INFLUENCE OF PISTON RING DESIGN ON THE CAPACITY OF A DRY-RUNNING HYDROGEN COMPRESSOR

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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.

In order to achieve the highest possible volumetric efficiency during oil-free compression of hydrogen, it is necessary to maximize the performance of dry-running sealing systems. The compression of small and medium flow rates to high pressures, as required for filling gas bottles and for process gas compression in the chemical industry, can be viewed as especially critical aspects. Due to the very small piston diameters typically involved in the final stages, the covering of the sealing-element joints plays a significant role here. The results of tests conducted with a variety of piston ring designs in a dry-running crosshead compressor are used to elucidate the most important differences and the related consequences for the hydrogen capacity.

CHAPTER 1

INTRODUCTION

Due to the extraordinarily high costs associated with potential production losses, high demands are placed on the reliability of crosshead compressors, which are today used mainly by the chemical industry for the purpose of compressing process gas. The notable rise in the performance of dry-running sealing systems over the last few years has been accompanied by demands for ever longer maintenance intervals.

However, not all such demands can be fulfilled with a sufficiently low specific energy consumption. Particularly as concerns the oil-free compression of hydrogen, the capability of simple sealing systems to manage high pressure differences is attended by a low volumetric efficiency. Although such sealing systems have a low price, they can lead to a design of unnecessarily large machines and penalize the operator with higher operating costs and often with a rapidly decreasing flow rate.

Gas leakages from piston-rod sealing systems are usually under critical observation, their rates being closely monitored, whereas gradual drops in flow rate often remain unregistered, or are tolerated until their consequences assume significant proportions. In the case of large crosshead compressors with a drive power of several hundred kW, an average loss of 10% in capacity over a period of 8'000 hours generates expenses which by far outweigh the costs saved by the sealing elements. In other words, one often saves at the wrong end.

Different concepts are used for sealing compression chambers of crosshead compressors, in accordance with the type of operation involved (single or double acting compression). These concepts range from relatively leaky designs such as piston rings with a butt or scarf joint, to gastight constructions costing up to twice as much. The results of tests conducted with a variety of piston ring designs in a dry-running crosshead compressor are used to elucidate the most important differences and the related consequences for the hydrogen capacity.

CHAPTER 2

VARIOUS STYLES AND APPLICATIONS OF PISTON RINGS

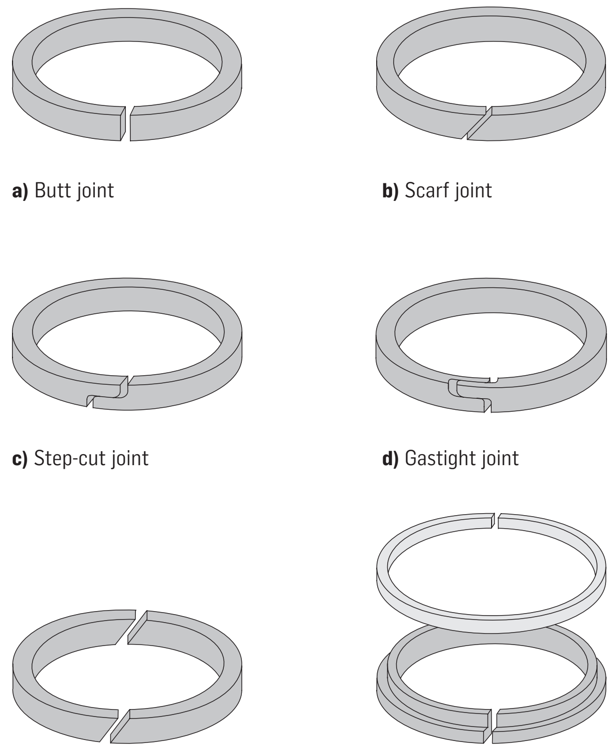
Piston rings for double-acting cylinders with large diameters usually are one-piece rings with a butt or scarf joint. **Fig. 1a/b**
In the case of these designs, the joint is not sealed either axially or radially, thus minimizing the risk of failure by fracture even for brittle materials. However, progression in wear is accompanied by a proportional increase in the flow area of the ring joint and a resultant increase in the quantity of leakage gas.

The covering of the sealing ring joint plays a major role, particularly in the case of single-acting compression stages with small cylinder diameters. Experiments with a single-acting air compressor[2] have shown that already at a ratio of 10% between the flow area of the joint and the total flow area of a piston ring (sum of joint, axial and radial flow areas), 77% of the total leakage gas flows through the ring joint. Under these conditions, the joint gap – as opposed to Bartmann's publication[1] – can be regarded as the dominating flow area for the leakage mass flow. All the more astounding is the occasional use of two-piece designs with scarf joints even for the compression of gases possessing a low molecular weight to high pressures. **Fig. 1e**

In accordance with the sealing function required, piston ring joints can be shaped to any degree of complexity within the bounds determined by the material properties and ring dimensions. Particularly when it comes to handling high pressure differences, however, dry-running materials with a strong tendency toward cold flow can cause a failure of the ring joint, while brittle materials are susceptible to fracture. A logically advisable increase in the axial dimensions of the piston ring is limited by the simultaneous increase in frictional heat which impairs tribological conditions. The use of high-temperature polymers can also create problems during the sealing of very light gases, because as the degree of wear increases, the high modulus of elasticity prevents full compensation of the gaps created between the sealing element and cylinder wall as a result of uneven material wear.
In addition, modern polymer blends not only improve the sealing efficiency, but often also enable longer operating periods, especially in the case of very dry gases. Consequently, a great challenge is posed by the gastight sealing of ring joints for dry-running compression of hydrogen to high pressures, as required during the filling of gas bottles, for example.

The simplest design of a joint seal – termed step-cut joint (Fig. 1c) – only involves covering in the axial direction, radial flow around the sealing element being possible. **Fig. 1c**
The abrupt reduction in the ring's cross-section on the transition to the joint overlap results in fracture in the case of brittle materials. **Fig. 2**
Although rounding the transition off carefully with a large radius lowers the risk of failure by fracture, a dynamic pressure difference p_{dyn} of more than approximately 3 MPa is not permissible for the polymer blend used here. **Fig. 3**

Fig. 1
Various styles of piston rings



a) Butt joint
b) Scarf joint
c) Step-cut joint
d) Gastight joint
e) Two-piece ring
f) Twin ring

In the case of piston rings with a gastight joint, the joint is also sealed radially. However, this type of joint sealing is very susceptible to fracture, due to the additional reduction in the cross-section of the overlap. **Fig. 1d**

In the case of the twin ring – another gastight piston ring design – the butt joint of the rectangular sealing ring is sealed by a surrounding, L-shaped cover ring. **Fig. 1f**
If both parts of the twin ring are made of the same material, the problems mentioned above might occur again either as failure by fracture or creep in the region of the joint, or as a result of insufficient wear compensation. With this style, however, it is possible to design the cover ring with small dimensions out of a high-temperature polymer, leaving the remaining space for the sealing ring consisting of a polymer blend, for instance. In addition, the distribution of the pressure difference along the frictional sealing surface[2] in the case of the twin ring results in unequal wear of the two ring parts. Especially if high loads are exerted on the twin ring, this effect can increase the wear of the sealing ring compared with the wear of the cover ring by a factor of up to 3. **Fig. 4**

CHAPTER 3

EXPERIMENTAL SET-UP AND METHODOLOGY

3.1 EXPERIMENTAL SET-UP

The influence of piston ring design on the hydrogen capacity was investigated using a dry-running, two-stage, horizontal cross-head compressor (Fig. 5) with a stroke of 160 mm, maximum speed of 850 min^{-1} and maximum drive power of 400 kW. The compressor was designed for a maximum final pressure of 20 MPa and a maximum average piston velocity of 4.0 m/s. **Fig. 5**

With a piston diameter of 75 mm, the first, double-acting stage compresses the hydrogen using a total of eight piston rings (four on each side of the centrally positioned guide rings). The second stage, also with a piston diameter of 75 mm and equipped with nine piston rings, increases the pressure single-acting at the rod side of the piston. The piston rod diameter in both stages is 50 mm. To cover a large range of pressure differences while minimizing gas losses, operation took place in a closed cycle at a suction pressure higher than atmospheric pressure. The gas cycle was fed with hydrogen having a dew point ≥ –65 °C.

The hydrogen flow rate was measured with a thermal mass flowmeter for potentially explosive atmospheres. **Fig. 6**
With the bypass concept, a fraction of the gas flow to be measured is routed via a small sensor tube, where it causes a temperature difference between two electric windings (acting as both heaters and resistance-temperature detectors) which is proportional to the mass flow rate[5]. Whereas the differential pressure flow meters still enjoying widespread industrial use also entail a determination of density, the thermal mass flowmeter directly supplies the sought quantity. Compared with simple and robust throttle elements such as orifices and nozzles, which can withstand adverse operating conditions, a disadvantage of the thermal mass flowmeter is that it needs to be carefully protected from gas pollution by means of a filter with a pore diameter of less than 5 micrometers. The device used allows the measuring of hydrogen flow rates up to 400 standard cubic meters per hour (scm/h) with an accuracy of ± 0.2% of full scale.

Fig. 2
Step-cut piston rings made of a brittle polymer blend and exhibiting fracture of the overlapping joints

Fig. 3
Avoidance of failure by fracture through a large radius at the transition to the step-cut joint (A = 1 mm, B = 2 mm, C = 4 mm)

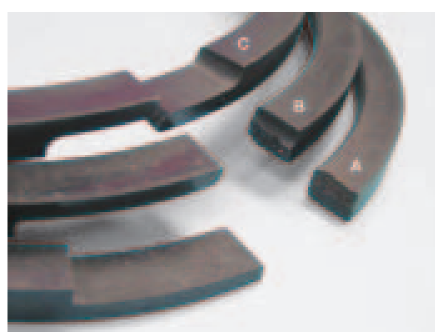
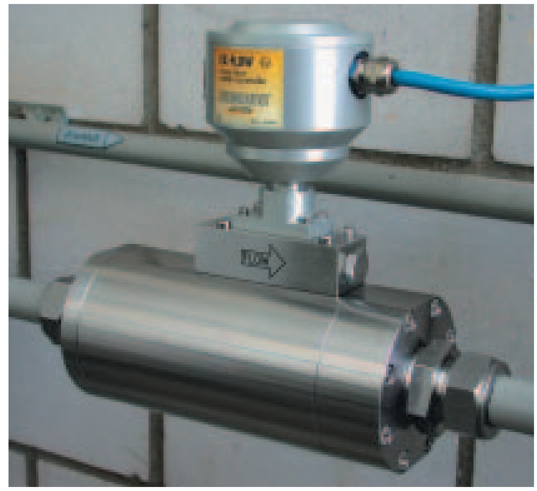


Fig. 4
Unequal wear of cover ring and sealing ring made of high-temperature polymer (right), compared with a new twin ring (left)

Fig. 5
Two-stage, crosshead hydrogen compressor for investigating the operational behavior of dry-running sealing systems

Fig. 6
Measurement of hydrogen flow rate by means of a thermal mass flowmeter for potentially explosive atmospheres

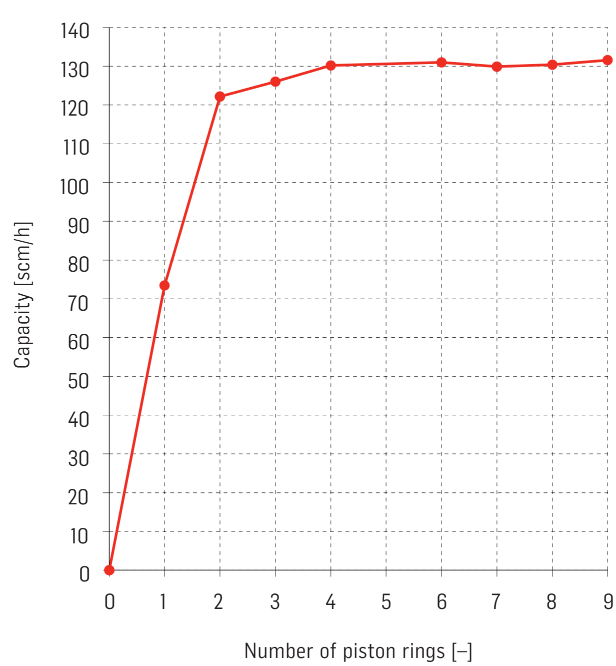


3.2 METHODOLOGY

The various piston ring designs also influence the pressure distribution inside the sealing system. In the case of new piston rings of a gastight design, the entire pressure difference is typically sealed by just one or two sealing elements at the ends of the sealing system[3, 4]. The pressure difference in designs where joint sealing is either absent or only present in the axial direction is distributed simultaneously to different extents among all the sealing elements[2]. For the latter, the first objective was to investigate the dependence of the capacity on the number of piston rings. Measurements involving different numbers of piston rings possessing a step-cut joint, single-acting compression from 2.4 to 6 MPa and a speed of 725 min^{-1} showed that the flow rate of new piston rings stops increasing once the number of sealing elements has reached four (Fig. 7). On dismantling of the piston, it turned out that all joints were aligned in a straight line in the presence of up to four piston rings, and in blocks of two or three sealing elements each in the presence of more rings. **Fig. 7**

In all tests, the true sealing rings were made of carbon/graphite-filled PTFE with an average coefficient of linear thermal expansion in the circumferential direction of 87 * 10^{-6} K^{-1} over a temperature range of 20 to 150 °C. In all designs, the piston rings had an axial dimension of 5 mm and a radial dimension of 6 mm (including an annular spring width of 1 mm in the case of the two-piece piston rings). The scarf joints of the one-piece piston rings cut at an angle of 30°, turned out to have an average clearance of 2.66 mm at room temperature. Given a radial piston clearance of 0.75 mm, this results in an area of 1.98 mm^2 left unsealed by the piston ring. The two-piece rings, whose scarf joints were also cut at an angle of 30°, turned out to have an average total clearance of 2.75 mm, resulting in a similar gap area of 2.04 mm^2 distributed among two joints. The average overlap of the two joint halves of the step-cut piston ring was 11.3°; the joint clearance of 1.93 mm results in an uncovered gap area of 1.43 mm^2 at room temperature.

Fig. 7
Dependence of capacity on the number of piston rings with a step-cut joint, given single-acting compression (p_s = 2.4 MPa, p_d = 6.0 MPa, n = 725 min^{-1})



CHAPTER 4

TESTS WITH DOUBLE-ACTING COMPRESSION

In the case of double-acting compression, the piston sealing elements are only subjected to the dynamic pressure difference p_{dyn} varying between the suction and discharge pressure, the static pressure component p_{stat} being omitted due to the identical suction pressure at both ends of the piston (refer to the notation for definitions). Double-acting pistons are designed with diameters ranging far beyond 1000 mm, the 75 mm variant used in these tests accordingly represents the low end of the diameter spectrum. However, of particular interest here was the effect of the high dynamic pressure differences, a case where smaller piston diameters are usually employed.

The tests were intended to first demonstrate the differences between the behavior of new gastight twin rings and that of piston rings with a scarf or step-cut joint used frequently for double-acting pistons. Of each of these three variants, eight piston rings were investigated at a speed of 494 min^{-1}, the suction pressure being raised at a constant compression ratio of 3.5 in steps of 0.4 MPa from 0.4 to 1.6 MPa. For these tests, the second compression stage was eliminated by removing the valves. At every load step, the compressor was operated for approximately one hour in order to achieve thermally stable conditions, and then for a subsequent hour during which the capacity was measured at 30-second intervals in order to form an average value. The average flow rates are plotted as a function of the suction pressure. **Fig. 8**

Apparently, there are only small differences between the flow rates of piston rings with step-cut and scarf joints with slightly higher values for the step-cut joint. Only the twin rings were able to achieve a somewhat higher capacity, amounting to 7.5% compared with the piston rings with step-cut joints. However, it seems more likely that this better flow rate is attributable not to a lower gas leakage between the two compression chambers, but to an improved sealing efficiency of the first twin ring located right next to the compression chamber, which results in a lower clearance volume.

As the degree of wear increases, all piston ring designs without complete joint sealing experience a continuous increase in the flow area of the joint until the leakage becomes unacceptably high. Although piston rings with a step-cut joint prevent a direct flow through the growing cross-section of the gap, a flow around the joint overlap is nevertheless possible. If one half of the joint overlap is removed in order to simulate a fracture of step-cut joints occurring frequently in practice, this results in an average joint clearance of 12.72 mm for the eight piston rings. Fig. 9 shows the effects of the uncovered gap area – now grown to an average of 9.45 mm^2 – on the flow rate. **Fig. 9**

A drastic reduction to roughly 40% in the case of the twin ring design was accompanied by a continuous rise in the outlet temperature to a peak value of 187 °C during the last load step. If a single, intact piston ring is used in the central section of the piston though, the capacity rises again and the temperature assumes the value of approximately 135 °C common during standard operation.

Fig. 8
Dependence of capacity on piston ring design in the case of double-acting compression

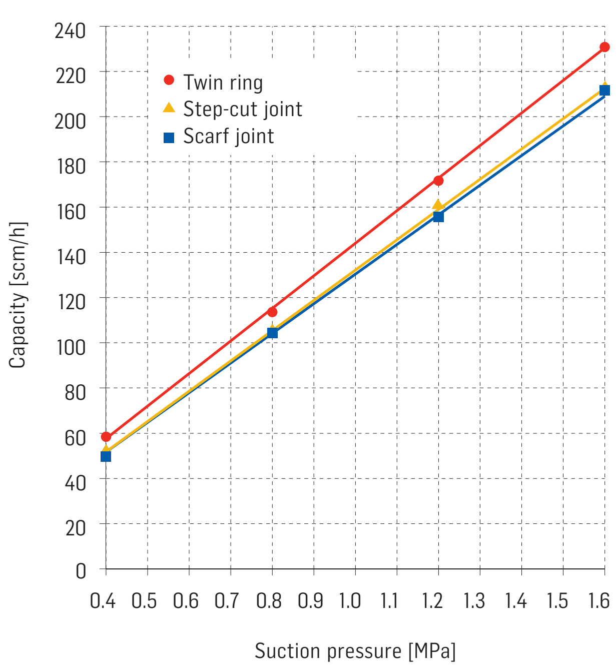
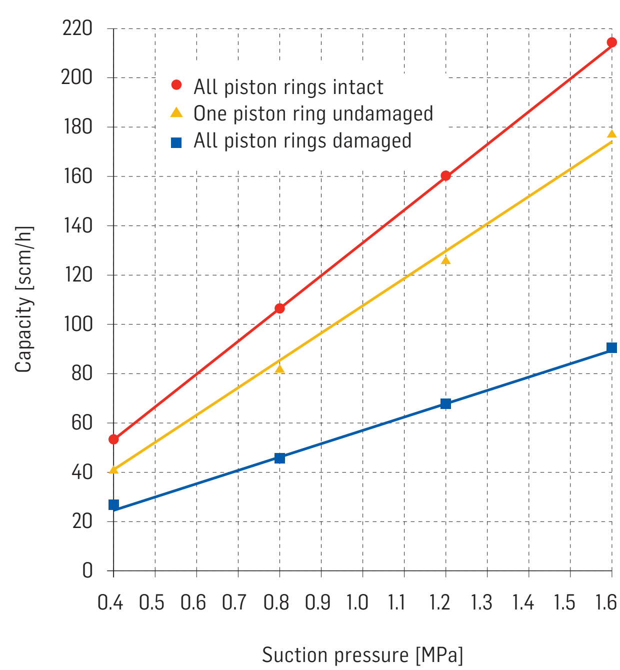


Fig. 9
Dependence of the capacity on the state of the step-cut joints of eight piston rings in the case of double-acting compression



This highlights the advantage of maintaining the best possible joint sealing also during double-acting compressions of hydrogen, even if the resultant benefits are hardly apparent in the new state of the piston rings. The twin ring proves advantageous here, since its sealing element is designed to wear completely in theory without permitting a direct flow through the joint.

CHAPTER 5

TESTS WITH SINGLE-ACTING COMPRESSION

If the pressure after the last sealing element of a single-acting piston is lower than the suction pressure of the compression stage under consideration, the dynamic pressure component p_{dyn} is supplemented by a static pressure difference p_{stat}. In contrast to the dynamic pressure difference, the static pressure component remains effective at a constant level during the entire rotation of the crankshaft, and a backflow into the compression chamber can be ruled out; consequently, the static pressure difference constitutes the primary load parameter influencing the leakage rate, therefore placing the highest possible demands on sealing technology. Table 1 shows the individual load steps and related pressure components for the tests involving the single-acting piston. **Table 1**

With nine piston rings in each case designed as twin rings, one-piece rings with a scarf joint, one-piece rings with a step-cut joint and two-piece rings with scarf joints, the flow rates were measured at various load steps and at a speed of 494 min^{-1} over a period of 1 hour. The average values are plotted as a function of the suction pressure in Fig. 10. As with double-acting compression, there is a small difference between the designs comprising scarf joints and step-cut joints, the values for the scarf joints being lower by 4.9% on average. The higher capacity in the case of piston rings with a step-cut joint is attributable to a slightly lower joint clearance (see chapter 3.2) and maybe to a somewhat better sealing efficiency due to the overlap, but compared with the gas-tight twin rings the flow rate is 16% lower. **Fig. 10**

Of note are the poor values obtained for the two-piece design, although the total gap area of its two scarf joints is only slightly larger than that of the one-piece variant's scarf joint, and although the two ring segments were pressed additionally against the cylinder wall by an annular spring. Consequently, the fact that the flow rate of the two-piece rings is 30.5% lower than that of the twin rings must be due to other disadvantages of this design, such as unstable and uneven contact between the segments and the cylinder wall, for instance. Removing the annular springs causes the flow rate to fluctuate strongly and drop as far as 52.3% of the twin ring level.

CHAPTER 6

TESTS WITH PROGRESSIVE WEAR

As notable variations in capacity were established already on new piston rings of various designs during single-acting compression of hydrogen, the next step was intended to investigate the influence of progressive wear on flow rates with and without elaborate joint sealing. In the case of designs without joint sealing, of particular interest here was the state of wear in which gas transport is no longer possible. To attain this state-of-wear within a reasonable period of time, only four piston rings were subjected to a suction pressure of 2.4 MPa and a final pressure of 6.0 MPa at a speed of 725 min^{-1}. As numerous preliminary experiments had revealed that the selected carbon/graphite-filled PTFE exhibits unfavorable wear characteristics in dry hydrogen, the subsequent test phases were also expected to be correspondingly short.

Fig. 11 shows the relative flow rates of piston rings with a scarf joint as a function of time, with respect to an initial value of 136 scm/h. After dropping sharply at the beginning of the experiment, the flow rates leveled off somewhat; after just 480 hours,

Table 1
Pressure exerted on the piston rings during single-acting compression

Suction pressure p_s [MPa]	1.2	2.4	3.6	4.8
Discharge pressure p_d [MPa]	3.0	6.0	9.0	12.0
Constant pressure after the last sealing element p_z [MPa]	0.4	0.8	1.2	1.6
Dynamic pressure difference p_{dyn} [MPa]	1.8	3.6	5.4	7.2
Static pressure difference p_{stat} [MPa]	0.8	1.6	2.4	3.2

Table 2
Average wear of individual piston rings after 480 hours of operation (p_s = 0.24 MPa, p_d = 6.0 MPa, n = 725 min^{-1}, piston ring no. 1 being located right next to the compression chamber in each case)

Design	Ring No. 1	Ring No. 2	Ring No. 3	Ring No. 4
Piston rings with a scarf joint [mm]	0.76	0.79	0.82	1.12
Conventional twin rings [mm]	0.69	0.51	0.61	1.15
Form-fit coupled twin rings [mm]	0.73	0.32	0.30	1.76

Fig. 10
Dependence of capacity on piston ring design during single-acting compression

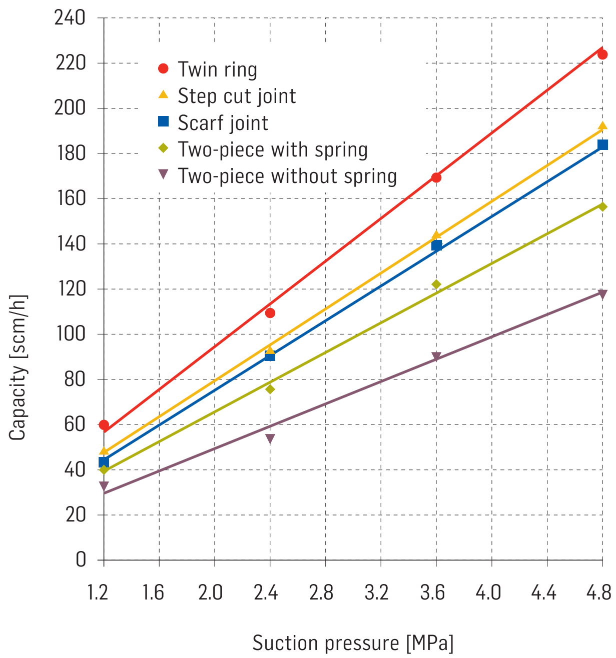
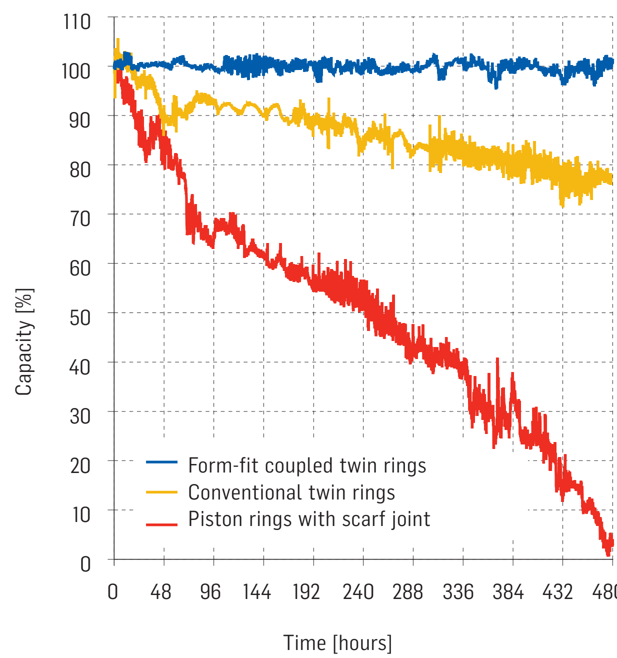


Fig. 11
Relative capacity vs. time of four piston rings with a scarf joint, conventional twin rings and form-fit coupled twin rings (measurement interval: 300 seconds)



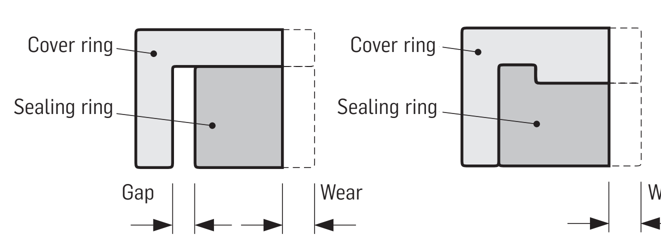
however, a state was reached in which the specified final pressure could still be maintained, but without any significant flow rate.
As mentioned earlier, all the sealing elements of piston rings without fully sealed joints participate in sealing pressure, but to different extents. Table 2 shows the average values for the radial wear of four sealing elements, each obtained by five measurements along the circumference of the sealing rings. The wear values rise from the first sealing element (positioned right next to the compression chamber) to attain a maximum value of 1.12 mm at the last sealing element. **Fig. 11, Table 2**
At room temperature, this results in an increase in the gap area of the joint from 1.98 to 7.04 mm². Although thermal expansion reduces this value slightly during operation, the indefinite temperature after the last sealing element is not likely to be higher than 40 to 60 °C. On the other hand, the uneven removal of material along the circumference of the sealing element and deformations in the region of the joint suggest the existence of additional flow areas during operation.

During single-acting operation, the flow rates of new twin rings were considerably higher than those of piston rings with a scarf joint. However, a test of four twin rings under the above-mentioned conditions reveals a gradual drop in the capacity to only about 3/4 of the initial value after 480 hours. A comparison between the wear rates of the two piston ring designs is only possible to a restricted extent, due to the unknown influence of the cover ring made of a high-temperature polymer.

In the case of the twin rings, the highest wear of 1.15 mm was also established for the last sealing element (Table 2), although the related cover ring only exhibited an average wear of 0.49 mm. Already described earlier on, this unequal wear of the sealing ring and cover ring gives rise during operation to a gap which contributes significantly to the drop in the flow rate. **Table 2, Fig. 12**

If gap formation is prevented through form-fit coupling between the sealing ring and cover ring (Fig. 12, 13), the capacity can be maintained at a value of 164 scm/h – corresponding to the flow rate of new twin rings – for the entire duration of the test (Fig. 11). **Fig. 11 and Fig. 12, 13**

Fig. 12
In contrast to the form-fit coupled design (right), unequal wear of the sealing ring and cover ring making up a conventional twin ring results in the formation of a gap (left), thus impairing the sealing efficiency



The large differences in the wear rates of the coupled twin rings are typical of gastight sealing elements which, in their new state, seal the entire pressure difference with just one or two sealing elements[3, 4]. The concentration of the dynamic and static pressure components at the two ends of the sealing system results here in a maximum value of as much as 1.76 mm at the last sealing element, a stable, high capacity being ensured by the outstanding wear compensation properties. A distribution of the pressure difference among several sealing elements, fundamentally beneficial to the life of a sealing system and exhibited by piston rings with a scarf joint, does not prove to be of any practical use for single-acting compression of hydrogen, in view of the disastrous consequences for the capacity.

Fig. 13
Form-fit coupled twin piston ring (patent pending)

CHAPTER 7

SUMMARY

Tests involving a dry-running compression of hydrogen using piston rings of various designs have shown that thorough joint sealing proves useful in maintaining a high volumetric efficiency for long operating periods, even in the case of double-acting pistons subjected to high loads. Although hardly any differences are evident in the new state, wear of designs without joint sealing leads to a drop in the capacity, accompanied by a rise in temperature, making it necessary to replace piston rings exhibiting a relatively low degree of radial wear, while designs with joint sealing allow a much longer utilization of the available ring thickness. The twin ring proves advantageous here, in that its sealing element is designed to wear down completely in theory without permitting a direct flow through the joint.

As observed on piston-rod sealing systems, the static pressure difference in the case of single-acting pistons constitutes the load parameter which mainly influences the leakage rate; this pressure difference needs to be sealed carefully to ensure a high volumetric efficiency despite progressive wear. Notable deviations become evident here between the various designs already in their new state. The two-piece design, in particular, exhibits very unfavorable characteristics comprising a low, extremely fluctuating flow rate, despite the fact that its total joint clearance is comparable with that of the one-piece design.

Especially in the case of single-acting pistons, wear compensation is extremely important for minimizing the flow area of the joint with progressive wear. Here, the sealing efficiency of the conventional twin ring deteriorates gradually because the distribution of the pressure difference along the frictional sealing surface results in unequal wear of the sealing ring and cover ring, thus creating gap between the two ring parts. Form-fit coupling between the sealing ring and cover ring makes it possible to maintain the capacity at the high level provided by gastight designs, thus optimizing utilization of the radial ring thickness.

Advantages offered by piston ring designs without joint sealing during compression of gases with a high molecular mass, such as resistance to creep and fracture, and a distribution of the pressure difference among several sealing elements, are cancelled in the case of hydrogen due to the sharp drop in the capacity to unacceptably low levels.

Notation

p_s suction pressure [MPa]
p_d discharge pressure [MPa]
p_z constant pressure after the last sealing element [MPa]
p_{dyn} dynamic pressure difference: $p_{dyn} = p_d - p_s$ [MPa]
p_{stat} static pressure difference: $p_{stat} = p_s - p_z$ [MPa]
c_m average piston velocity [m/s]
n speed [min^{-1}]
PTFE polytetrafluoroethylene

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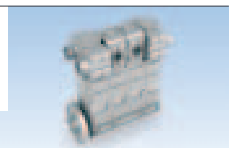

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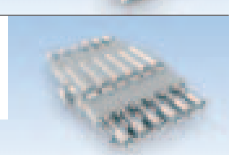

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