes on the male rotor and a small MR-wrap angle. In this case the area between the lobes in comparison to the opening in the housing becomes very small, ***Fig. 7***.

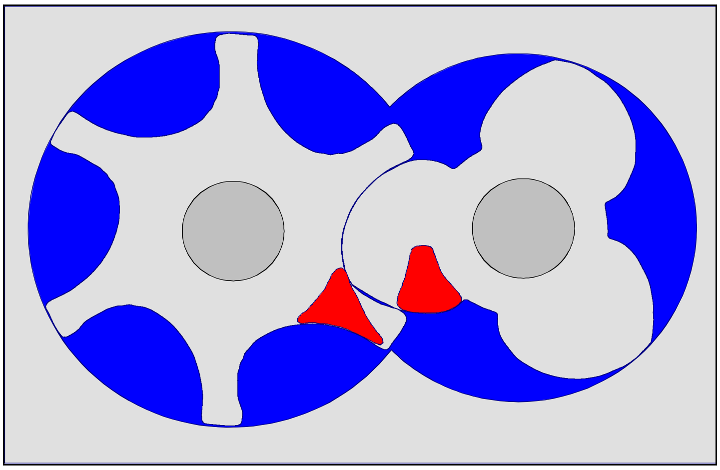

Fig. 7: ***Comparison between the axial inlet area in the casing and the area between the lobes pictured for an asym. SRM profile 3+6; Φ_{MR} = 300° L/D = 1,5; vi = 5***

Bild 7: ***Vergleich der axialen Einlassflächen im Gehäuse und der Zahnlückenflächen dargestellt für ein asym. SRM-Profil 3+6; Φ_{HR} = 300°; L/D = 1,5; vi = 5***

This is the reason for increasing pressure loss at the inlet and with it an increase in entropy. The function of a control disc at the inlet, see ***Fig. 8***, is to modify the values of the opening parts (opening in the housing and area between the lobes) with a view to a minimal entropy production.

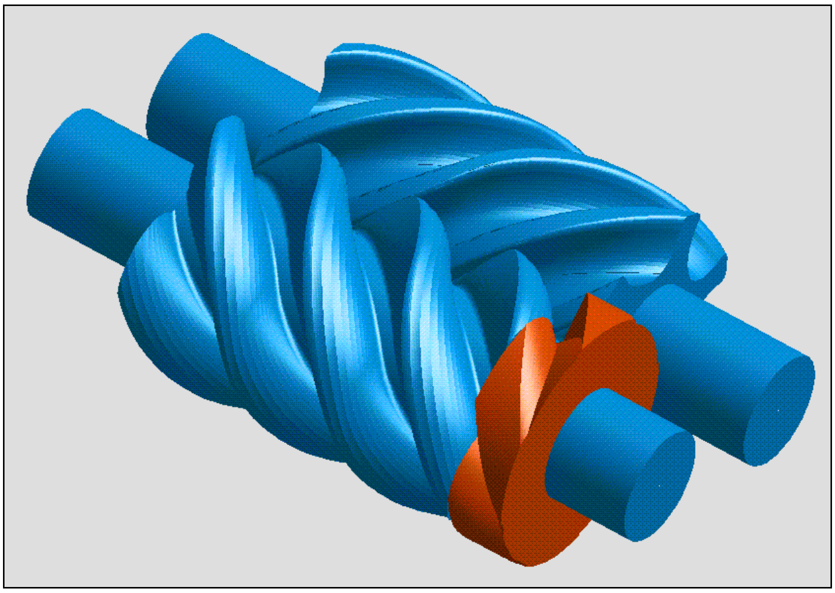

Fig. 8: ***Example for an inlet control disc on the male rotor***

Bild 8: ***Beispiel für eine Einlasssteuerscheibe auf dem Hauptrotor***

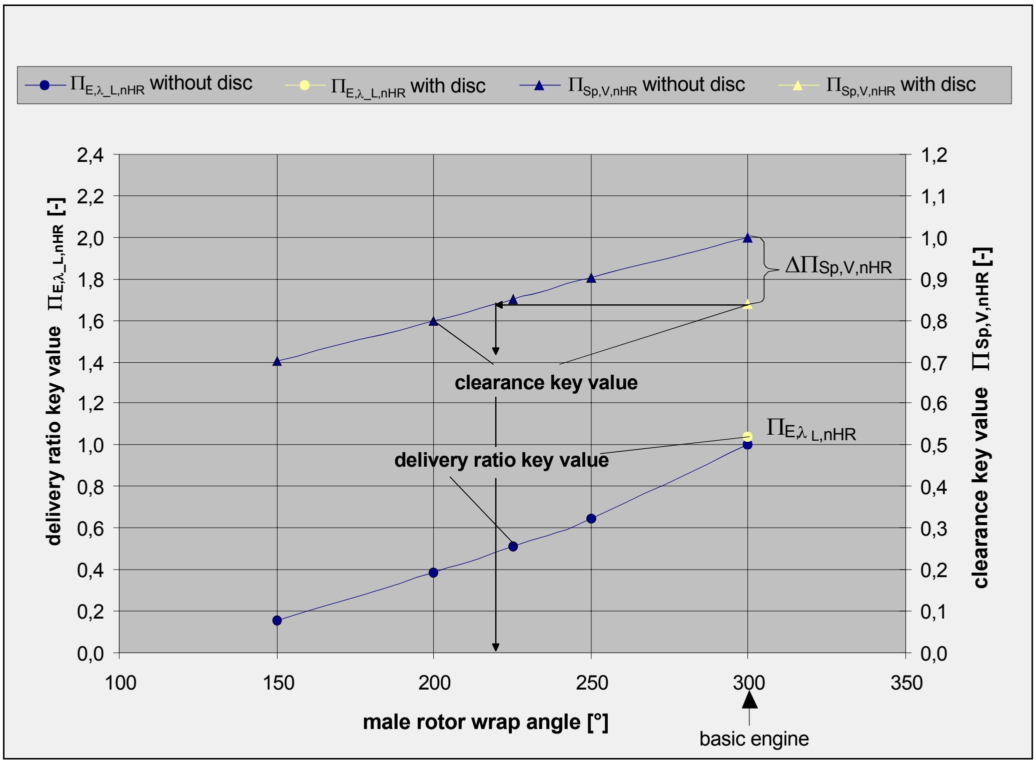

Fig. 9: ***Inlet key value and key value of clearances for rotors with a control disc on the male rotor in comparison to ordinary rotors (pictured for an asym. SRM-Profil 4+6; L/D = 1,5; vi = 5).***

Bild 9: ***Darstellung der Einlass- und Spaltkennzahlen für ein Rotorpaar mit Steuerscheibe auf dem Hauptrotor im Vergleich zu konventionellen Rotoren (dargestellt für ein asym. SRM-Profil 4+6, L/D = 1,5; vi = 5).***

But an assessment of the modification always has to consider the clearance situation too. The consequences can be discussed by comparison with the basic engine in ***Fig. 9***.The delivery rate key figure shows nearly the same value as the basic engine, but the clearance key value is about 15% less.

The reason for this is an increase in the chamber volume because of the additional volume of the control disc. For an equal inner volume ratio therefore the rotation angle of the filling process as well as the time decrease. This has a negative effect on the pressure loss, but a positive one on the clearance situation. An engine without a control disc would need a male rotor wrap angle of about 220° for a comparable clearance situation, but the pressure loss would be much worse.

5.2 Conical rotors

The idea of using conical rotors in comparison to cylindrical ones is to increase the inner volume ratio by keeping the edges of the inlet at the same male rotor rotation angle.

For comparable inner volume ratios we have the options to increase the inlet area by moving the edges for inlet control or to decrease the clearances by reducing the male rotor wrap angle or the number of lobes.

These investigations into conical rotor geometries will not at first be influenced by possibilities of manufacturing considerations.

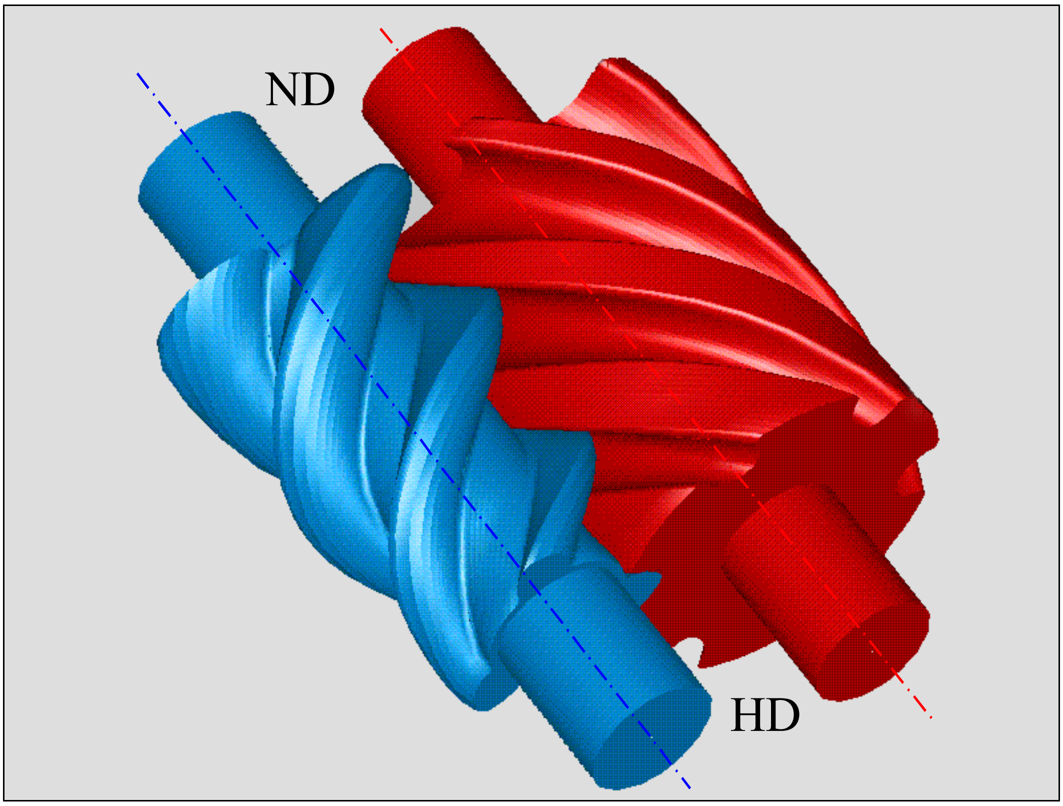

Fig. 10: ***Example for a conical male rotor with cylindrical female rotor and parallel axes based on the asym. SRM 4+6 with*** $\Phi_{MR} = 300°$; $D_{MR} = 80 \div 105$ ***mm.***

Bild 10: ***Beispiel für ein parallelachsiges Rotorpaar mit konischem Hauptrotor und zylindrischem Nebenrotor auf der Basis des asym. SRM 4+6 Profils mit*** $\Phi_{HR} = 300°$; $D_{HR} = 80 \div 105$ ***mm.***

First reflections about the geometry have arrived at a design with parallel axes for male and female rotors and a conical outer diameter for the male rotor, ***Fig. 10***. A comparison with cylindrical geometry is given in ***Fig. 11***.

A comparison at the same MR-wrap angle shows slightly higher values for the clearance key figure (negative effect) but much greater values for the delivery rate key figure (positive effect). The clearance situation is mainly influenced by an increasing male rotor rotation angle at the end of the filling process for the conical rotor, which also means an increase in time. This and the increasing widths of the female rotor lobes on the high pressure side are also the reasons for an improvement of the delivery rate. Therefore the geometries should rather be compared at conditions with the same key figures. A comparison at the same clearance key value in relation to the basic engine shows an MR-wrap angle of about 240° and an increase of the delivery key value of about 40 %. For the same delivery rate key value (MR-rap angle 180°) there is a decrease in the clearance key value of about 12 %.

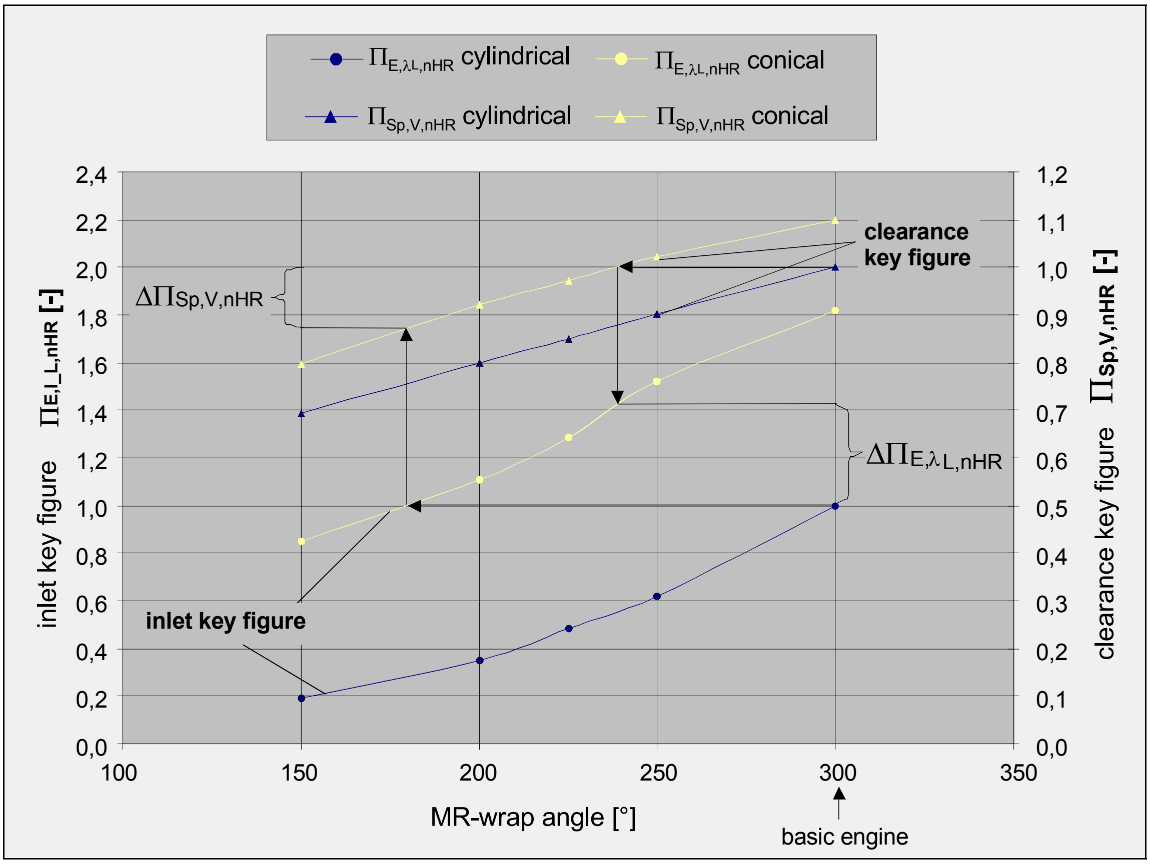

Fig. 11: ***Inlet und clearance key values for the evaluation of rotors with conical male rotor in comparison to cylindrical rotors (calculated for an asym. SRM profile 4+6; Φ_{MR} = 300°; vi = 5).***

Bild 11: ***Spalt- und Einlasskennzahlen für ein Rotorpaar mit konischem Hauptrotor im Vergleich zu zylindrischen Rotoren (berechnet für ein asym. SRM-Profil 4+6; Φ_{HR} = 300°; vi = 5).***

6 Outlook

On the basis of the above calculations for modifications of the inlet area (inlet control disc and conical rotors), a positive effect on chamber filling may be expected. But there are further problems to investigate.

By the use of a control disc an additional volume part is connected to the chamber similar to the exhaust chamber of a piston engine. This is a fixed volume, which has to be filled at the beginning and which can not be completely emptied. The influence of this volume on energy conversion remains to be investigated.

The advantage of conical rotors is mainly produced by an improvement in the delivery rate resulting from an increase in the radial inlet area at the female rotor. It has to be examined if the options for geometrical optimisation which apply in the case of cylindrical rotors are also valid for conical ones or if other factors are involved.

A final point to make is that the manufacturing of conical rotors is difficult at present.

Symbols

Symbol	Dimension	Signification
A, A*	m^2	area
b, b*	m	clearance widths
c	$m\ s^{-1}$	velocity
$\bar{c}$	$m\ s^{-1}$	middle velocity
D	m	outer diameter
L	m	length
m	kg	mass
n	s^{-1}	rotation speed
p	$kg\ m^2\ s^{-2}$	pressure
R	$m^2\ s^{-2}\ T^{-1}$	gas constant
T	T	thermo-dynamical temperature
t	s	time
u	$m\ s^{-1}$	circular velocity
V	m^3	volume
$\dot{V}$	$m^3\ s^{-1}$	volume flow
vi	-	inner volume ratio
z	-	number of lobes
$\bar{z}$	m	middle clearance height
α	(rad)	male rotor rotation angle
Φ	(rad)	male rotor wrap angle
λ_L	-	delivery ratio
Π	-	pressure ratio, dimension less key figure
ω	s^{-1}	angular velocity

Indices

Symbol	Signification
E	inlet
Ex	expansion
f	virtual
Füllbeginn	start of the filling process
Ge	housing gap
HD	high pressure
HR	male rotor
i	inner
K	chamber
Kopf	blow hole
nHR	male rotor speed
ND	low pressure
NR	female rotor
PE	profile meshing gap
St	front clearance
uHR	male rotor circular velocity
V	volume
vi	inner volume ratio
0	start
11	basic engine

7 References

/1/ **Rinder, L.** Schraubenverdichter, In: Springer-Verlag, Wien 1979

/2/ **Zellermann, R.** Optimierung von Schraubenmotoren mit Flüssigkeitseinspritzung, Fortschritt-Berichte VDI Reihe 6, Nr. 357, Dortmund 1996

/3/ **Dosdall, H.** Simulation und Untersuchung des stationären und instationären Betriebs-verhaltens der Heißgasschraubenmaschine, Fortschritt-Berichte VDI Reihe 6 Nr. 365, Dortmund 1996

/4/ **Dreißig, B.** Beitrag zur Auslegung von trockenlaufenden Schraubenmotoren, Dissertation Universität Dortmund 1990

/5/ **Huster, A.** Untersuchung des instationären Füllvorgangs bei Schraubenmotoren. Fortschritt-Berichte VDI Reihe 7 Nr. 336, Dortmund 1998

/6/ **Kauder, K., Kliem, B.** Zweiphasen-Schraubenmotor-Probleme des Füllvorgangs. Schraubenmaschinen Forschungsberichte FEM Nr.6, S. 37-48; Universität Dormund, FG FEM, 1998 ISSN 0945-1870

/7/ **Kauder, K., Fost. C.** Untersuchungen zur Verbesserung des Füllvorgangs von Schraubenmotoren Teil 1. Schraubenmaschinen Forschungsberichte FEM Nr.9, S. 47-62;Universität Dormund, FG FEM, 2001 ISSN 0945-1870