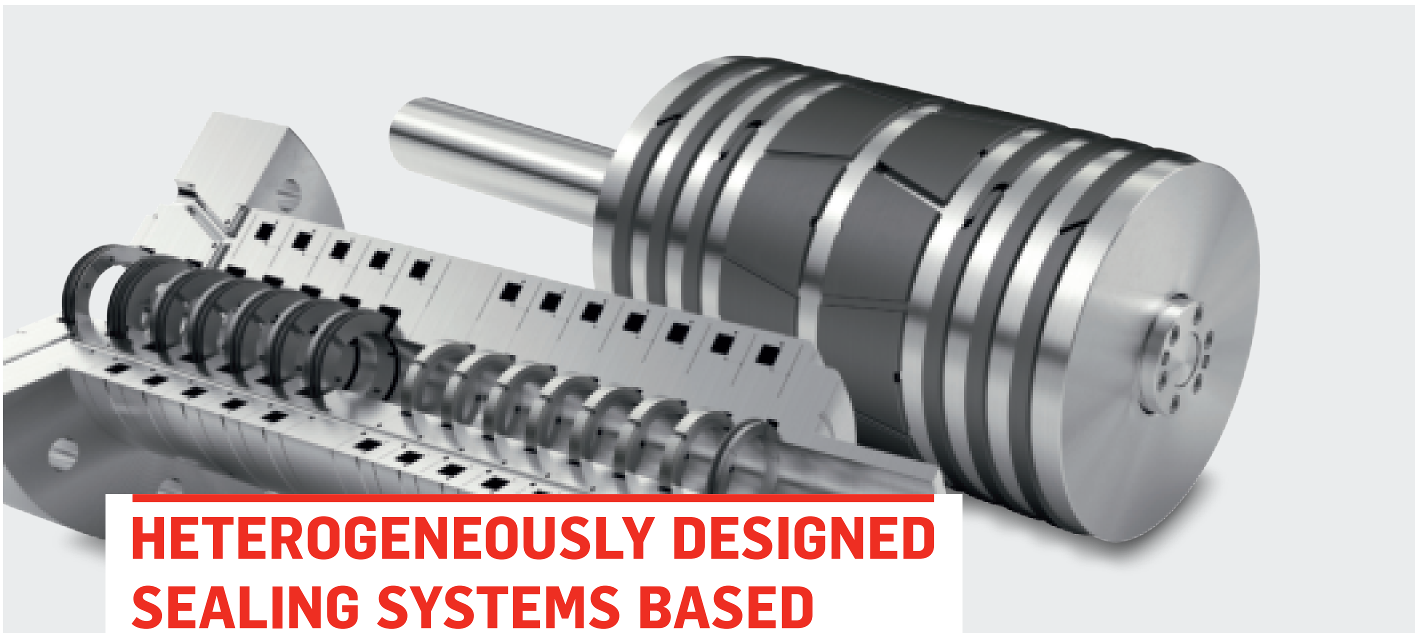


HETEROGENEOUSLY DESIGNED SEALING SYSTEMS BASED ON THE REDURA® PRINCIPLE

Dr. Norbert Feistel

Compressors for a Lifetime™

Dr. Norbert Feistel received his degree in Mechanical Engineering (Dipl.-Ing.) from the University of Karlsruhe, Germany, in 1987. In 1988 N. Feistel joined the R&D Group of Burckhardt Compression in Winterthur. After approximately two years, in which N. Feistel's activities concentrated mainly on the laby-rinth piston compressors, his responsibilities are now for the development of oil-free sealing systems. In 2002 he gained a PhD at the Uni-versity Erlangen-Nuremberg, Germany, with a thesis on the operational behavior of dry-running sealing systems in crosshead compressors.

Sealing systems are frequently subjected to both static and dynamic pressure components. Comprehensive test bed trials using various designs of sealing elements have shown that the two pressure components have significantly different effects on the sealing elements and on the operating behaviour of the entire sealing system. Consequently, heterogeneous sealing systems should be used, comprising at least two different designs, each optimised for the stress from the respective pressure component. Redura® Sealing Systems meet this requirement and, through their optimised design and combination, significantly enhance performance, reliability and mean time between overhaul compared to conventional sealing systems.

CHAPTER 1

INTRODUCTION

Crosshead compressors offer a large number of design alternatives for the implementation of single or multiple compression stages along a crankshaft. This leads to a broad spectrum of pressure loads for the associated sealing systems. As a result – depending on the arrangement of compression chambers and on the respective compression ratio – completely different composed pressure differences emerge. These range from purely dynamic pressure, to completely different combinations of dynamic and static pressure components, to an entirely static pressure difference. The operating behaviour of the sealing system is correspondingly varied. Design engineers are thus faced with the challenge of designing sealing systems optimised for any one of the various pressure combinations. A large selection of sealing element designs, and an even larger number of sealing element materials composed of extremely varied plastic composites, are available to meet this challenge. Precise knowledge of the behaviour of the individual sealing elements under the influence of such disparate load parameters is necessary to design reliable and high-performance sealing systems.

CHAPTER 2

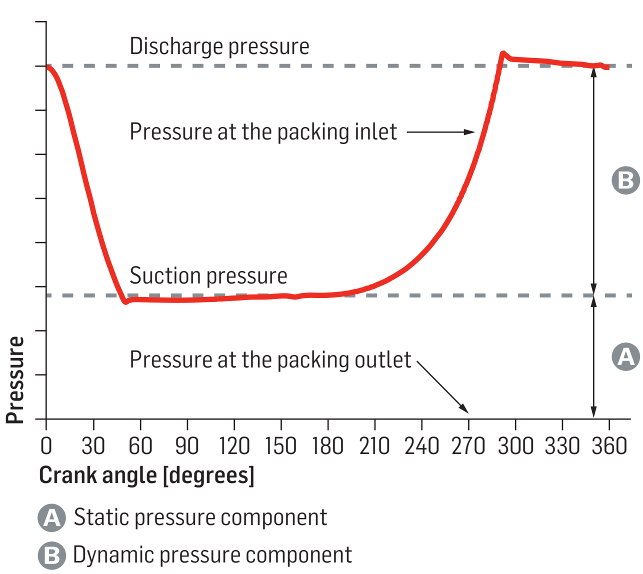
EVALUATING THE PRESSURE LOAD ON SEALING SYSTEMS

In order to assess the pressure load acting on a sealing system, it is usual to consider the total pressure difference present over the complete sealing system. Frequently, hardly any attention is paid in this process to the different dynamic and static pressure components. When designing a sealing system, this can have fatal consequences, since the two pressure components differ considerably in terms of their influence on the sealing elements and on the operating behaviour of the entire sealing system.

The dynamic pressure component is a characteristic feature of the compression process running in a reciprocating compressor. This is the difference between the compression pressure and suction pressure, changing between a maximum value and zero in the course of each crank rotation. The important parameter is the maximum value, which is defined as the difference between the final pressure and the suction pressure of the relevant compression state.

The static pressure component is the difference between the suction pressure of the compression stage under consideration and the constant pressure after the last sealing element. This can be the ambient pressure, or may be the suction pressure of the same or of a lower compression stage. The static pressure component has a constant effect during a rotation of the crank. Fig. 1 illustrates the pressure components defined in this way, taking the pressure on a piston rod sealing system (packing) as an example. **Fig. 1**

Fig. 1
Breakdown of the pressure difference into a dynamic and a static pressure component



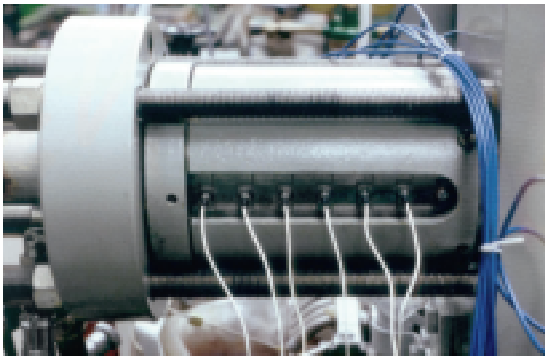
The separation of the pressure load into a dynamic and a static pressure component may sound academic, but it has a real background. This emerges from the question of the distribution of the pressure difference over a given number of gastight sealing elements. "Gastight" refers here to sealing rings which provide complete (i.e. axial and radial) covering of all ring joints. Designs of this sort are typically used in the packings.

To answer this question, Burckhardt Compression has been carrying out extensive trials using a specially equipped experimental compressor since the beginning of the 1990s[1]. Its tailrod packing allows the pressures and temperatures in the individual chambers to be measured, as well as a loss-free measurement of the packing leakage. **Fig. 2**

Over more than 20 years of experimental operation, innumerable packing configurations consisting of highly varied designs and materials have been extensively tested. They have shown that when sealing systems are in their new condition, the usual contact-free throttle rings only make a very small contribution to sealing the dynamic pressure component, so that most of it is almost entirely absorbed by the first gastight sealing element arranged at the immediate vicinity of the compression chamber. In contrast, the position of the static pressure components when a new sealing system is first brought into operation depends on a large number of factors such as the design, differences in manufacturing quality, or deviations from the optimum fitting position. This means that in the case of sealing elements with a complete covering of all ring joints, it is possible that both the dynamic and static pressure components only act on the first frictional sealing element, causing it to be overloaded. Preferably the last sealing element at the packing outlet is loaded by the static pressure component. **Fig. 2**

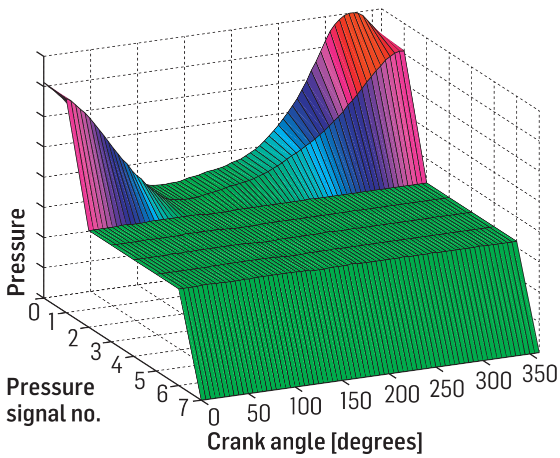
With progressive wear, the dynamic pressure component then shifts further into the packing. Throttling leads to its uneven distribution over several sealing elements. The static pressure component moves, in whole or in part, back and forth between the other sealing elements, and the distribution over a number of sealing elements does not remain stable. The preferred position for the static pressure component, however, is the last sealing element. The division of the two pressure components at the two ends of the sealing system is thus found to be a typical feature. This property allows the different effects of the two pressure components at the respectively loaded sealing elements to be considered separately.

Fig. 2
Measuring the pressure in the chambers of an experimental packing in order to determine the loads on the individual sealing elements

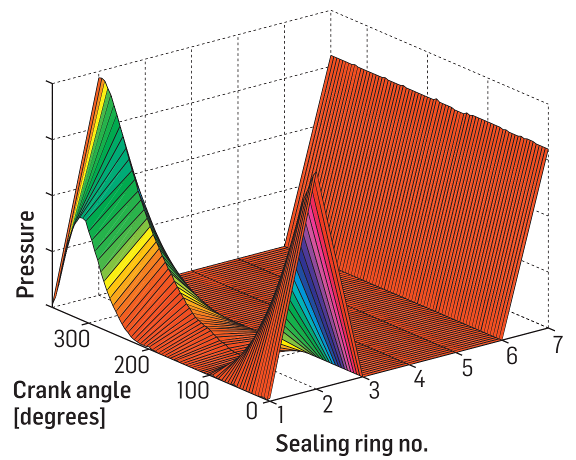


Measurement on the test packing

Cross section of a typical packing



Measured pressure curves



Calculated pressure differences

CHAPTER 3

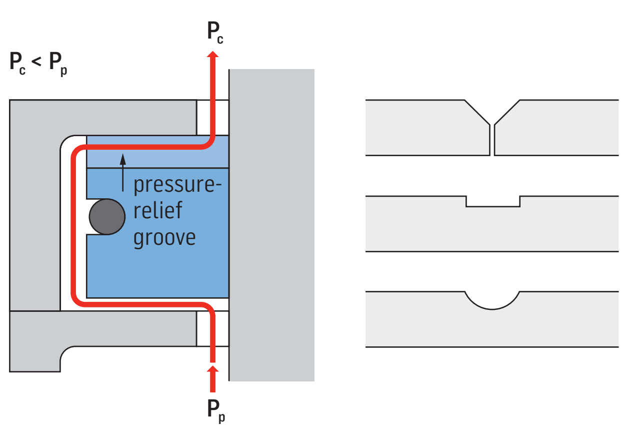
DIFFERENT EFFECTS OF THE DYNAMIC AND STATIC PRESSURE COMPONENTS

Many observations on the operating behaviour of sealing systems, taken in practice or found in the analysis of the state of the sealing elements employed, that may appear contradictory are more easily understood if we look more closely at the composition of the respective pressure load. The different effects of the dynamic and static pressure components on the operating behaviour of a sealing system can be illustrated very clearly by the leakage and by the wear to which the individual sealing elements subject.

3.1 EFFECT ON THE LEAKAGE

Due to the leakage between the individual sealing elements of a sealing system, pressure differences in the direction of the compression chamber arise when subject to a pressure curve that changes over time. During the compression phase, the pressure in the chambers between the sealing elements directly adjacent to the compression chamber thus rises to levels above that of the suction pressure and can – depending on how worn the sealing elements are – almost reach the final pressure. If the pressure in the cylinder then drops again toward the suction pressure, the result is that pressure is released back into the compression chamber. The return flow of the gas temporarily stored in the sealing element chambers can be considerably improved by what are known as pressure-relief or return flow grooves, so that in this phase the sealing elements have almost no sealing effect, and therefore also do not demonstrate any wear. Pressure-relief grooves are usually arranged directly at the sealing elements, and are implemented in the form of round or rectangular radial channels on the side facing the compression chamber, or in the vicinity of the segment ends in the shape of triangular chamfers. If, due to the particularities of the design or to the available space, the return flow grooves cannot be integrated into the sealing element, it is alternatively possible for them to be fabricated directly in the sealing element chambers. **Fig. 3**

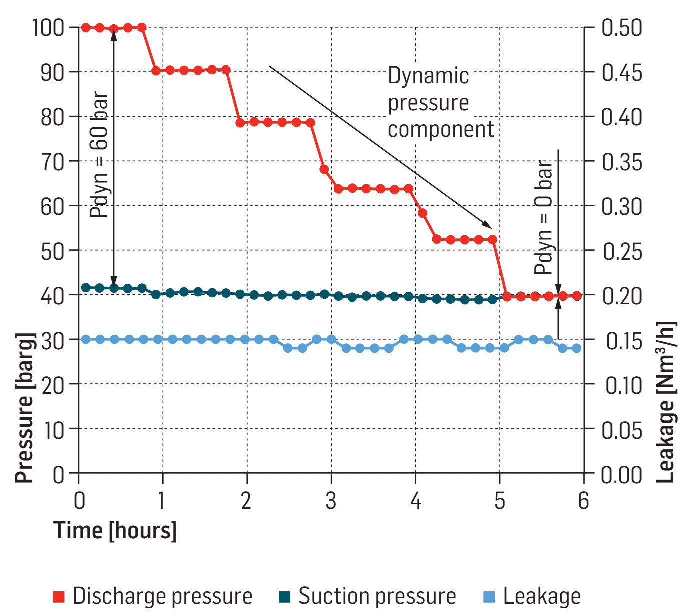
Fig. 3
Backflow of gas which is temporarily stored between the sealing elements through pressure-relief grooves



P_c = Pressure in the cylinder
P_p = Pressure in the packing ring cup

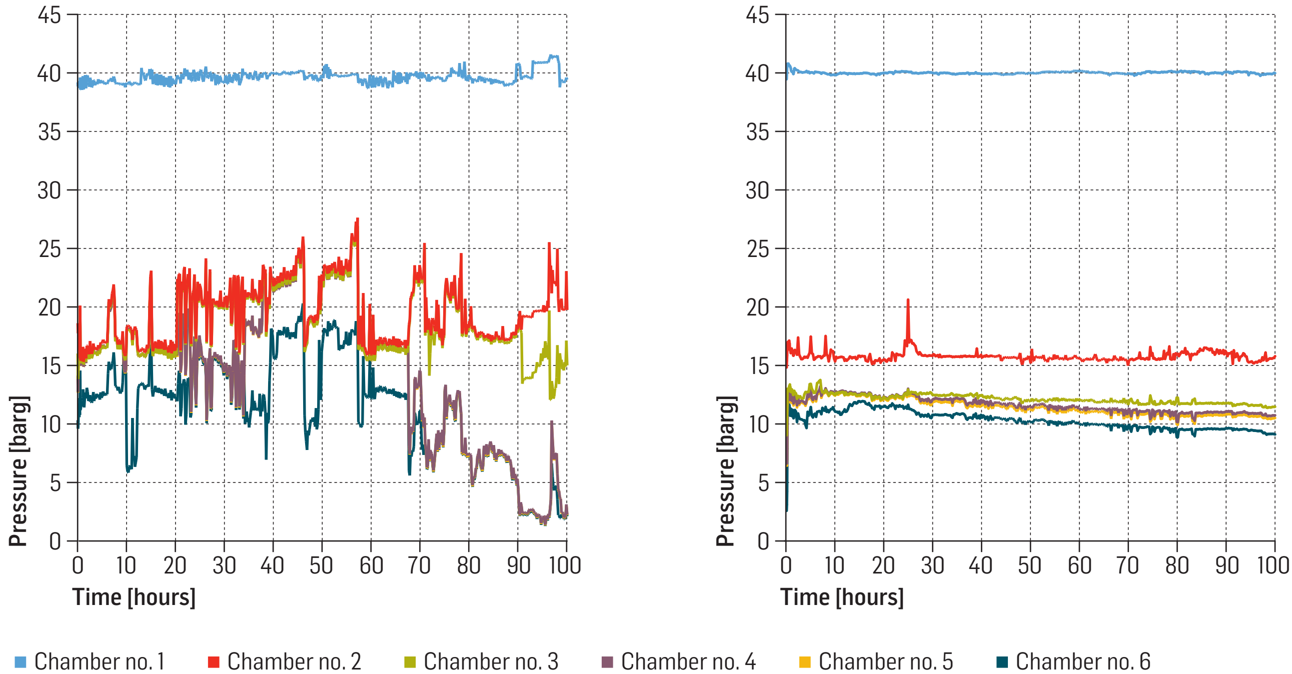
The dynamic pressure component can therefore disperse back into the compression chamber through pressure-relief grooves during the suction stroke, and has no effect on the leakage. The leakage figures measured in a series of trials show this clearly. In a series of trials, the final pressure was reduced in steps, starting from the suction pressure of 40 barg and a final pressure of 100 barg, until only a purely static pressure load with the magnitude of the suction pressure remained. The leakage was not seen to react in any way worthy of mention to the changes in the dynamic pressure component during the entire duration of the tests. Thus the static pressure difference constitutes the primary load parameters influencing the leakage rate. **Fig. 4**

Fig. 4
Effect of a stepwise reduction of the dynamic pressure components on the leakage of a packing



The dispersion of the dynamic pressure component back into the compression chamber also has a positive effect on the stability of the pressure distribution inside the sealing system. If sealing elements without pressure-relief grooves are used, the dynamic pressure component also drifts in the direction of the packing outlet (Fig. 5, left), so resulting in an unstable load on the individual packing elements. This unstable pressure distribution leads to rotational and translational movements of the sealing

Fig. 5
Pressures measured in the chambers for two packings with six sealing elements each, with pressure-relief grooves (right) and without (left)



elements, as a result of which damage to the sealing elements and to the chambers occurs. The conditions in the packing with pressure-relief grooves are far more stable in comparison (Fig. 5, right). **Fig. 5**

3.2 **EFFECT ON WEAR**

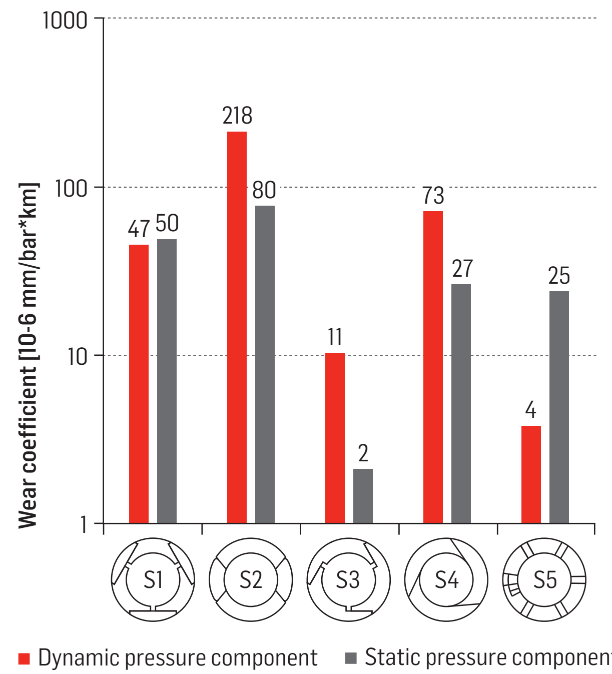
Comprehensive tests with various types of packing sealing elements showed that the dynamic and static pressure components, depending on the design, lead to different wear behaviour, although the sealing elements tested were all manufactured from the same material[1]. If the average wear of the sealing element is related to the effective average pressure difference during the test and to the sliding distance, a parameter is obtained that is similar to the wear coefficients commonly used in tribology.

Fig. 6 shows the wear coefficients obtained in this way for some of the investigated packing ring designs subjected to dynamic and static pressure loads. Particularly noticeable here is the pronounced sensitivity of the segmented designs tested (S1 to S4) to the dynamic pressure component, whereas the one-piece design (S5) withstands the dynamic load significantly better. **Fig. 6**

As has been shown above, the dynamic pressure component in a sealing system with intact, gastight sealing elements, has no effect on the leakage. It does, however, lead to high wear, failure by fracture and/or plastic deformation of the sealing elements – particularly in the case of segmented packing rings. Throughout the duration of operation therefore the true sealing elements must be securely protected from the damaging effect of the pulsating pressure load by a barrier of what are known as pressure breaker rings. As explained above, this requirement is far from satisfied with packings consisting of a single, contact-free throttle ring, and as a result the load of the dynamic pressure component is mainly distributed on the true sealing elements. Depending on the design and material composition of the sealing elements, this can significantly reduce the service life of the sealing system, even at low to medium loading. Above a critical threshold, the dynamic pressure component causes severe damage to the sealing elements, and may even lead to premature failure of the whole sealing system.

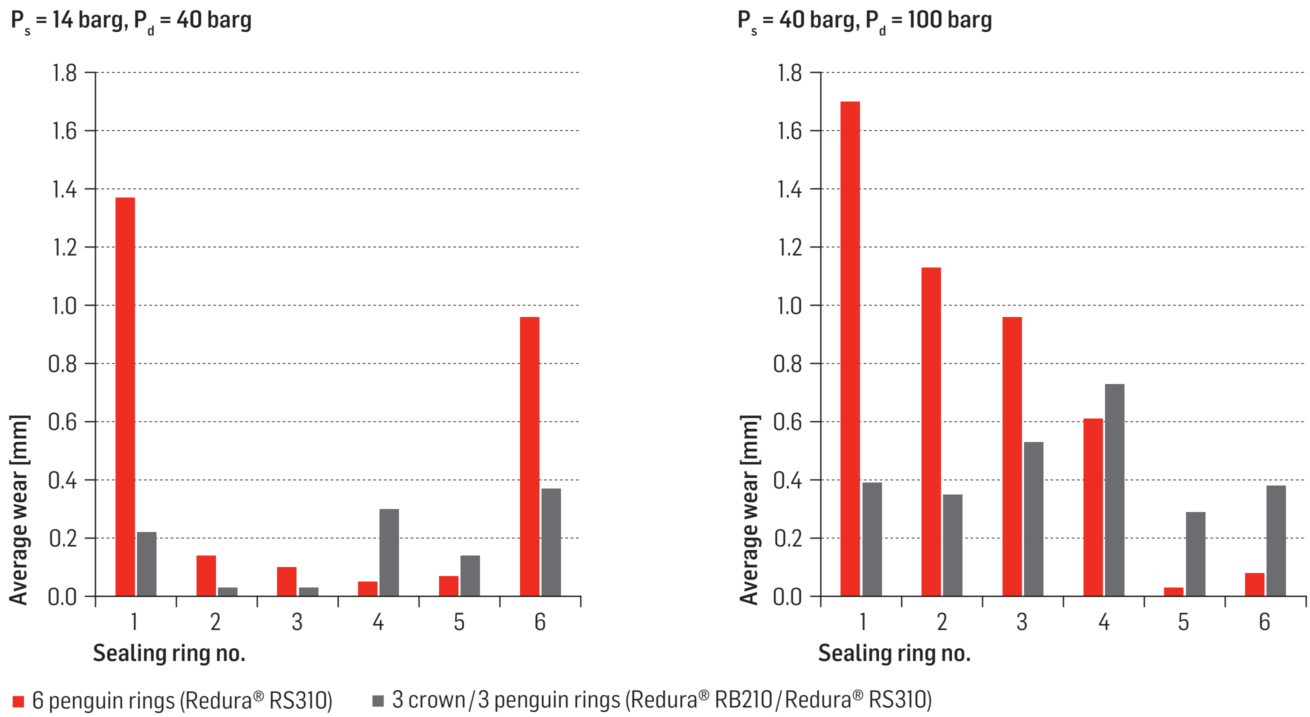
Experiments with differently designed packings, each having six sealing elements made of a PTFE/PPS-polymer blend subject to two different pressure loadings in a dry-running hydrogen compressor show this strikingly. Loading a packing of six sealing elements having a tangential step cut (penguin rings, Redura® RS310), with a suction pressure of just $P_s = 14$ barg and a final pressure of $P_d = 40$ barg thus leads, after 500 hours of operation, to increased wear at the first penguin ring arranged directly adjacent to the compression chamber (Fig. 7, left). If the dynamic pressure component component is raised from 26 bar to 60 bar ($P_s = 40$ barg, $P_d = 100$ barg), then in addition to a significantly increased wear rate, failure by fracture also occurs in the region of the joint sealing of the first three sealing elements, and the test therefore had to be terminated early after 264 hours due to high leakage (Fig. 7, right). **Fig. 7**

Fig. 6
Wear coefficients of different sealing element designs under dynamic and static pressure load



Replacing the first three penguin rings with a design better suited to the dynamic pressure component, for example with the crown ring (Redura® RB210)[2], results in a heterogeneous packing case design. The crown ring, which is not as tight as the penguin ring, is more robust in respect to loading by the dynamic pressure component, and has a wear-reducing pressure relief groove. This packing configuration allowed the 500-hour trial to be completed easily and without failure by fracture.

Fig. 7
Wear values for a homogeneous (6 penguin rings) and a heterogeneous (3 crown / 3 penguin rings) packing under two different loads



CHAPTER 4

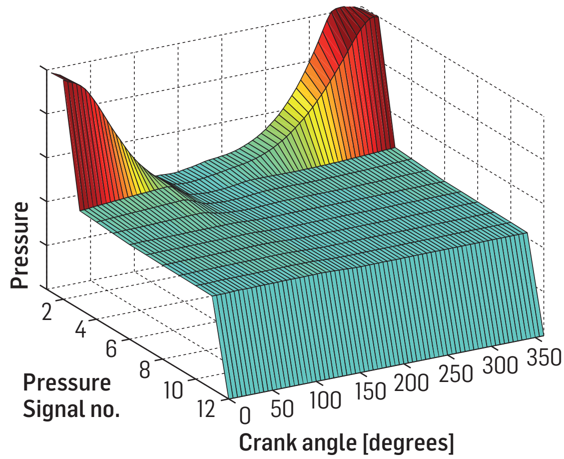
SEALING SYSTEM OPTIMISATION THROUGH HETEROGENEOUS DESIGN

The different effects of the dynamic and the static pressure components on the operating behaviour of the sealing system as a whole, and on the wear of the various sealing element designs, allows sealing systems to be optimised. This requires sealing elements which are specifically designed for the different effects of the two pressure components. In order to withstand the dynamic pressure component, robust designs, which are not necessarily very tight, and which allow pressure distribution over several sealing elements, are particularly appropriate. As an important and positive side-effect, overloading an individual sealing element through the simultaneous load with both pressure components is avoided. **Fig. 8**

The magnitude a nd the proportion of the two pressure components relative to the total pressure difference will place different demands on the required ring design. In the case of low to medium loads, typical, standard sealing elements may be sufficient. They already have the required properties, and only have to be optimised for the particular application in a heterogeneous sealing system. Piston rings with overlapping joints, for example, will have a particularly large radius near the transitional area, which minimizes the risk of failure by fracture during operation [3]. For high loads, sealing elements specially developed for this purpose are used. Their design differs significantly from the typical, widely used designs. For example, what are known as retained piston rings have proven themselves extremely well for protecting the true sealing rings from the dynamic pressure difference of single-acting pistons [4].

For double-acting pistons, the use of sealing systems with heterogeneous designs is also valuable, although a static pressure component is absent from this kind of compression. A particular challenge here is that of preventing the piston being overflowed from one compression chamber to the other, as this would lead to a drop in the capacity while at the same time bringing an increase in temperature. Here again, sealing elements with an efficient joint sealing have been proven. They ensure a high sealing efficiency over a particularly long period of operation, and are positioned in the centre of the piston. They are protected from the dynamic pressure component by robust piston rings that are arranged close to the compression chambers.

Fig. 8
Distribution of the dynamic pressure component over various sealing elements of an improved heterogeneously designed sealing system (right) compared to a conventional sealing system (left)



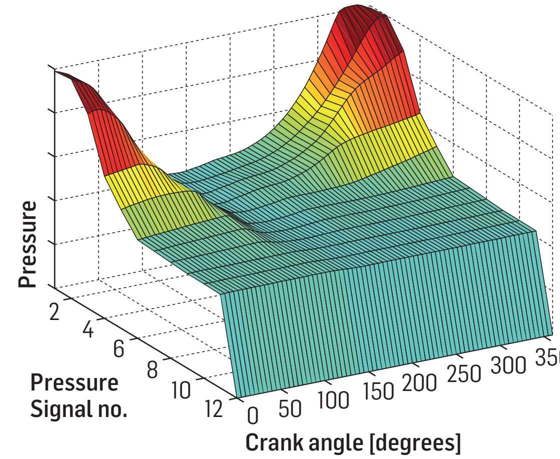
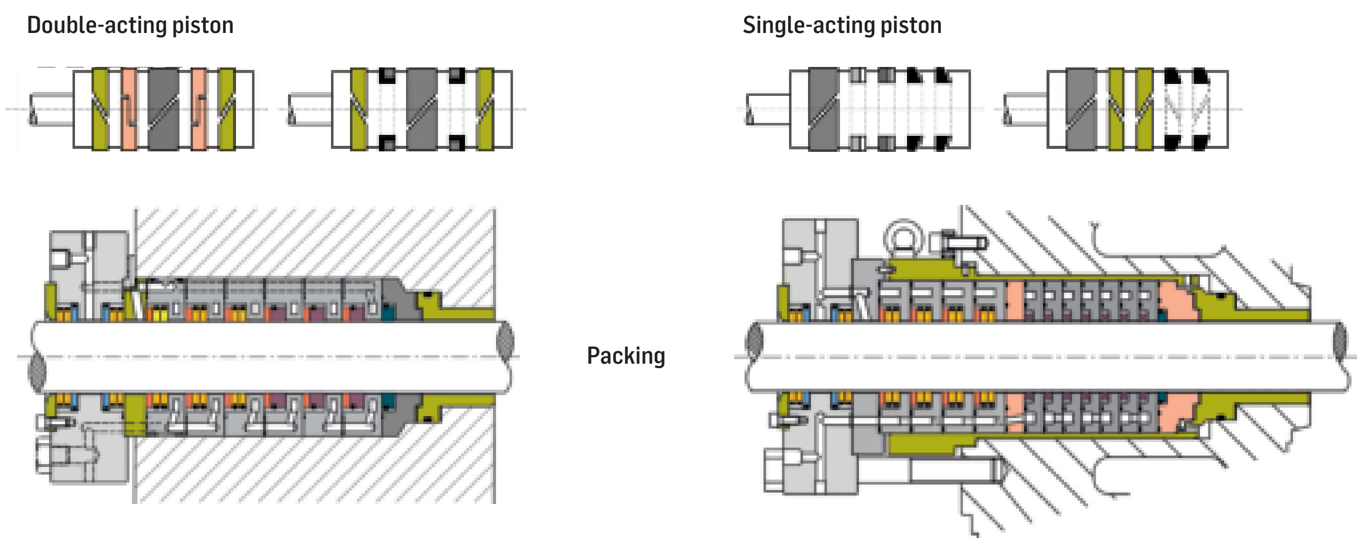


Fig. 9
Heterogeneously designed Redura® Sealing Systems for single- and double-acting pistons as well as for packings



If the dynamic pressure component can be kept away from the actual sealing elements for as long and as completely as possible, premature failure by fracture can be avoided. The achievable service life of the entire sealing system now depends, in addition to many other influences, most of all on the capacity of its sealing elements to compensate for wear. This maintains the frictional contact in spite of progressive wear. New concepts for wear compensation permit high exploitation of the radial ring diameter as a wear path, while at the same time maintaining a constant sealing efficiency [5]. Further development focused on this will allow the product portfolio to be expanded in future, bringing sealing elements with improved wear behaviour and with highest sealing efficiency.

The static pressure component is, however, also an important parameter for the design of the sealing system. In addition to the permissible mechanical stress, the friction energy generated by the sealing element is of particular importance. If a limit value, which depends on a large number of parameters, is exceeded, it is no longer possible to dissipate the frictional heat from the sealing surfaces, and the sealing system is thermally destroyed. The maximum permitted static pressure component is therefore an important design criterion, particularly for dry-running sealing systems. In practice, this is often neglected in favour of the total pressure difference. Table 1 here shows, by way of example, the two pressure components of four different sealing systems, all of which have a final pressure of

Table 1
Dynamic and static pressure components for four different sealing systems, each with a final pressure of 100 barg

Sealing system	Suction pressure [barg]	Static pressure after last sealing element [barg]	Dynamic pressure component [bar]	Static pressure component [bar]
Double acting piston	40	40	60	**0**
Single acting piston	40	16	60	**24**
Packing, normal application	40	0	60	**40**
Packing, recycle application	90	0	10	**90**

100 barg. Starting with a double-acting piston, in which the static pressure component is absent, it increases up to a value of 90 bar for a packing operating in a recycling stage. Such recycling stages are not primarily used to increase the pressure, but rather are there to convey, for instance, gas that has not yet reacted back to a reactor. Whereas the static pressure component of the three other sealing systems under dry running conditions do not present a particularly large challenge, the high value of 90 bar itself represents an overload for many sealing systems based on filled PTFE. In spite of the same final pressure, these sealing systems therefore demonstrate significant differences in their operating behaviour, and these must be taken into account in their design. Fig. 9 shows schematically different heterogeneously designed Redura® Sealing Systems for single- and double-acting pistons as well as for packings. **Table 1, Fig. 9**

Nomenclature

P_{dyn}	Dynamic pressure component
PTFE	Polytetrafluoroethylene
PPS	Polyphenylene sulfide
P_s	Suction pressure
P_d	Final pressure
P_c	Pressure in the cylinder
P_p	Pressure in the packing ring cup

Literature

1 Vetter, G.; Feistel, N.:
Behavior of dry-running piston rod sealing systems in crosshead compressors
Hydrocarbon Processing, September 2004, pp. 99–107

2 Feistel, N.:
Service life improvement of piston-rod sealing systems by means of pressure-relieved sealing elements
4th EFRC Conference, Antwerp, Belgium, 2005, pp. 23–30

3 Feistel, N.:
Influence of piston-ring design on the capacity of a dry-running hydrogen compressor
3rd EFRC Conference, Vienna, Austria, 2003, pp. 114–149

4 Feistel, N.:
Zehn Jahre erfolgreicher Einsatz von TID-Kolbenringen bei hohen Druckdifferenzen
Industriepumpen + Kompressoren 2, 2010, pp. 75–82

5 Feistel, N.:
Performance improvement of dry-running sealing systems by optimization of wear compensation
9th EFRC Conference, 2014, pp. 239–247

Burckhardt Compression

REDURA®
RINGS & PACKINGS ARE CHARACTERIZED BY

LONGEST MTBO (MEAN TIME BETWEEN OVERHAUL) AT LOWEST LEAKAGE

HIGHEST AVAILABILITY

LOWEST LIFE CYCLE COSTS

COMPRESSOR COMPONENTS
BEST PERFORMANCE AND LONGEST LIFETIME

- Compressor valves
- Redura® rings & packings
- Capacity control systems
- Capital parts
- Labyrinth piston compressor components
- Hyper secondary compressor components

SERVICES
THE FULL RANGE

- Burckhardt Valve Service
- Spare parts logistics
- Field service
- Technical support
- Revamps & upgrades
- Component repair
- Condition monitoring & diagnostics
- Training

Burckhardt Compression AG
CH–8404 Winterthur
Switzerland
Tel.: +41 (0)52 262 55 00
Fax: +41 (0)52 262 00 51
24-hour emergency tel.: +41 (0)52 262 53 53
aftersales@burckhardtcompression.com
www.burckhardtcompression.com

Your local contact

23.19.14.40-1 – Printed in Switzerland on FSC certified paper

Compressors for a Lifetime™