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DEVELOPMENTS IN DRY RUNNING SEALS FOR RECIPROCATING COMPRESSORS

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ABSTRACT

Reciprocating compressors of the crosshead design have sealing elements around the piston and piston rod. These seals can be of lubricated or nonlubricated design. This paper will give details about our R&D-work for dry running piston rod sealings or packings for high pressure applications. Starting with an overview of packing design some details are given about tribological tests. Measurements within packings give informations about the physics in this special dry running sealing element. Calculations of temperatures in the contact area between a dry running sealing ring and piston rod point to the main problem to be solved: The energy input due to friction. A new sealing ring design for packings is presented. Whith this new development we are able to seal a pressure difference of 100 bar across the packing and achieve a sufficiently long life.

INTRODUCTION TO THE PACKING DESIGN

Reciprocating compressors of the crosshead design must be sealed at piston and piston rod. These seals can be of lubricated or nonlubricated design. In the early 90's we started a R&D-project to develop dry running seals for high pressure applications. This paper will give details about our development work, especially for dry running piston rod sealings or packings.

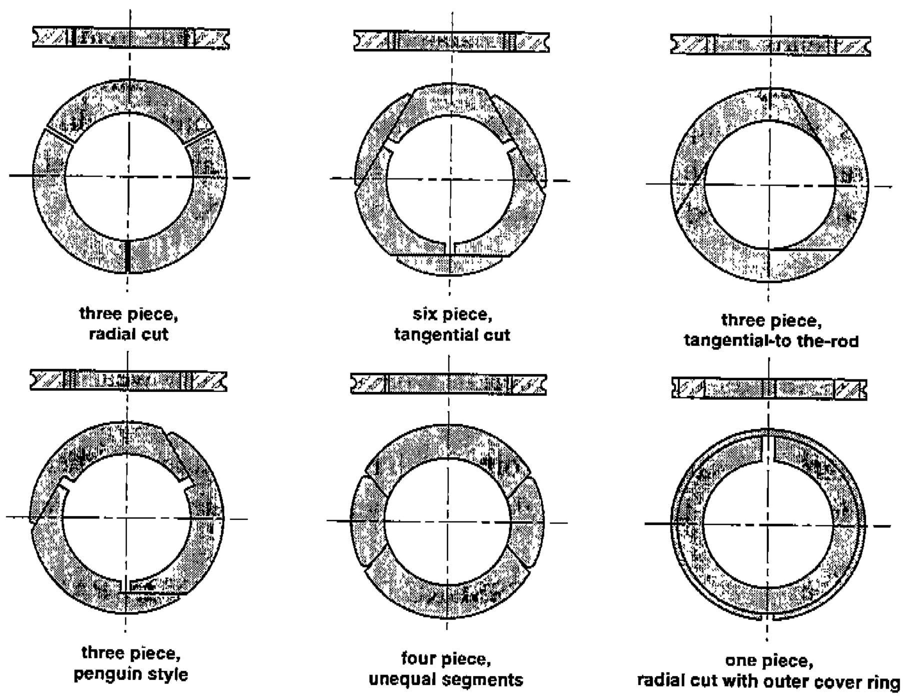

Fig. 1: Various styles of packing rings

Various styles of packing rings are known (Fig. 1). A standard ring for low pressure differences is the six piece, tangential cut ring. There are also other ring styles. A three piece ring with a cut tangent to the rod is often used. Another design is the penguin style ring, which avoids mechanical problems at the small tips of the other ring design. The four piece ring is an old design and not often used today. Still another design is a one piece radial cut ring with an outer cover ring, which seals radially and has a garter spring effect. All these rings need a secondary cover ring to seal the gaps in the axial direction. Therefore the three piece, radial cut ring is often used. There are also packing rings with gas tight joints on the market. Their advantage is that a cover ring is not needed.

A typical dry running packing for low pressure differences has a design as follows: Next to the compression chamber - the high pressure side - there is a pressure breaker which has to reduce the dynamic pressure load. Often a three piece, radial cut ring is used for this purpose. Our standard design has a three-ring-design as the sealing element. Three to six such elements are usual. The sealing ring is a six piece, tangential cut ring. It is covered by a three piece, radial cut ring. With these two rings we have a sealing effect in the axial and radial directions. An anti-extrusion-ring (mostly made of metal) is necessary because the strength of PTFE is low. Temperature is one of the main parameters we have to control in dry running sealings. Especially in the packings, this is the biggest problem because dry running materials have a very low thermal conductivity. Therefore we have to improve the heat removal from the packing when the pressure difference across the packing is increased. This is done by cooling.

TRIBOLOGICAL TESTS

Sealing rings are subject to wear. One of the main parts in our work was to find solutions to manage this wear in order to achive a sufficiently long life of the dry running sealing rings. The wear progress depends mainly on the load parameters. The allowable load parameters have to be measured as a function of dry running materials and environment. This gives the basis data for the design of dry running sealings, with an upper limit of the allowable values of pressure difference, average piston velocity and temperature for a chosen material. These measurements are performed in a specially designed tribometer. Materials with good results are then tested in compressors.

We have to be very careful in selecting the right material for a given application. Measurements of wear over time for different dry running materials when used for compressing a bone dry gas gave the following results: There are materials which have very low wear rates and others which have very high values. For instance for one filled PTFE we got low wear rates and for another filled PTFE we got very bad results. The same we have found with high temperature polymers. This shows that there is a need to have a lot of know-how when selecting a dry running material for given conditions.

PRESSURE DISTRIBUTION IN PACKINGS

We wanted to know more about the physics in the packings. For this reason a special test compressor was designed and built. It is a compressor with one cylinder and a tailrod. With this design it is easy to test different packing designs and measure the dynamic processes within the packing.

Without detailed information we assume a linear pressure reduction in the packings. Each sealing element reduces the pressure by a constant value. Then the energy input given here by the product of pressure and average piston velocity "pv" is equal for all rings and below an allowable value. A measurement gives different results. In Fig. 2 the measured pressure distribution is plotted over the number of sealing rings and crank angle. Often only the first and the last ring are loaded by pressure difference. All other rings don't work until these two rings are worn out.

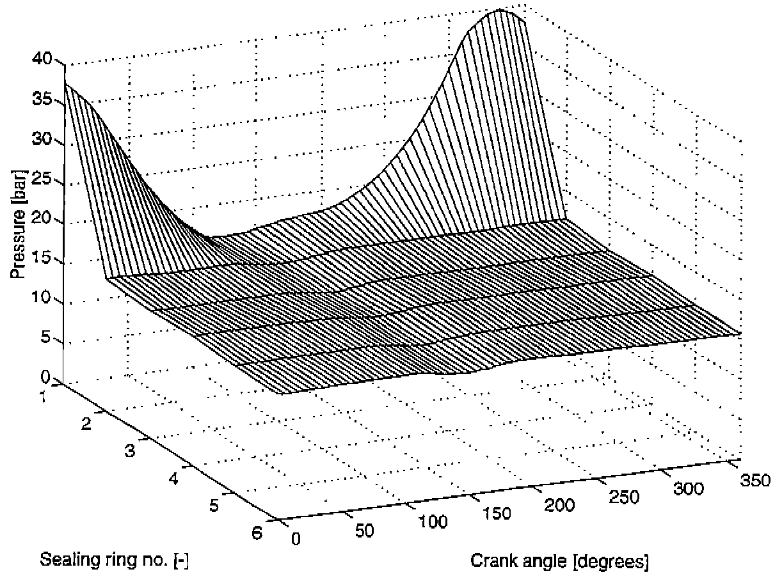

Fig. 2: Typical pressure distribution in a dry running packing

TESTING OF DIFFERENT PACKING DESIGNS

In testing various packing designs at the same conditions (pressure difference, compressor speed, etc.) we wanted to know the different sealing effects and other physical differences and develop an optimized packing. In Fig. 3 the average wear is plotted vs. different packing ring designs. All packings were tested for the same period of time. S1 is our standard design as I have explained before. Except the packing marked by S5 all others are known designs. S5 is our newly developed packing.

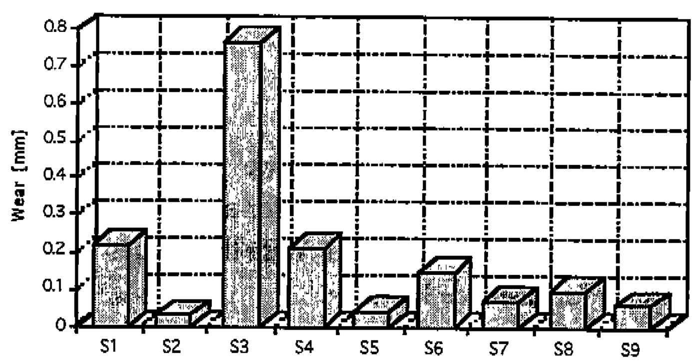

Fig. 3: Average wear of different packing designs (p_s = 16 bar, p_d = 40 bar, v = 3.2 $^{m}/_{s}$)

The measured leakage is shown in Fig. 4. Some known designs have very low wear rates at a sufficient low leakage. Our new design (S5) has the lowest leakage of all tested packings. We could reduce the leakage of our standard design (S1) to one fifth. The wear rate of this new ring design has also the lowest value of all tested packings. Details about this new design are given later on.

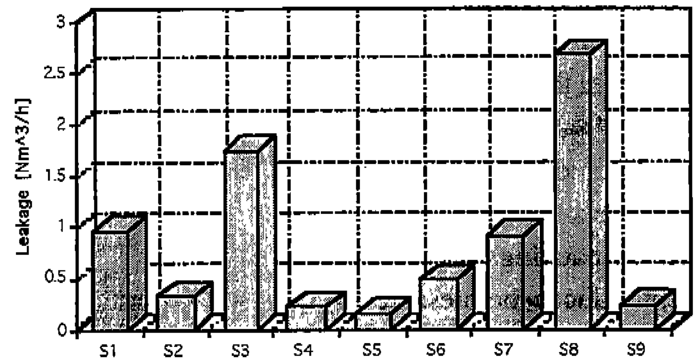

Fig. 4: Leakage of different packing designs (p_S = 16 bar, p_d = 40 bar, v = 3.2 $^m/_s$)

CONTROLLING TEMPERATURE WITH THE NEW DESIGN

The main part of our development work was to get more information about the heat energy production within the packing. With a computer program, which we wrote to calculate the physics in the packings, we got a helpful tool. Fig. 5 shows calculated temperatures when sealing a compressed gas (in this case nitrogen) with different ring styles at high loads in the contact area between ring and piston rod, and also the temperatures in the volume between two sealing rings. Only the last ones can be measured. With our standard design we got a temperature of nearly 400°C in the contact area between ring and piston rod. This is far too high for all known dry running materials. With an improved design we reduced this temperature to one third. With this values we are below the limits of dry running high temperature materials, for instance PEEK.

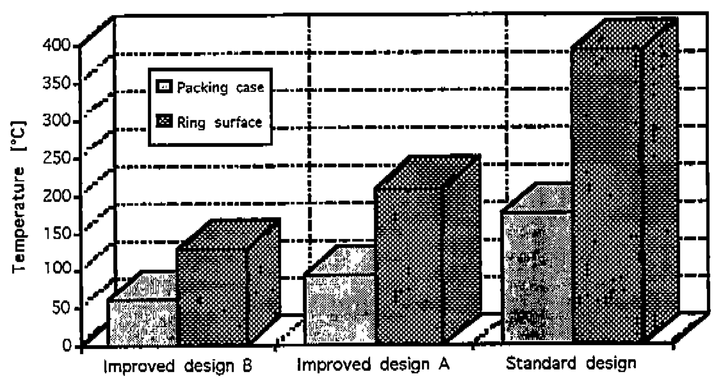

Fig. 5: Calculated temperatures for different ring styles

To verify these calculations we made measurements. The result is shown in Fig. 6. Without compression in the cylinder we got the increase of temperature in the packing by friction for different pressure differences. With the improved design the temperature is indeed only one third of that we meas-ured with the standard design. This confirms that our computer program calculates correct results.

The newly developed packing ring is explained in Fig. 7. It is a twin ring design with an excentric ring shape. This excentric design gives an excellent form adaptability of the ring to the piston rod because of a constant bending around the circumference. It is a gas tight design with a step cut joint at the L-shaped sealing ring and a cover ring with a garter spring. A high temperature polymer is required to control high pressure differences. Only with modern tools like Finite Element Analysis were we able to design the right geometry. To reduce the energy input by friction we optimized the contact surface as shown in the lower part of Fig. 7. With a small angle ß we got an additional effect: Because of the pressure difference over the ring, gas is forced into the contact area, creating a gas bearing. This is reducing the load and therefore also the temperature increase and wear.

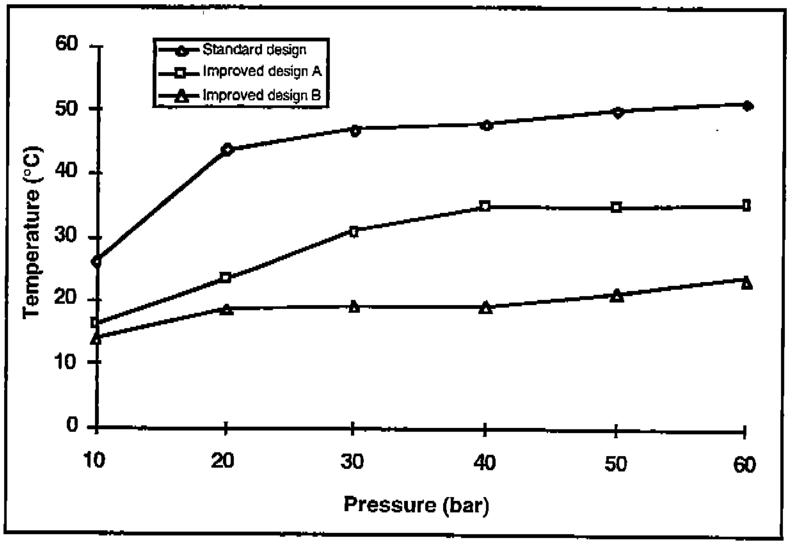

Fig. 6: Mean packing temperature for different ring styles

With this ring design we could realise a packing, which is sealing against a discharge pressure of 100 bar. We also improved our design of the pressure breaker. We reduce the dynamic pressure in more than one step, which decreases the load for each ring. The measured wear was extremely low. We got a sufficient life of the packing rings to obtain more than one year compressor running time. The length of this new packing is not more than that of our standard packing for sealing a maximum allowable discharge pressure of 40 bar.

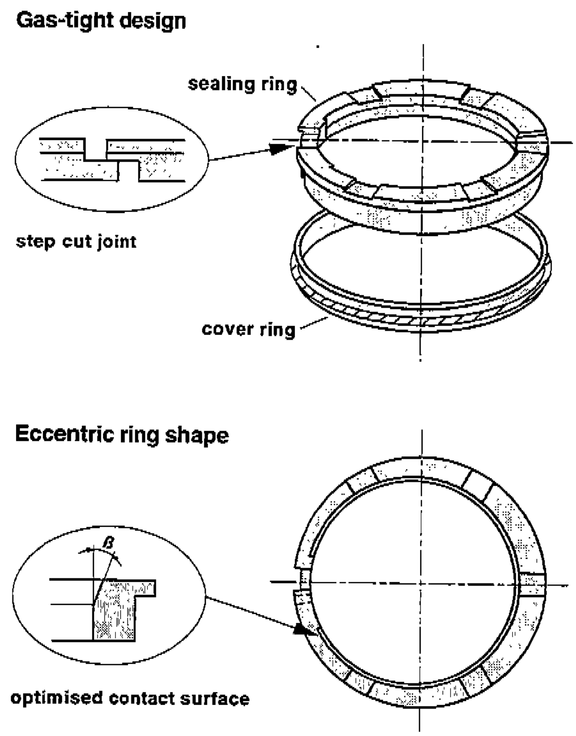

Fig. 7: Optimized packing ring based on the twin-ring design

CONCLUSION

Our goals in this R&D project were to master nonlubricated sealing element technology for pressure differences across the packing of at least 100 bar. The average piston velocity was to be increased to the values typical of lubricated reciprocating compressors.

A new design for piston rod sealings was shown. The temperature, which is the chief factor affecting the packing ring life, is the most difficult factor to control. The above mentioned goals were reached with new designs by using high temperature polymers. We have built up a high level of know-how about the physics within dry running seals. We are now able to manage such difficult tribological systems at high loads.