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Michael Beinert
GHH RAND Schraubenkompressoren GmbH - Ingersoll Rand Industrial Technologies, Germany, mbeinert@irco.com

Jan Hauser
GHH RAND Schraubenkompressoren GmbH - Ingersoll Rand Industrial Technologies, Germany, jhauser@irco.com

Non-contacting Seals in Screw Compressors

Michael BEINERT[1]*, Dr. Jan HAUSER[1]

[1]GHH RAND Schraubenkompressoren GmbH, Ingersoll Rand
Oberhausen, Germany
Tel +49 208 699 4193, Fax +49 208 699 4105
e-mail: mbeinert@irco.com

ABSTRACT

Screw compressors are currently the most frequently used compressor type in the field of industrial compressed air production. Because end products are only allowed to come into contact with absolutely clean compressed air in many fields of industry, there is a considerable demand on the market for the production of compressed air that is entirely oil-free. The high acquisition costs of dry-running compressor systems and the disadvantages in energy terms compared to oil-injected designs need to be offset by a long useful life and guaranteed reliability. The compressor's integrated sealing system makes an important contribution here. High peripheral speeds and pressure differences largely rule out the use of simple, contacting seal systems and drive forward continuous optimization in relation to function and production costs.
Based on the calculation of compressible outflows at annular gaps and experimental studies, the sealing performance of various gas choke seals for use in screw compressors will be analyzed and evaluated. Here, the main criteria of pressure reduction and barrier effectiveness, as well as the size and geometric complexity of the seal will be considered.
This paper shows that the flows simulated with the help of a chamber model, using pressure-dependent flow coefficients and overflow factors, allow a good comparison in qualitative terms with the real, measured permeability curve for designing sealing systems for dry-running screw compressors.

1. INTRODUCTION

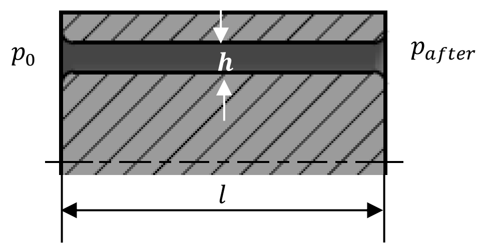
There are two contrasting requirements for shaft seals in screw compressors. On the one hand, they need to seal off the shaft journal as tightly as possible in order to prevent air which has already been compressed from escaping into the environment, but must also prevent bearing oils from permeating into the working chambers. On the other hand, they have to occupy as little space along the shaft journal as possible, in order to thus minimize rotor deflection and facilitate narrow gaps limiting working space, while at the same time ensuring operational reliability. The design of shaft seals thus contributes indirectly towards minimizing flow volumes through gaps between adjacent working chambers and towards improved operating characteristics
In this paper, non-contacting labyrinth seals will be evaluated comparatively for use in screw compressors as part of an investigation. The focus will be on the experimental investigation of sealing performance relative to various geometric parameters. Furthermore, the theoretical background to gas choke seals will be included, calculation procedures will be established, applied and evaluated in terms of the quality of the representation they provide by means of a comparison with the experimental values calculated. Particular focus will be placed on changes to the geometric parameters of a labyrinth seal, which is used in many applications due to its robust properties.

2. FUNDAMENTAL DESIGN OF CONTACT FREE SEALS

Non-contacting seals are used in mechanical engineering where the limitations of contacting seals are exceeded. These limitations particularly include high temperatures and relative speeds which quickly cause excessive material wear on contacting seals. A non-contacting seal is characterised by its so-called sealing gap, which separates the surfaces to be sealed from one another, thereby largely preventing wear. By designing this sealing gap to be as narrow as possible, leakage can be kept within an acceptable range. The group of choke seals used in the screwing machines can be distinguished by separating them into three groups:

- Choke seal
    - Gap seal
    - Labyrinth gap (transparent labyrinth)
    - Labyrinth seal (full labyrinth)

Choke seals are characterised by having as small a flow cross-section as possible, designed in such a way as to have the maximum possible flow resistance in order to limit leakage as far as possible. By the very nature of the principle, absolute leak-proofing is not possible with this non-contacting design. Flow resistance can be achieved using the width and length of the gap (gap seal) or by the flow path having as complex a structure as possible (labyrinth and labyrinth gap).



l = gap length

h = gap height

p_0 = pressure before gap

p_{after} = pressure after gap

Figure 1: Sealing gap

While the gap seal generally refers to a single annular gap between the shaft and the body (Figure 1), a labyrinth is a series of choking points and chambers. A distinction is made here between the so-called real labyrinth and the so-called transparent labyrinth or labyrinth gap. The latter is characterized by the fact that the choking points and chambers do not form an interlocking structure as with a real labyrinth, but are situated in a straight line. The real labyrinth (Figure 2, left) represents a significantly more complex sealing option than the transparent labyrinth, which can be realized for example using a single socket with grooves that are simply pushed over the shaft journal during assembly. (Figure 2, right)

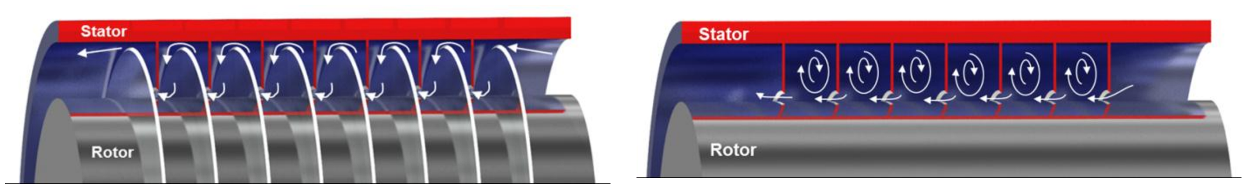


Figure 2: Full labyrinth (left) and transparent labyrinth (right)

Within the screw compressor category of machinery, non-contacting sealing elements are primarily used in dry-running machines, since particularly high rotational speed and temperatures occur while these are in operation. All sealing systems on dry-running machines have in common the separation of pressurized air and bearing oil, which prevents contamination of the pressurized air. While sealing against air is accomplished using a gas choke seal, so-called threaded shaft seals are frequently used for sealing against liquid media such as oil. Viewed in cross-section, these are no different from the labyrinth types specified above, but the grooves run in a helix formation around the shaft axis. When the shaft is rotated, the thread applied produces a feed effect which conveys the leaked oil on the shaft in the axial direction. The direction in which the oil is carried in this depends on the direction of rotation.

Whereas the gap seal obtains its choking effect from the continuous rubbing on the smooth gap walls, the sealing performance of the labyrinth is mainly based on the so-called labyrinth effect. Here, fluid energy is dissipated using varying expansion and turbulence processes. Generally, a labyrinth separates two areas under differing pressures. The medium that is to be constricted flows through the labyrinth from the higher pressure area to the lower pressure area. In the process, it passes through all choking points, over which the gas expands isentropically.
In a full labyrinth, the flow is completely diverted through the labyrinth combs after every expansion process. In a transparent labyrinth, the combs are in a straight line, and there is no geometrically forced rerouting into the chambers. Hence the flow through a transparent labyrinth differs much more markedly from the flow through an ideal labyrinth than that through a full labyrinth. Streamlines occur, flowing directly over the comb tips without being diverted and entering the labyrinth chambers.

Gap seal, full labyrinth, transparent labyrinth, their geometry (gap height, division, comb width, depth) and their flow parameters (relative pressures, peripheral speed, eccentricity)). Some of these factors for the transparent labyrinth and the smooth gap are presented below.

3 Experimental investigation of gas choke seals

3.1 Comparison of different surface geometries and investigation of fundamental relationships

Few research findings are publically available on compressible gap flows in the range of gap heights under investigation. Therefore, the fundamental relationships between permeability, gap height and surface geometry will be investigated in order to derive insights and design recommendations. To accomplish this, the sealing performance of sealing rings with various geometric parameters have been investigated on a test bench. The design of the sealing rings corresponds to the transparent labyrinth with smooth shaft surface shown above. Starting from an existing geometry, the parameters division, comb width, chamber width, groove depth and bridge geometry were varied. Figure 3 shows an sample of the sealing variations investigated.

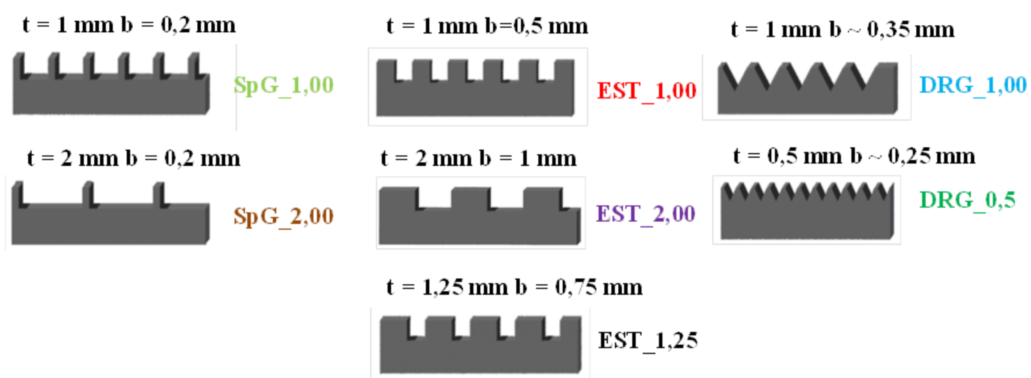


Figure 3: Schematic diagram of sealing variations investigated (sample)

Because the influence of peripheral speed in the operating area on leakage is negligible compared to that of gap geometry, the readings presented below have been determined for a constantly stationary shaft.

Gap height 0.1mm (Figure 4)

With a gap height of 0.1mm, the flow is turbulent across virtually the whole pressure range. The transitional range is enclosed in the diagram by the red lines. The lower and upper limits of the transitional range are set by the Reynolds numbers of 500 and 2000. No significant changes in the shape of the curve, and thus no meaningful change in the resistance coefficient in the sense of a transitional range are established by applying a large volume flow range across a small pressure range. What emerges more clearly, however, is the influence of surface geometry on the leakage flow volume. The smooth ring is significantly more permeable in comparison to the other varieties. The narrow chamber (SpG) varieties are the least permeable. The middle ground is occupied by the labyrinth varieties which, due to their high b/B ratio, correspond more to a coarsely grooved gap (DRG and EST).

Gap height 0.025mm (Figure 5)

Reducing the gap height from 0.1 mm to 0.025 mm significantly reduces the measured leakage flow on all varieties tested, as expected. The maximum permeability of the smooth annular gap is at a comparable level to the variations SpG_1.00 und DRG_0.5 or The middle measurement range is occupied by the finely grooved gap EST_1.00 The volume flow curves of the variations SpG_2.00, EST_2.00 and the coarsely grooved gap EST_1.25 are in the upper measurement range. The volume flow extending across the transitional range is distributed over a greater pressure range compared to the previous gap widths. Clear turning points are manifest in the variations with rugged surfaces throughout this fine transitional range. Here, the gradient of the volume flow curve falls within the range of Re = 500 and then passes through a turning point to assume an almost constant value at higher Reynolds numbers. Only variation SpG_1.00 does not have a pronounced turning point. The smooth gap, by contrast, describes a parabola and does not have any pronounced turning point across the entire pressure gradient.

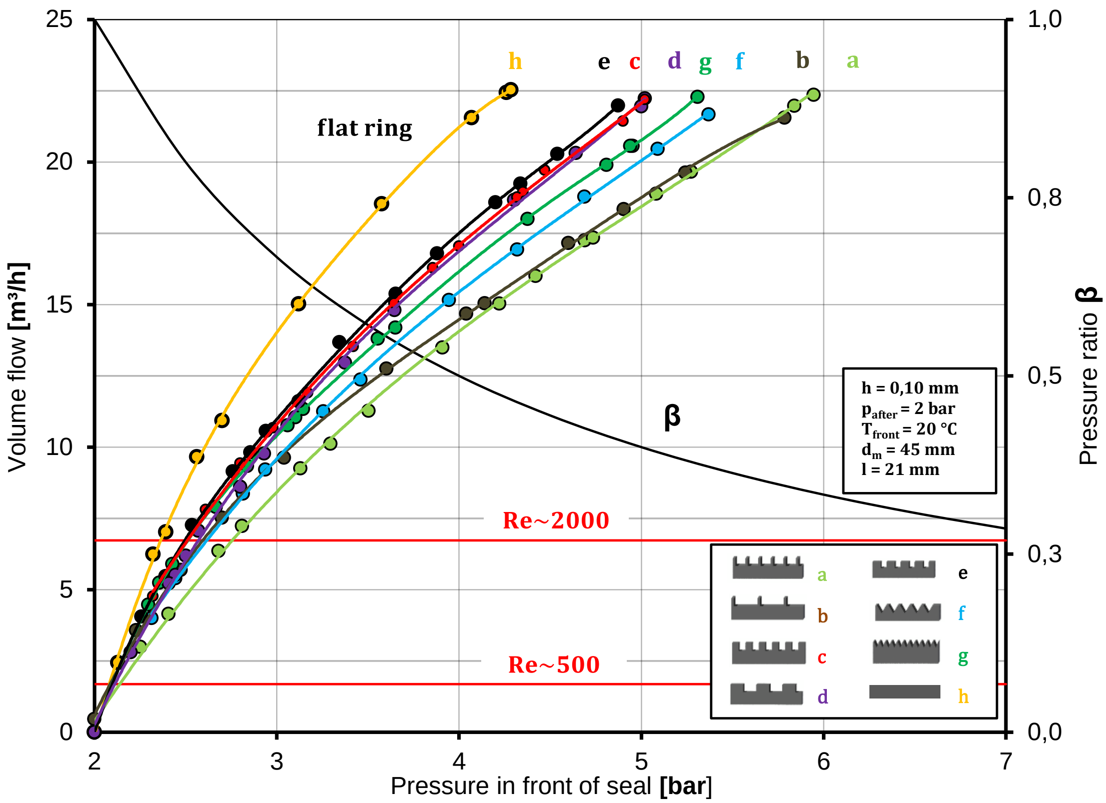


Figure 4: Sealing performance of the variations shown in Figure 3 where gap height = 0.1 mm

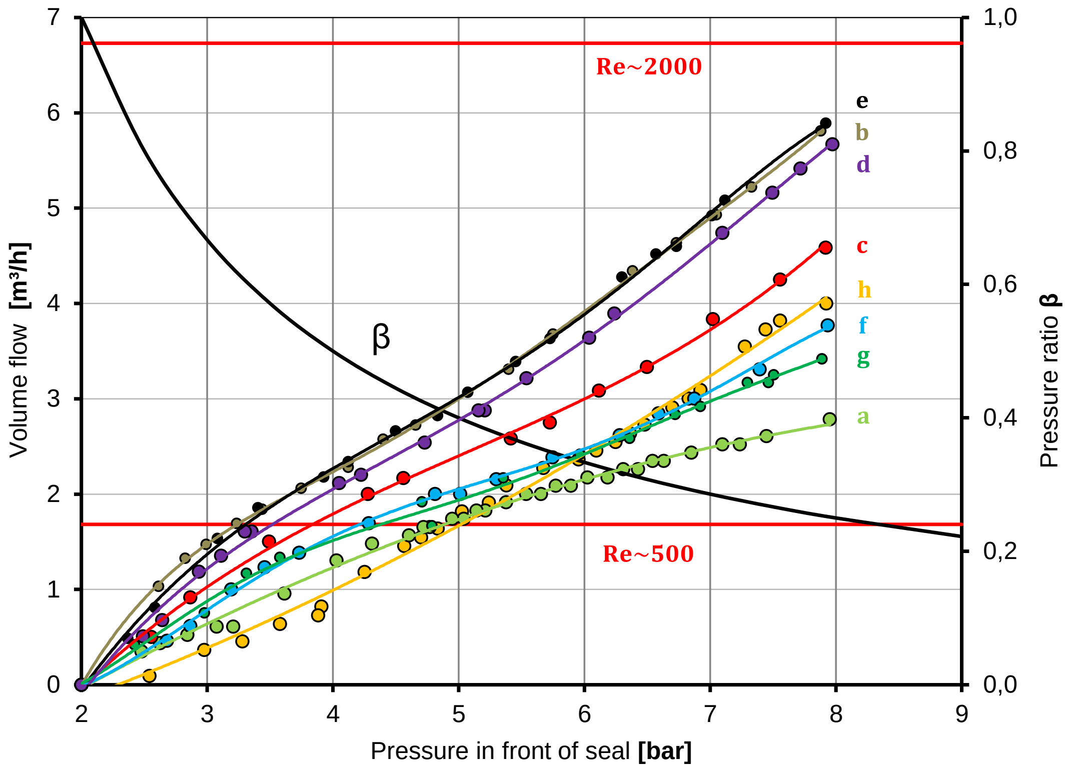


Figure 5: Sealing performance of the variations shown in Figure 3 where gap height = 0.025 mm

Permeability relative to the gap height in different variations

Looking at the diagrams in Figures 4 and 5, the relationship between gap height and the choking effect of the different variations can be clearly discerned. The application of the leak flow curves across the gap heights of selected variations is therefore shown in figure 6. As can be seen in the diagrams, the gradients of the volume flow curves increase disproportionately for all variations, with this correlation being particularly pronounced for the smooth gap. This is explained by the fact that, as the gap height decreases, the flow through the smooth gap increasingly shifts to the laminar range across the whole pressure range measured, ending up continuously laminar at a gap height of 0.025 mm for the smooth ring. The resistance coefficient increases as the Reynolds number decreases, with this correlation being particularly pronounced in the laminar range. The measured values shown illustrate clearly how, starting from a smooth annular gap, the leakage volume can be significantly reduced by the geometric structure.

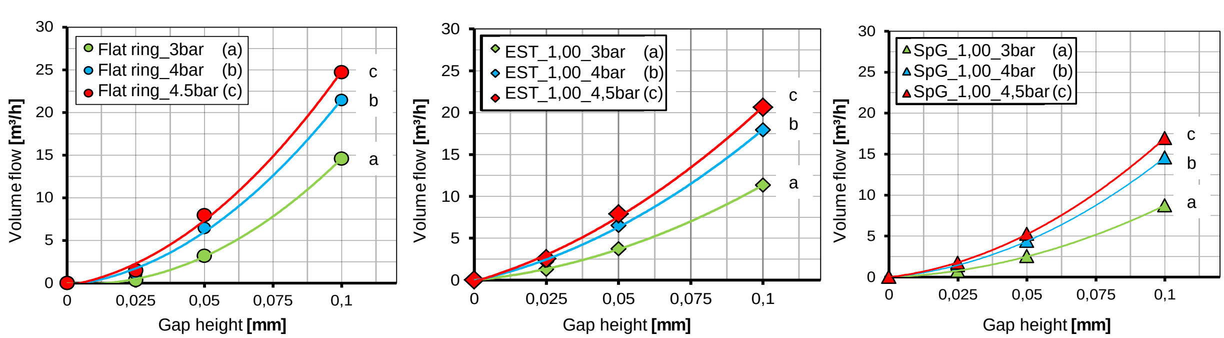


Figure 6: Effect of gap height on permeability for different geometric variations

4 Experimental investigation of gas choke seals

4.1 Model for calculating leakage through labyrinth seals and annular gaps

An iterative calculation procedure based on the modelling of the labyrinth seal as a transient system is used for the calculating flows for labyrinth seals. While the volume flow can be assumed to be known with a stationary model, an initial pressure distribution is assumed for the transient chamber model, then the procedure is simulated until convergence. The entire system thus progresses to a natural state of equilibrium.

Firstly, the labyrinth seal previously considered in the experiment is transferred to a chamber model. For this, the seal is considered as a series of n annular chamber-shaped control volumes connected to one another by n + 1 choking points The system parameters of the control volumes are assumed to be adiabatic for simplicity, whereby the state variables of the ideal gas are taken as homogenous across the spatial extent of the chamber. Furthermore, a uniformly constant temperature distribution is assumed. Hence the temperature is the same in all chambers. This assumption only applies to an ideal labyrinth, but is also justified as a good approximation in real transparent labyrinths, as experiments have shown. The complex fluid mechanical processes in the turbulence chambers are shown in simplified form. That is to say that the incomplete turbulence of the kinetic energy that causes prior speed and hence increased volume flow into the choking points is taken into account using an overflow coefficient. The dissipation and flow constriction at the choking points itself is taken into account by a flow coefficient. The overflow factors used are the factors of Neumann, Hodkinson and Vermes as well as a modified model by Zimmermann. Figure 7 provides a graphic representation of the transfer of a labyrinth seal to a multi-chamber model. This shows the transient state in which the volume flows have not yet assumed the same value through the individual choking points. Rather, in addition to the volume flows m_{one} entering and leaving the chambers, the volume flows $\Delta\dot{m}$ effectively flowing into and out of these volumes also occur, which arise from the mass balance around the annular chamber.

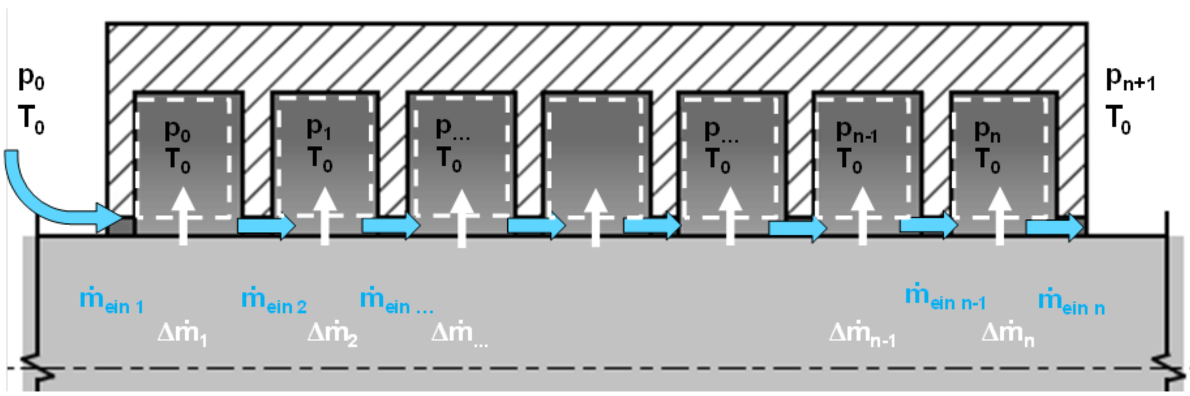


Figure 7: Labyrinth seal as chamber model in transient state

The calculation of annular gap flows is based on a fundamental iterative method for calculating compressible tube flows, which has been modified for the application being analyzed. Due to the influence of factors such as changes in viscosity, acceleration work and flow formation, this type of flow cannot be calculated using a closed formula. The gap is divided into multiple segments for the iteration. This makes it possible to record the effect of temperature change on viscosity and hence the Reynolds number and resistance coefficient. Moreover, it enables the progression of the state variables to be followed over the height of the gap. The segments are passed through accordingly one after the other during iteration.

5. Verification of calculation procedure

5.1 Verification of algorithm for calculating flows in annular gaps

Below, the results of the calculation model will be compared with the measured permeability curves in order to be able to comment on the quality of the representation.

Although adherence to the critical pressure ratio is also demonstrated when verifying the model, verification of the model for calculating annular gap flow will be limited to a comparison of the measured and calculated permeability curves. The measured and calculated permeability curves for three different gap heights (or l/h ratios) can be seen in Figure 8. As well as the iterative calculation procedure, an isothermal calculation procedure has also been used without taking accelerating forces into account. For each of the three measured gap heights, the iterative calculation procedure has been carried out once while taking into account dissipation during the initial adiabatic transfer, and once without. In the case of a super-isentropic initial adiabatic transfer, an initial hydraulic loss of 0.5 has been used, corresponding to a sharp edged inlet.

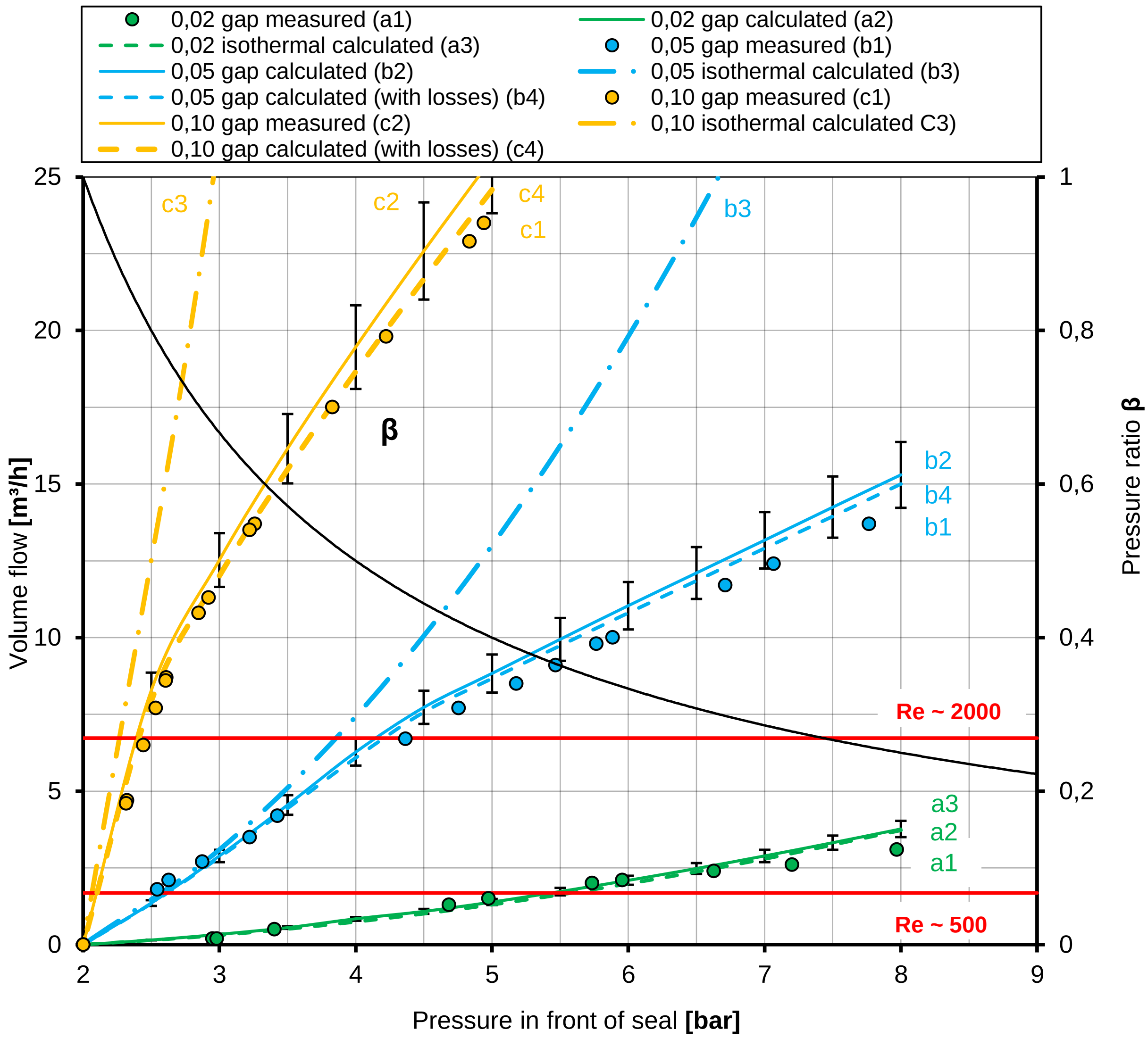


Figure 8: Comparison of calculated permeability curve with a measured permeability curve for the smooth annular gap. (Gap height: 0.148 mm)

Looking at the permeability curve for the smallest gap width of 0.023 mm, it is apparent that both the iterative and isothermic calculation procedures provide very good results. Both curves describe the typical parabola that is characteristic of a very long and narrow gap. The situation changes if the gap width is increased to 0.048 mm. Here, the isothermic calculation procedure corresponds to the measured values only up to a pressure of approx. 4 bar. All three curves approximately describe a parabola up to this point. However, when the pressure is further increased, both the measured permeability curve and the permeability curve resulting from the iterative calculation procedure pass through a kink, after which their course becomes more or less linear. From this pressure level upwards, the progression of the permeability curve resulting from the iterative calculation procedure can be seen to diverge sharply both quantitatively and qualitatively. The assumption of a parabolic speed distribution as the basis for the isothermic calculation procedure is no longer justified from a Reynolds number of around 2,000 upwards, due to the occurrence of turbulence.
The divergence between calculation and measurement is less than 15 percent throughout, which represents a good result, considering the application. In view of the measured results, the isothermic calculation of gap flows, neglecting the initial adiabatic transfers and accelerating forces, is revealed to be a valuable instrument for quickly calculating flows for relatively long and narrow gaps.

5.2 Verification of calculation of labyrinth seal flows using the chamber model

Verifying the program with regards to its suitability for calculating leakage volume flows entails the problem of assuming a flow coefficient and an overflow factor. In the classical labyrinth theory, both coefficients are constants by a factor of which the ideal labyrinth volume flows are expanded and adjusted to reality. Therefore, the leakage volume flows will initially be calculated assuming an ideal labyrinth consisting of a series connection of ideal nozzles. The permeability curves will then be adjusted to the measured values by a constant factor representing both the overflow effect and flow coefficient. The measured and calculated permeability curves after adjustment by constant factors are shown for a gap width of 0.06 mm in Figure 9. As can be seen in the diagram, the gradients of the calculated and measured curves diverge from one another over the entire pressure range. For smaller divisions in particular, it is striking that the gradient of the calculated volume flow curve is comparatively large in the lower pressure range compared to the measured curve, while it is too low at higher pressures.

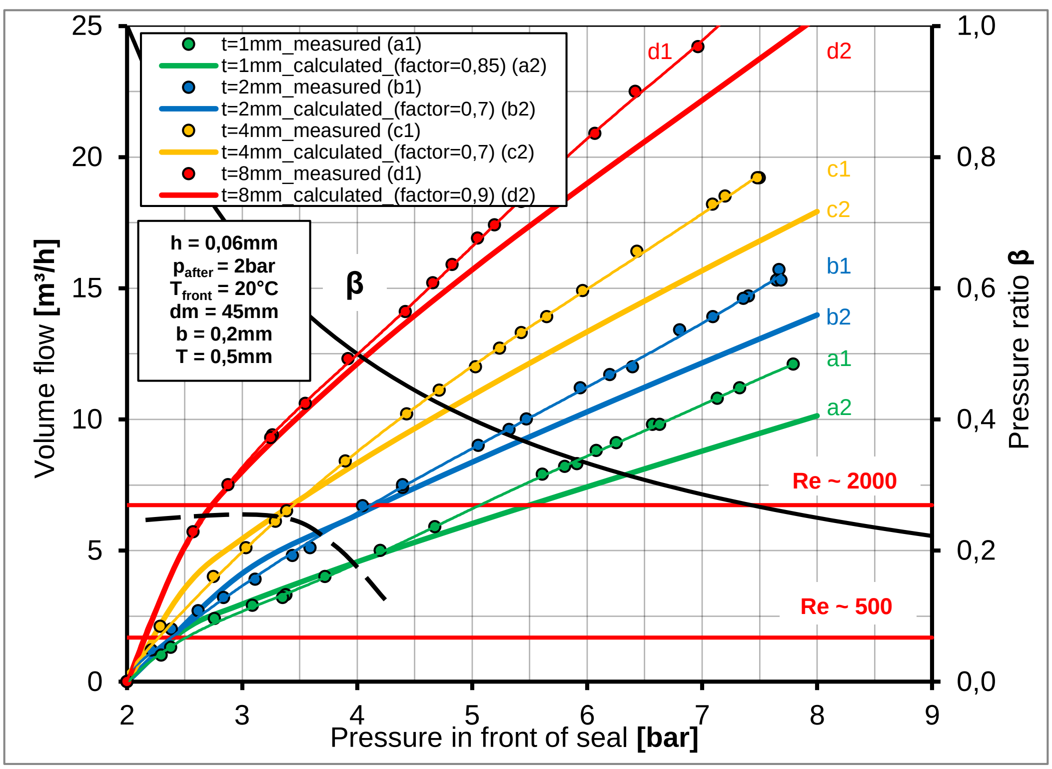


Figure 9: Comparison of calculated and measured permeability curves (h = 0.06 mm)

These divergences can be explained by considering measured flow coefficients for a single labyrinth comb relative to the pressure ratio. It can be determined that the flow coefficient of a labyrinth comb remains constant across a large pressure range, but falls rapidly from a certain pressure ratio upwards. This behavior provides an explanation, albeit an incomplete one, for the qualitative divergence of the calculated and measured permeability curves. If the pressure before the labyrinth is relatively low (high general pressure ratio), the pressure ratios of the individual choking points will be correspondingly high, yielding an accordingly low flow coefficient. As the general pressure ratio falls, however, the local pressure ratio also falls across the choking points and the flow coefficient increases. By using a flow coefficient that is dependent on the pressure ratio, the course of the calculated volume flow curve can be adjusted to the progression of the measurements.

6 Summary and outlook

The present paper provides a sample of a fundamental analysis of the sealing performance of gas choke cylinders relative to geometric parameters for dry-running screw compressors.
Starting from the existing sealing systems, the geometric parameters of individual elements were changed and a practical investigation into their effect on leakage characteristics undertaken. In the process, the relationships known from the relevant literature were verified on the one hand, and useful relationships brought about by slight geometric changes were discussed on the other. Building on the fundamental insights, algorithms were established to calculate leakage through the relevant seal types. In the case of gap seals, a procedure was used for calculating compressible outflows, which was modified for use with annular gap flows. The procedure achieves good consistency with the measured results, and proves to be suitable for calculating leakage volume flows through annular gaps.

The simulation of the flows through labyrinth seals using a simplified chamber model proves suitable to a limited extent for calculating leakage volume flows in advance. By using pressure-dependent flow coefficients and using overflow factors, qualitative consistency between the calculated and measured permeability curves was achieved. The qualitative divergence was within an acceptable range, given the application. Moreover, the combination of two approaches for overflow factors enabled a good prediction of the permeability of seal prototypes. The representation of a labyrinth flow through a chamber model assuming homogenous fluid properties is a significant simplification, considering the complex fluid mechanical processes. However, this representation allows the leakage characteristics to be used, qualitatively and with sufficient precision, for estimating sealing elements and as a basis for more detailed models. The long-term aim is to use the insights obtained constructively for designing sealing systems in screw compressors and to expand the existing guidelines to include these.

NOMENCLATURE

b	Chamber width	(mm)
d_m	Mean diameter	(mm)
h	Gap height:	(mm)
l	Gap length	(mm)
$\dot{m}_{ein}$	flow volume	(kg/s)
n	Sequential numbering	(-)
p	Pressure	(bar)
p_0	Pressure before gap	(bar)
p_{after}	pressure after gap	(bar)
T	Temperature	(°C)
T	Chamber depth	(mm)
T_0	Stagnation temperature	(°C)
t	Division	(mm)
β	Pressure ratio	(-)
ζ	Loss coefficient	(-)

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