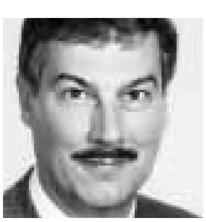
Burckhardt Compression

TEN YEARS OF SUCCESSFUL OPERATION OF TID PISTON RINGS

AT HIGH PRESSURE DIFFERENCES

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. He began his professional career as a design engineer with Mannesmann Demag Foerdertechnik in Offenbach/Main, Germany.
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 labyrinth piston compressors, his responsibilities are now for the development of oil-free sealing systems. In 2002 he gained a Ph.D. at the University Erlangen-Nuremberg, Germany, with a thesis on the operational behavior of dry-running sealing systems in crosshead compressors.

The limits of dry-running frictional sealing systems can be attained and even exceeded during oil-free sealing of gases at pressures amounting to 200 bar or more. The flow rates required in such applications are often relatively low, hence resulting in very small cylinder diameters particularly in the final compression stage. This poses a problem of optimization for sealing elements subjected to high loads: Such elements must not only possess the necessary strength and rigidity, but also be sufficiently capable of compensating wear. In practice, most sealing systems here do not provide a very satisfactory sealing efficiency and/or notably undershoot the generally accepted standard of 8'000 hours for minimum service life.

One solution for these applications relies on "retained" piston rings protecting the true friction rings against excessively high dynamic pressure differences. This well-known piston ring design is now combined with segmented piston rings resembling packing rings to a heterogeneous sealing system. Ten years of experience in operating such sealing systems have provided a positive picture, particularly where dry-running compression of hydrogen is concerned.

CHAPTER 1

INTRODUCTION

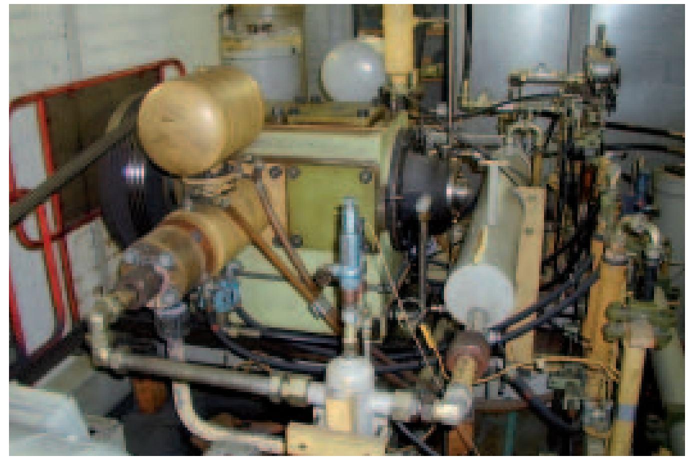
Though oil-lubricated Process Gas Compressors are still used mainly to handle very high pressure differences, oil-free sealing systems have also remained in demand, even for applications involving high final pressures, whether to fill gas containers or in the chemical industry, e.g. to manufacture pesticides. In the case of the frequently employed lubricated compressors, the lubricant is carefully removed from the gas flow after leaving the compressor. Quite elaborate at pressures of 200 bar or more, this process is dispensed with by oil-free compressors. During oil-free compression in this pressure range, however, the limits of dry-running friction seals can be attained or even exceeded, particularly at high average piston velocities. In this context, the contactless labyrinth principle has proven very successful in compressing gases of a high molecular mass up to final pressures in excess of 200 bar. So-called retained piston rings providing frictionless operation after having been run-in are found in similar applications.

Before the introduction of high-temperature polymers, materials based on filled PTFE were used preferentially for dry-running compression of hydrogen by means of frictional piston rings. Regardless of the employed filler, however, these compounds have inadequate high temperature strength during operation at high pressure differences, even if glass or carbon fiber is used. The outcome is significant creep under load[7]. This effect was especially noticeable in the area of the ring joint, so that a transition was soon made to simple piston ring designs without elaborate joint sealing, large numbers of piston rings being installed to improve sealing efficiency. In view of the high pressure differences, however, such systems do not provide very effective sealing[5].

Wherever gas properties permit, plastics from the group of high-temperature polymers such as polyetheretherketone (PEEK), polyimide (PI), epoxy resin (EP) etc. have proven especially suitable in handling high loads thanks to their favorable high temperature strength. Modified specially for dry-running, these high-temperature polymers mitigated the problem of cold flow. In practice, however, most applications of this sort require relatively low flow rates, sometimes resulting in particularly small values of less than 100 mm. for the cylinder diameter in the final compression stage. This poses a problem of optimization for the highly loaded sealing elements of the final stage. They must exhibit the necessary strength and rigidity as well as possess a good ability for wear compensation. However, impermissibly high rigidity would lastingly affect the wear compensation of one-piece piston rings. To avoid this, segmented designs are also employed preferentially in combination with a pressure relief mechanism[6].

Finally, dry-running sealing systems handling high loads often suffer from unfavorable friction pairs for a given application. Because "self-supporting" piston rings should comprise high-temperature polymers, this notably restricts the choice of materials available. Ring materials with better tribological characteristics often prove unfeasible due to their inadequate strength, thus usually falling short of the target service life of 8'000 hours, even with gases of a high molecular mass.

Fig. 1
Nitrogen compressor for investigating the operational behavior of dry-running piston sealing systems at a final pressure of 250 barg



CHAPTER 2

BENCH TESTS

In the early 1990s, the quest for dry-running sealing systems operable even at high pressure differences led to the installation of a test bench permitting various piston ring materials and designs to undergo continuous testing for the longest possible periods of time. In the final stage of a dry-running compressor serving as the test facility, dry nitrogen was compressed in single-acting mode from a suction pressure of 64 barg to 250 barg at an average piston velocity of 3.125 m/s and a final compression temperature of 175 °C. Given a constant pressure of 16 barg at the opposed side of the piston, this results in a maximum pressure difference of 234 bar over the entire sealing system, the dynamic pressure component of 186 bar posing a special challenge to sealing rings. The built-up piston installed for a cylinder diameter of 38 mm. enables an installation of up to ten piston rings. A bush made of nitrided steel serves as the counter body in the tests described next. **Fig. 1**

2.1 CONVENTIONAL PISTON RINGS

The first design recommended by a number of sealing-element manufacturers for dry-running sealing systems subjected to high loads was based on conventional piston rings incorporating a high-temperature polymer. A favorite variant for use with dry nitrogen was a homogeneous sealing system comprising piston rings with a scarf joint and made of modified PEEK.

The recommended cross-section of 5 * 5 mm, in conjunction with the small cylinder diameter of 38 mm and a flexural modulus of elasticity of more than 3 GPa at room temperature for modified PEEK – a very high value for dry-running materials – resulted in remarkably rigid piston rings. To ensure proper contact with the cylinder wall despite progressive wear, the piston rings are designed as self-tensioning rings without a separate tension ring. In a well-established procedure they are manufactured from circular rings larger in diameter by about 3% to 5%, a corresponding section being removed prior to installation in the smaller cylinder. This inevitably results in incomplete contact between the installed piston ring and cylinder circumference[8]. **Fig. 2**

If, instead of an additional tension ring, the piston ring's self-tensioning effect is to be retained as a means of minimizing clearances between sealing ring and cylinder when the compressor is started, there are various methods of achieving uniform contact in the installed state; however, each of these methods entails a further, costly work step. Used often for this purpose is a mechanism which varies the piston ring's radial thickness along the circumference, minimum values being set in the region of the joint.

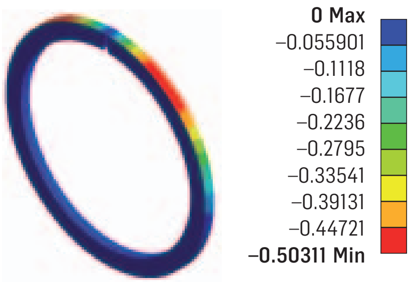
Despite reduced pre-stressing and an optimized ring shape, however, stable operation could not be achieved with a sealing system comprising ten piston rings each with a scarf joint. One reason presumed for the inadequate performance is high leakage through the scarf joints.

Consequently, piston rings with effective joint sealing should improve the situation. Suitable for this purpose is the twin ring which, depending on design, can provide consistently excellent sealing efficiency over a long operation period[5]. To protect this extremely tight design against the destructive effect of the high dynamic pressure difference, two endless rings with minimal radial clearance and six two-piece rings were arranged between the twin rings and the compression chamber.

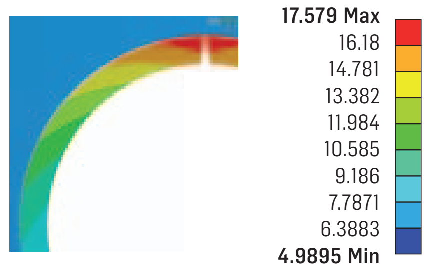
Though a number of variants of this sealing design were put successfully into operation, their operational behavior turned unstable after just 416 to 1'070 hours, intense fluctuations in leakage and peak values of more than 40 Nm^3/h being observed here. Accordingly, uninterrupted overshoot of a leakage limit of 25 Nm^3/h for more than three days was then defined as a criterion for aborting subsequent tests. An analysis of the dismantled sealing elements revealed a high degree of wear only on the filler rings of the twin ring configurations, all other components still being in sound condition. In particular, it was evident that the

Fig. 2
Deviations between a self-tensioning piston ring and the cylinder contour

Gap
Type: Gap
Unit: mm
Time: 1
06.07.2009 15:50



Total Deformation
Type: Total Deformation
Unit: mm
Time: 1
06.07.2009 15:48



two-piece piston rings were not able to prevent the rise in leakage following failure of the twin rings.

2.2 RETAINED PISTON RINGS

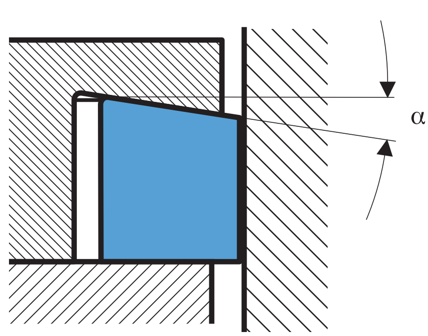
A retained piston ring is based on the principle of limiting radial wear to a point where the ring's sealing function is transformed from that of a classic friction ring to that of a contactless gap seal. For this purpose, the piston ring's cross-section is designed such that the ring, after undergoing a defined degree of running-in wear, makes form-fit contact with at least one chamber wall to prevent any further material loss. Earlier designs of retained piston rings incorporating a T-shaped cross-section permitted a use of filled PTFE at final pressures of up to 200 bar and more. However, this type of piston ring is very elaborate to manufacture and requires a built-up piston, for instance. As a result, its use is not widespread.

This series of tests was also meant to examine a retained piston ring's potential for use at high pressure differences. Initial tests with a classic variant of this design did not reveal any significant advantages over conventional friction rings: A distinct rise in leakage was observed after about 600 operating hours. During subsequent optimization measures, it became clear that the distance of running-in wear to achievement of the retained state must be as short as possible so that the intrinsically uneven material loss along the piston ring's circumference does not negatively influence sealing efficiency in the gap-seal mode.

To improve sealing efficiency, the piston ring's radial flutter in the retained state during exposure to the dynamic pressure difference was to be avoided. An obvious option here is to stabilize the piston ring by means of a tension ring. However, this additional component can be omitted if it is possible to design the piston ring such that friction forces prevent it from fluttering in the ideal position, i.e. at a minimal radial clearance between the sealing surface and cylinder wall. This can be achieved by designing the ring shoulder as a wedge surface whose angle α is equal to that of the confining surface in the chamber ring, and setting the angle α to a value which fulfills the self-locking (clamping) condition[1]. **Fig. 3**

Using ten such optimized retained piston rings made of modified PEEK, it was possible to set an unprecedented record of 1'500 hours before a substantial rise in leakage. During dismantling, seven piston rings turned out to have made the transition to the fixed state, i.e. they were fixed between their chamber walls, so that the potential of the sealing system's friction mode was almost fully exhausted. Further attempts to increase service life to the target of 8'000 operating hours using a sealing system based solely on retained piston rings under observance of the specified termination criterion for leakage proved unsuccessful.

Fig. 3
Schematic representation of a retained piston ring in the fixed state



2.3 COMBINATIONS OF RETAINED AND CONVENTIONAL PISTON RINGS

Since the performance of the sealing system entirely comprising retained piston rings proved to decline at a relatively early stage, a combination of retained piston rings and conventional twin rings was selected to improve the operational behavior. Four retained piston rings were positioned near the compression chamber to protect the six succeeding twin rings against the high dynamic pressure difference. All sealing elements were again made of modified PEEK.

Compared with the previously tested variants, this sealing system exhibited an impressively low leakage rate which only began to fluctuate below the specified termination threshold of 25 Nm^3/h after about 2'900 hours. After 4'008 hours, however, the operational behavior became unstable and the test had to be aborted. An analysis of the sealing system revealed no failure by fracture of the twin rings, though all four retained piston rings had attained the fixed state. This made it clear that retained piston rings provide sufficient protection also when operating as contactless sealing elements. However, the retained piston rings were not able to prevent notable material loss on the twin rings, which led to the unstable operational behavior toward the end of the test. Accordingly, a plan was made to further lengthen service life by means of a PEEK material with better tribological characteristics. However, it soon became apparent that the original modified PEEK comprising PTFE, graphite and carbon fiber each in a proportion of 10% by weight had possessed the most favorable wear properties of all. Especially surprising was the decline in service life to just 2'130 hours in the case of the less suitable variants, despite their identical composition. One of the probable reasons for this was the influence of the various techniques, e.g. compression molding, extrusion and injection molding, available for manufacturing modified PEEK.

In a last optimization step, the ratio between retained piston rings and twin rings was adjusted by adding two more twin rings. This was meant to increase the proportion of friction rings in order to lengthen the efficient sealing period. However, this measure anticipated as beneficial also turned out to be counterproductive. Though the two extra twin rings provided an additional wear distance, the two remaining retained piston rings no longer offered enough protection. Consequently, the twin rings were exposed to the dynamic pressure difference's destructive effect, hence shortening the service life to just 1'300 hours.

CHAPTER 3

DEVELOPMENT OF A TID PISTON RING

Tests with conventional, one-piece piston rings showed that they are not suitable as self-supporting elements for small cylinder diameters. Examinations of alternative variants comprising segmented piston rings revealed a completely unacceptable sealing efficiency, and though the retained rings offered sufficient protection against dynamic pressure difference, the sealing efficiency of this homogeneous system was also disappointing.
The load parameters of a single-acting piston are similar in terms of composition to that of a piston-rod sealing system (packing). Here, too, the dynamic pressure difference exerted on the sealing system is usually accompanied by a static pressure difference which places high demands on sealing technology[6]. The segmented sealing elements normally employed in the packing had proven quite suitable for this task, making it advisable to select a similar design for use in a single-acting piston. Due to the restricted installation space provided by the piston, however, the cover ring usually forming part of conventional sealing-ring pairs was to be left out. Selected in view of this was the packing-ring design comprising segment sections tangential with respect to the internal diameter (TID) which delivers satisfactory sealing efficiency even without a cover ring.

An additional tensioning element is absolutely necessary here to activate segments during compressor start and to compensate wear. To protect the cylinder against substantial damage resulting from dislodgement of the tension ring from its groove, PEEK filled with carbon fiber in a proportion of 30% by weight was used instead of metal as the spring material. The major difference to standard, segmented piston rings here is that the three segment joints are sealed axially by an endless base ring installed with a very low clearance in the cylinder bore. Accordingly, a TID piston ring consists of three TID segments, a tension ring and a base ring. **Fig. 4**

The base ring's supporting function allows the TID segments to be designed with a very small axial dimension of just 3 mm, which significantly reduces friction power[2]. In the initial variant, the TID segments and base ring again comprised modified PEEK, and TID piston rings were combined with four retained piston rings to ensure protection against the high dynamic pressure difference. **Fig. 5**
The tension rings used to push the TID segments radially outward require these to be fixed in the piston ring grooves by means of a slotted plastic sleeve until insertion into the cylinder.

The combination of retained piston rings and TID piston rings brought success to efforts to achieve a service life of 8'000 hours for dry-running sealing systems exposed to high loads. However, this system's operating characteristics differed notably from those of the combination of retained piston rings and twin rings. Whereas the leakage rate exhibited by the twin rings remained very low over a long operating period before suddenly rising to

Fig. 4
Design of a TID piston ring comprising TID segments, a tension ring and a base ring

impermissibly high levels, the TID piston rings provided efficient sealing only during the first 1'000 hours, after which the leakage began to fluctuate within the permissible range. It took 8'206 hours for the operational behavior to finally become unstable, the leakage rate then rising above the specified termination threshold. An analysis of the TID piston rings revealed a low average wear of just 1.53 mm. With an original radial thickness of 4 mm, these rings were still in working order. The drop in sealing efficiency was therefore attributed to the state of the tension rings, all of which were broken and no longer in working order.

Because the TID piston ring does not possess any overlapping joints or bridges which could fail under pressure, and because the base ring effectively prevents flow into the clearance between the cylinder and piston, a high-temperature polymer is not absolutely necessary for the TID segments. Consequently, the next optimization measure was to employ a special polymer blend which exhibited outstandingly low wear rates in dry nitrogen but had not been usable yet as a self-supporting piston ring, due to its excessively low heat resistance. The only worrisome factor here was the relatively high temperature of 175°C which, accompanied by the high pressure difference, negatively influences the wear characteristics of the PTFE/PPS blend. Accordingly, a decision was made to conduct a basic test with a limited duration 1'000 hours at a reduced final pressure of 200 barg and a final compression temperature of just 150 °C. The outcome was encouraging: No leakages were measured during the entire test, and the sealing elements dismantled subsequently showed no signs of wear or overloading.

Finally, a composite material comprising carbon fibers wound in a PEEK matrix was selected for the tension rings in order to optimize their fracture strength, and the resultant sealing system put again into continuous operation under normal test conditions with a final pressure of 250 barg. This sealing system achieved the target of 8'000 hours with operating characteristics much more favorable than those of the PEEK variant. Only after 8'425 hours was the test ended for the purpose of analysis, with the sealing system still fully functional. By then, the leakage rate which had fluctuated negligibly the whole time was still just 35% of the termination threshold, having reached a peak value of only 12 Nm^3/h in the course of the entire test. The friction rings' average wear of just 0.59 mm clearly confirms the polymer blend's superiority over the previously used modified PEEK in this application. Assuming that the variant incorporating the polymer blend also provides proper operation before an average wear of 1.53 mm is attained, this would enable a further, significant increase in test duration. Table 1 shows the differences between the examined variants, indicating clear advantages for the combination of retained piston rings and TID rings. **Table 1**

Fig. 5
Sealing system for a single-acting piston, incorporating a combination of retained and TID piston rings

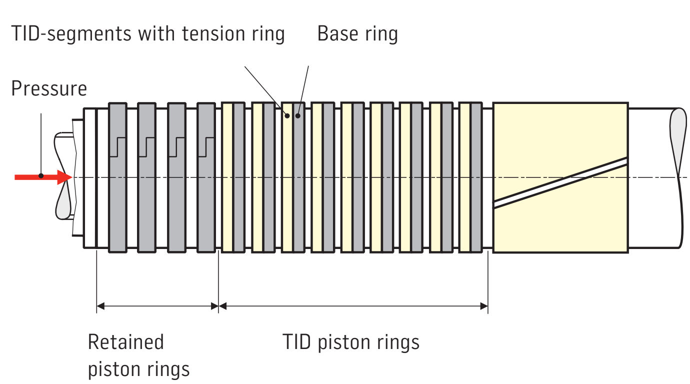


Table 1
Comparison of various sealing systems tested in dry nitrogen at a final pressure of 250 barg

Sealing system	Conventional piston rings	Retained piston rings	Retained piston rings / twin rings	Retained piston rings / TID rings	Retained piston rings / TID rings
Material	mod. PEEK	mod. PEEK	mod. PEEK / mod. PEEK	mod. PEEK / mod. PEEK	mod. PEEK / polymer blend
Test duration	416–1'070	664–1'545	1'300–4'008	8'206	8'425
Sealing efficiency	poor	poor	good	moderate	good

CHAPTER 4

PRACTICAL EXPERIENCES

For dry-running compression to high pressures, many crosshead compressors incorporate step pistons in the last two compression stages. The advantage of this, given an appropriate arrangement, is that the compression chamber of the final stage exposed to high load need not be sealed off additionally against the piston rod, i.e. the maximum load on the step piston's dry-running packing corresponds to the second-last compression stage, hence significantly lengthening the service life of this sealing system. A disadvantage of compression chambers arranged in this manner is the resultant single-acting mode with a static pressure difference as the primary load parameter influencing the leakage rate on at least one of the two sealing systems, depending on the pressure level in between.

Fig. 6 shows a typical example of such a dry-running compressor, whose two-crank design provides double-acting compression in the first stage (illustrated on the right), and single-acting compression in the step piston's second and third stages. So far, these heavy-duty dry-running Process Gas Compressors, manufactured by Burckhardt Compression AG, are used preferentially for oil-free compression of hydrogen from a suction pressure of 16 to 17 barg to a final pressure of 200 to 225 barg. The final pressure is often not constant, instead varying in dependence on the trailer pressure in the case of trailer filling compressors, or between a storage tank's minimum permissible pressure and the specified final pressure in applications found in the chemical industry. These compressors' flow rates lie between just 500 and 1'500 Nm^3/h, resulting in small diameters of 50 to 100 mm in the final stage. **Fig. 6**

4.1 COMBINATION OF RETAINED PISTON RINGS AND TWIN RINGS

Such compressors were first designed in 1994 using the knowledge of sealing technology available at the time: The second stage involving final pressures between 85 and 100 barg comprised solely twin rings, while the third stage comprised a combination of retained piston rings and twin rings similar to the test bench variant described in section 2.3. All sealing elements in these stages were made of modified PEEK. The expected annual operating period of these compressors was about 4'000 hours, a value which should also be achievable by the final stage's sealing system.

Observed at a relatively early point in time, the first deviations from ideal operational behavior were caused by fracture failure. It turned out that the gas-tight joints of the retained piston rings made of extruded, modified PEEK were not able to withstand the mechanical loads. Without adequate joint sealing, the retained piston rings' sealing efficiency in hydrogen did not suffice to protect the twin rings downstream against the high dynamic pressure difference. Fig. 7 (left) shows the effect of a high dynamic pressure difference of 100 bar on twin rings made of modified PEEK. **Fig. 7, left**

Fig. 6
Two-crank, three-stage, dry-running compressor for compressing hydrogen to a final pressure of 200 barg

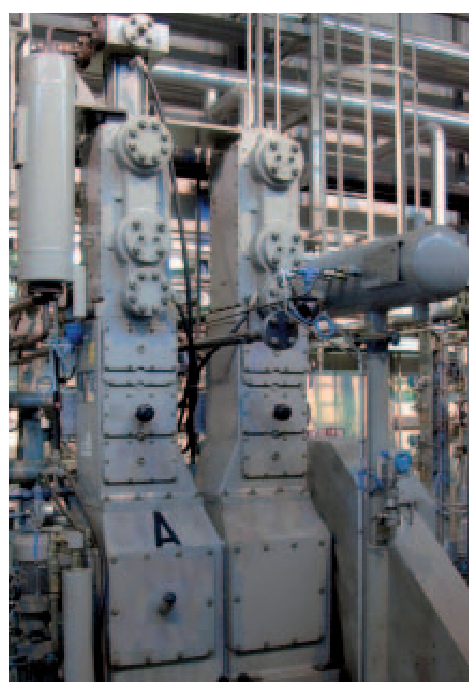


Fig. 7
Failure by fracture (left) and uneven material loss on twin rings made of modified PEEK and subjected to high pressure difference

The use of injection-molded, modified PEEK with much better mechanical properties made it possible to eliminate fracture failure[3]. Nonetheless, only operating periods ranging from 2'000 to 3'000 hours were achieved before the flow rate began to fall below the guaranteed value. An analysis of the dismantled sealing elements showed that unequal wear of cover ring and sealing ring and uneven removal of material along the circumference had led to inadequate sealing efficiency long before attainment of the permissible radial wear. Use of a form-fit coupled twin ring successfully stopped the unequal material removal between the two parts of the twin ring. This has an extraordinarily positive influence on sealing efficiency in the second stage[5]. However, the uneven material loss along the ring's circumference, occurring especially in the highly loaded third stage, remained unavoidable. **Fig. 7, right**

4.2 COMBINATION OF RETAINED PISTON RINGS AND TID PISTON RINGS

After the positive experiences made with TID piston rings in the bench tests, the first system incorporating the advantageous polymer blend was installed in July 1999 at a customer's facility as a replacement for the twin-ring version operated hitherto. Until then, a maximum service life of 3'323 hours was achieved by the hydrogen compressor at a final pressure of 200 barg and cylinder diameter of 60 mm in the final compression stage. The TID piston rings' delicate design and modified installation procedures initially caused wonderment over the new sealing system. After successful commissioning, however, the system was able to prove itself convincingly, even in handling hydrogen: The twin-ring variant's typical service life was easily exceeded with no deterioration in the high sealing efficiency. It was not until March 2001 that the sealing system was dismantled and analyzed after an operating period of 7'331 hours.

Especially encouraging were the low wear of the TID segments and the sound condition of the tension rings made of composite fiber. However, the base rings made of modified PEEK were all broken. The problem of failure by fracture was later remedied by switching over to injection-molded modified PEEK of a higher strength. This design ultimately achieved a service life of 14'747 hours, far exceeding the minimum requirement of 8'000 hours for sealing systems, also at high pressure differences during dry-running.

Today, all of Burckhardt Compression AG's dry-running compressors designed for high pressure differences incorporate a sealing system based on TID piston rings. An analysis of operating data gathered during the last ten years typically revealed an increase in service life by at least a multiple of four compared with the twin-ring variant. If a built-up piston is already present, conversion from a twin-ring design to TID piston rings usually poses no problems. Slightly larger than twin rings in terms of axial and radial dimensions, TID piston rings often need to be accompanied by new piston cups. If a cylinder was originally furnished with piston rings made of modified PEEK which now need to be replaced with TID piston rings made of a different material such as a polymer blend, conversion can be performed very easily simply by removing the PEEK piston rings' transfer film and producing a new counter-body surface compliant with the new material's specifications.

CHAPTER 5

POTENTIAL OF THE TID PISTON RING

TID piston rings essentially increase the efficiency of heavy-duty piston sealing systems in three different ways:

a) Segmentation of the piston rings, which increases radial dimensions without hindering wear compensation.

b) Very high sealing efficiency of segmented piston rings, attributable to the combination of the TID design for the sealing element and the supplementary base ring for axial sealing of the ring joints.

c) Elimination of the sealing rings' self-supporting function through the use of a base ring, and abandonment of bridges or overlaps for the joints of the sealing element, which can consequently be chosen from a wide spectrum of materials.

Malfunctions during operation of TID piston rings so far have been caused mainly by a breakage of base rings. Tested therefore as ring materials in addition to various PEEK qualities with improved mechanical attributes were also bronze and composite fibers, the test results proving successful and revealing a further increase in load handling capacity. Extending the spectrum to materials previously unfeasible due to their inadequate strength for self-supporting designs offers the greatest potential for further improvement in performance of TID piston rings. Because the TID segments still comprise friction rings, however, the selected materials must meet the same minimum application-specific tribological requirements as those laid down for conventional designs.

Of special importance to the operational reliability of TID piston rings are the retained piston rings installed upstream. Besides providing efficient protection against the dynamic pressure difference, which can prove especially destructive for sealing elements with a high sealing efficiency, retained piston rings also distinctly separate the true friction rings from the compression chamber and the temperatures prevailing therein. This expedites a use of materials with a low maximum operating temperature.

CHAPTER 6

SUMMARY

In a dry-running nitrogen compressor, various concepts of oil-free piston sealing were compared together during continuous tests at a final pressure of 250 barg. In this process, conventional, one-piece, self-supporting piston rings turned out to be unsuitable for small cylinder diameters. Examined alternative variants comprising segmented piston rings exhibited a wholly inadequate sealing efficiency, and though the retained rings provided sufficient protection against the dynamic pressure difference, their capacity to seal as a homogeneous system was also disappointing.

Because the load parameters of a single-acting piston are similar in terms of composition to that of a piston-rod sealing system, a logical conclusion was to furnish single-acting pistons with a similar design based on the packing-ring design comprising segment sections tangential with respect to the internal diameter (TID). A TID piston ring comprises three TID segments, a tension ring and a base ring which not only contributes toward joint sealing, but also enables a use of ring materials which would otherwise be unsuitable for handling high pressure differences, due to their inadequate strength.

The operational reliability of TID piston rings at high pressure differences depends greatly on a presence of frictionless retained piston rings. Besides providing efficient protection against dynamic pressure difference, these rings distinctly separate the friction rings from the compression chamber and the temperatures prevailing therein. Since 1999, heterogeneous piston sealing systems comprising TID piston rings and retained rings have been successfully employed in dry-running hydrogen compressors operating at high pressure differences. An analysis of operating data gathered over the last ten years revealed an at least four-fold increase in service life compared with twin-ring designs.

Notation

PTFE Polytetrafluorethylene
PEEK Polyetheretherketone
PPS Polyphenylensulfide

References

1 Burckhardt Compression AG:
Piston compressor for an oilfree gas compression
European Patent EP0541482 B1, 1992

2 Burckhardt Compression AG:
Sealing element for dry running systems and the use of a sealing element of this kind
European Patent EP0985108 B1, 1997

2 Burckhardt Compression AG:
Method of manufacturing a sealing element, and sealing element manufactured by that method
European Patent EP0916044 B1, 1997

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

5 Feistel, N.:
Influence of piston ring design on the capacity of a dry-running hydrogen compressor
3rd EFRC-Conference, Vienna, Austria, 2003, pp. 141–149

6 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

7 Feistel, N.:
Dry-running sealing systems in practice – new challenges by new materials
5th EFRC-Conference, Prague, Czech Republic, 2007, pp. 51–62

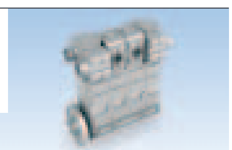
8 Retz, N.:
Investigation of the deviations between a self-tensioning piston ring and the cylinder contour by means of FEM
Internal development report of Burckhardt Compression AG, Winterthur, 2009

Burckhardt Compression

RECIPROCATING COMPRESSORS

LEADING TECHNOLOGY FOR LOWEST LIFE CYCLE COSTS

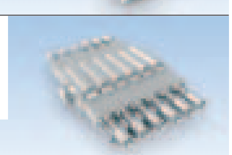
Laby® Compressors
Contactless and oil-free



Laby®-GI Compressors
Fully balanced

Process Gas Compressors
API 618 – full compliant

Hyper Compressors
Safe and reliable up to 3'500 bar



COMPRESSOR COMPONENTS

BEST PERFORMANCE AND LONGEST LIFETIME

Valves:
- Burckhardt Poppet Valve™
- Burckhardt Plate Valve™
- Manley® valve, licensed by Burckhardt Compression

Piston rings

Guide rings

Piston rod packings

Packing rings

Oil scrapers

Hyper/secondary compressor parts

CUSTOMER SUPPORT SERVICE

THE FULL RANGE OF SERVICES

Burckhardt Valve Service

Spare parts logistics

Field service

Technical support

Component repair

Engineering services

Revamps

Condition monitoring & diagnostics

Training

FOR ALL YOUR RECIPROCATING COMPRESSORS

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

Your local contact

21.16.14.40 – Printed in Switzerland on FSC certified paper

Compressors for a Lifetime™