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(54) Title                      Spiral Groove Seal System for Sealing a High Pressure Gas

(72) Inventor                GLENN G. PECHT  
                              PAUL L. FELTMAN JR.

(73) Patent Granted to    JOHN CRANE INC., A CORPORATION ORGANISED UNDER THE LAWS OF THE STATE OF DELAWARE, UNITED STATES OF AMERICA, OF 6400 OAKTON STREET, MORTON GROVE, ILLINOIS 60053, UNITED STATES OF AMERICA.

# SPIRAL GROOVE SEAL SYSTEM FOR SEALING A HIGH PRESSURE GAS

This invention relates to seals for restricting leakage of a fluid about a shaft extending through a housing, such as a turbine or compressor, and in particular a gas under high pressure.

Prior to this invention, mechanical seals for sealing gases, such as hydrogen, at high pressures, for instance, exceeding 84.4 bar (1200 pounds per square inch) gauge, were of a type using two spaced apart mechanical end face seals. This system was of the "wet", contacting type which had to be cooled and lubricated by a buffer liquid pumped between the two spaced mechanical seals. The system for circulating the buffer liquid (frequently oil) included pumps, coolers and reservoirs which added complexity and expense. Further, the mechanical seals, even with their complicated support system, often had a short duration. Complete shutdown of the system is required to replace the seals.

Mechanical end face dry running gas seals operate without any lubrication. Seals of this type are shown and described in U.S. Patent No. 4,212,475, issued to Josef Sedy and assigned to the present assignee, incorporated by reference herein.

Another type of seal utilizes the teaching of U.S. Patent No. 4,212,475 to create a seal in equipment handling liquids under high pressures, such as liquefied light hydrocarbons. This type of seal creates shear heating of the sealed fluid between the seal faces of a sealing module of the type having spiral grooves described in U.S. Patent No. 4,212,475.

WO 88/09891 discloses a seal system for sealing a housing and rotatable shaft against leakage of high vapour-pressure liquids. The seal system contains axially spaced spiral-grooved seal modules mounted between the shaft and the housing. The upstream seal module functions to change the phase of the liquid, while a downstream module operates as a dry running gas seal.

One object of the present invention is to provide a dry gas end face seal arrangement which is particularly appropriate for use in equipment handling light gases under very high pressure, for example pressures which may exceed 126.6 bar (1800 p.s.i.) gauge, and to accomplish this by a breakdown or split of pressure across two or more such seals without need for external pressure control devices. The solution proposed by the inventors herein is a series of modules each having a pair of sealing faces, of which one seal face is of the spiral groove type, across which there is a pressure drop of predictable (calculated) proportion achieved by altered face geometry (e.g. groove dimension) in each module. Thus, for example, the pressure drop may be distributed in approximately equal proportions across two or more separate modules, thereby relieving seal stress or overloading without need for external controls.

Another object of the invention is to enable pressure breakdown, or splitting, to be easily and inexpensively accomplished simply by using spiral groove seal rings of substantially the same profile differing only in groove depth or dam width dimensions, a matter of etching rings of similar profile to afford successive stages of pressure step-down. There can be as few as two modules; there may be three, four or even five in tandem.

#### Summary of the Invention

The present invention is directed to a mechanical end face seal system which is particularly appropriate for compressors handling a gas under very high pressure. This is accomplished through the use of a seal arrangement preferably having a high pressure upstream module, and both an intermediate and a downstream seal module, each mounted on a shaft and housing associated with the compressor or pump. However, the principles of the invention may be incorporated

in an end seal arrangement having only two modules. Each seal module has a primary ring affixed to the housing and a mating ring affixed to the shaft. The rings have opposing, radially extending faces, one of which has a plurality of radially directed, circumferentially spaced spiral pumping grooves extending from one circumference thereof. The grooves of the upstream module exposed to the highest pressure have a predetermined groove restriction from the standpoint of groove dimension, which is considered optimum, while the grooves of each downstream module have a greater groove dimension than the upstream module, thereby imposing less restriction.

In a preferred form, decreased restriction is achieved by increasing the groove depth of the downstream module.

#### Brief Description of the Drawings

Fig. 1 is a cross section, axially, of a housing and shaft incorporating a preferred embodiment of the invention; and

Fig. 2 is an end view of a segment of one of the sealing rings of the preferred embodiment of the invention.

#### Description of the Preferred Embodiment

Fig. 1 illustrates a seal generally indicated at 10 constructed in accordance with the preferred embodiment of the invention. The seal is designed to seal gas under very high pressure, for example, hydrogen gas exceeding about 126.6 bar (1800 p.s.i.), in a pump compressor or turbine housing. Although the invention is at times described with reference to hydrogen gas as a fluid, it is to be understood that other gases or liquids under very high pressures can be handled with the appropriate changes and adjustments to the system to conform the essential parameters to the fluid being sealed.

A stuffing box or housing is indicated at 12 enclosing the seals and affording a shaft passage 14. A shaft 16 extends through the shaft passage from a high pressure chamber 18 in the housing 12 to the atmosphere at 19.

The seal assembly inside the housing is a tandem arrangement of axially spaced mechanical end face seals of the spiral groove type, each having opposed ring portions respectively secured to the housing 12 and shaft 16. The seals are generally of the type shown and described in U.S. Patent No. 4,213,475, referred to above. The tandem seal assemblies include a first upstream or inner seal module 20 located in the high pressure chamber 18, at least one second intermediate seal module 22, and, preferably, a third downstream or outer seal module 24 respectively located within annular intermediate and venting chambers 21 and 23 inside the housing.

Each seal module has a pair of annular rings comprising stationary primary rings 26A, 26B, 26C (ring stators) with radially extending sealing faces 28A, 28B, 28C. The primary seal ring elements are opposed to mating rings 30A, 30B, 30C having radially extending sealing faces 32A, 32B, 32C opposite the sealing faces, respectively, 28A, 28B, 28C of the primary rings. To prevent rotation, the primary rings 26A, 26B, 26C are secured to the housing by a retainer assembly, as will be described, while allowing limited axial movement. The mating seal rings 30A, 30B, 30C are affixed for rotation with the shaft 16 by a spacer sleeve assembly comprising a ring sleeve 33 and a shaft sleeve 34 keyed thereto, effectively locked to shaft 16 for rotation therewith as will be described.

The seal shown is exposed to a fluid to be sealed at the radially outer diameter of the seal components. The principles of the invention, however, are applicable to seals

having the higher pressure at the radially inner diameter of the relatively rotating seal faces.

5 The sleeve assembly, as noted, includes a shaft sleeve 34 which fits upon the shaft 16 and is fixed to the shaft by a clamping end plate 35 secured to shaft sleeve 34 by a bolt 36. Alternatively, a drive key arrangement (not shown) may be used to secure shaft sleeve 34 to the shaft 16. At the inboard or upstream end, an O-ring 37 provides a seal between the sleeves 33 and 34. There are additional O-ring  
10 seals requiring no elaboration.

The upstream or high pressure seal module 20, communicating with the high pressure chamber 18, includes an annular support flange 38, Fig. 1, which is an integral portion of sleeve 33. Pin 40 connects ring 30A for  
15 rotational drive to flange 38.

The sleeve assembly further includes spacer sleeves 48 keyed for rotation to sleeve 33. The sleeves have support flanges 50, holding pins 46 which engage and drive the mating rings 30B, 30C of the intermediate and outboard seal modules  
20 22, 24, respectively. O-rings 44 are interposed between the opposed faces of the support flanges 50 and the rotating seal rings. The spacer sleeves 48 extend to and engage the radial surfaces 32A, 32B adjacent the inside diameter of the mating rings 30A, 30B.

25 Alternatively, an arrangement which includes an annular split wedge in turn held by a clamping ring can be used, as is shown and described in the aforementioned WO 88/09891. The exact construction utilized to secure the rotary rings of the seal assemblies to  
30 shaft 16 is not of critical importance to the practice of the invention.

As noted above, sleeve 34 is held to shaft 16 for

rotation therewith by a drive plate 35. The ring sleeve 33 at its inner or upstream end bears against a radially projecting flange 52 on shaft sleeve 34. The ring sleeve 33 at its outboard end has a lock collar 53 mounted concentrically thereon. The inner end of the lock collar bears against the opposed radially inward circumferential surface of mating ring 30C. The outer end of the lock collar 53 has a head or flange 53H secured to the end face of ring sleeve 33 bolt 53B. By tightening bolt 53B the stack of mating rings and spacer sleeves is held in place. By tightening bolt 36, sleeve 33 is forced axially inward along sleeve 34 into locking engagement with flange 52. Thus, specific locks and keys, when sleeve 34 rotates, provide the drive for the mating rings 30A, 30B and 30C together with the related flange supports 50, to rotate therewith.

The retainer assembly for the primary rings of the seals comprises inboard, intermediate and outboard retainers 54, 55 and 56 connected one to another by cap screws 57 or otherwise in any suitable fashion. The inboard retainer 54 serves to support the inboard primary ring 26A. The intermediate retainer 55 supports the intermediate primary ring 26B and the outboard retainer 56 similarly supports the outboard primary ring 26C. Each retainer carries multiple springs 58 and a spring disk 60 which together urge the primary seal ring faces into engagement with the grooved faces of the mating rings. The disks 60 and springs 58 permit primary rings 26A, 26B and 26C to move axially in either direction. O-ring seals 61 provide a secondary seal between the primary rings and retainers 54, 55 and 56.

A gland plate 62 is connected to housing 12 by screws (not shown). The gland plate has a flange 64 engaging



the outer end face of the retainer 56. The retainer is connected to the flange by cap screws 66.

The transfer of gas at reduced pressure will be described in detail below but it may here be mentioned that a vent opening 68 communicates venting chamber 23 with a passage 70 in the housing exposed to atmospheric pressure, allowing gas released from chamber 23 to be transferred to a flare stack or other environment for capturing, using or disposing of the vented gas. Further, downstream seal module 24 acts to seal the small amount of gas which may leak from chamber 23; but because of the low pressure of the gas in chamber 23, leakage through seal module 24 will usually be negligible. Thus, the amount of gas transferring across the dam of this seal module is little (not collectible) and the final pressure at the I.D. may be close to nil. In this sense the third seal may serve as a safety in the event of an obstructed vent for example. The advantage of venting chamber 23 via vent passage 68-70 is to bring the noncollectible leakage around the shaft downstream of said module 24 to a minimum.

It will be noted that the I.D. and sloped radial surfaces of the retainers 54 and 55 are spaced from the opposed surfaces of the flange supports 50, thereby providing transfer passages 21P and 23P for gas leak flow (pumped) across the dam of seal module 20 into intermediate chamber 21 and for gas leak flow (pumped) across the dam of seal module 22 into chamber 23. A passage 72 to chamber 21 may be provided with a gage attached to be able to read the pressure in chamber 21.

Mating rings 30A, 30B, 30C each have a series of spiral grooves 92 (shown in phantom in Fig. 1) which will now be discussed in detail with reference to Fig. 2. Thus, Fig. 2 shows a portion of a mating surface on a mating ring

opposed to a primary ring. For purposes of description, the face 32A of mating ring 30A is shown in Fig. 2. The face has a plurality of circumferentially spaced, spiral grooves 92 extending radially from an open end at the outer circumference partially across the radial width of the face 32A. The ungrooved surface 94 defines a sealing dam or land which in cooperation with the opposed face of the primary seal ring provides a hydrostatic static seal when the shaft 16 is not rotating. Faces 32B and 32C of the seal modules 22 and 24 are similarly configured to present pumping grooves as 92 and lands 94.

When shaft 16 is not rotating, pressurized gas is sealed by a hydrostatic film between the rings at the sealing dam. When the shaft starts rotating, the grooves 92 pump gas radially inwardly across the seal faces causing the seal faces to open slightly.

As the gas is transferred across the dam of seal module 20 to passage 21P it enters chamber 21, and the operational parameters of the seal provide for a pressure drop from full pressure, e.g. about 126.6 bar (1800 p.s.i.) gauge in chamber 18, to approximately one-half of that pressure, or about 63.3 bar (900 p.s.i.) gauge in chamber 21, opening seal module 22. This is a virtually instantaneous pressure drop because the instant the upstream seal of module 20 opens against the closing bias of the related springs 58, gas is transferred at the same moment to chamber 21 through passage 21P; seal module 22 is likewise instantly opened and the gas pressure is instantly divided, relieving the high pressure in the high pressure upstream chamber 18. It should be kept in mind that while the pumped actual volume (leakage) of fluid across the dam of the second seal is larger than the first, at the actual pressure of the

module, the mass flow rate across each seal interface is constant at all times.

Intermediate seal 22 itself provides for a further pressure drop across the relatively rotating seal faces from about one-half of full pressure to a pressure slightly above atmospheric pressure. Preferably chamber 23 has a pressure of about 0.63 to 1.06 bar (9 to 15 p.s.i.) over atmospheric pressure, which is the result of the restriction presented by vent passage 68-70 exposed to atmospheric pressure. In fact, the third seal, and if used, may be present as a mere leakage collector of small capacity. As described above, this gas is vented through vent 68 for disposal in one form or another.

No seal can be expected to perform constantly in accordance with design limits without some change due to the rigors of performance, coupled with the possibility of mechanical and thermal imbalance not possible to predict or anticipate. Consequently, the third seal may perform redundantly, and if there is a change in pressure profile during service, the result will be a self-adjusting accommodation of pressure drop across the three seal faces as a back-up to either of the upstream seals. Regardless of whether there are two or three seals in tandem, no one seal is subjected to the full pressure load and operates at a level well below its design capability.

The above described system provides a number of unique attributes and advantages to presently known sealing systems. First and foremost, presently known mechanical end seals of the spiral groove type having the same geometry (I.D., O.D. and thickness) can be utilized by altering parameters of the groove design, such as groove depth and dam width, together sealing against much greater pressures than has heretofore been possible with a greatly increased seal

integrity at higher pressures. This results in a much reduced level of gas loss due to leakage. Specifically, the pressure split across each of the spiral groove type sealing modules can be achieved by components that can be converted  
5 from presently known configurations simply by etching the grooves to different depths in the preferred form of practice.

In comparison, trying to accomplish a split of pressure of the high magnitude here involved (e.g.  
10 126.6 bar or 1800 p.s.i. gauge) with other kinds of mechanical end seals would require external controls, considerable structural changes and additions of compensating elements because of the great pressure differentials involved. Greater bulk would be required in comparison to an end seal of the spiral groove  
15 type so employed under the present invention as to exploit the leak phenomenon at the I.D. The greater the bulk, the greater the chance for imbalance, there are more areas subject to wear, and so on.

Further attributes of this system are the ability  
20 to seal gases of much higher pressures than have heretofore been possible without the use of complex and elaborate external pressure control systems and for massive seal geometry.

These attributes can be readily perceived by  
25 assuming a pressure of 126.6 bar (1800 p.s.i.) gauge in chamber 18 and a shaft speed of 10000 rpm. If the spiral groove depth in the first seal module is 0.005 mm (200 microinches) and if this depth parameter is increased to 0.01 mm (400 microinches) in the second seal,  
30 the pressure drop across the first seal is reduced by approximately fifty percent to 63.3 bar (900 p.s.i.) gauge, relieving the first seal of considerable stress. For most applications, this fifty percent reduction is adequate. Nonetheless, in

like manner, when a third seal is present, the spiral grooves in the mating ring of that seal result in a further pressure drop across the relatively rotating faces of the second seal, and this reduction may result in virtually nil pressure in chamber 23 if vented (68-70) as described above.

A change in dam width can replace or supplement the groove depth parameter difference. There may be circumstances where two upstream seals (e.g. seal modules 20 and 22) are each to be at substantially full pressure and the third seal module becomes the one responsible for the pressure split. Various terms and analogies may be used to explain the phenomenon involved at the seal faces when splitting the pressure. The principle employed is that no seal is perfect, and we exploit that circumstance by employing a pair (at least) of tandem end face seals of the spiral groove type so that leakage may be exploited merely by varying groove dimensions without need to alter any other geometry of an inventory of such seals.

The total gas leakage across each seal, taken as a mass (lbm/min) is the same at each seal. Because the spiral grooves of the downstream seal present a larger volume than those of the upstream seal, the gas has more room for expansion within the larger groove volume and hence is of less density (occupying a larger actual volume) than the gas transferred thereto by the upstream seal grooves. Therefore, while it may be convenient to refer to different volumes (or pumping capacity) being pumped across the respective seal interfaces per unit of time (and at any instant of time), it is to be borne in mind the mass flow rate across each seal is

the same at all times.

Another and perhaps more apt description is that the seals may be considered flow restrictors (valves) in which the downstream seal, with deeper spiral grooves (in the preferred form) offers less restriction to gas flow than the one upstream, at the higher pressure. The pressure must therefore drop, as when opening the nozzle valve of a garden hose: the rate of water flow is the same as before, when the nozzle was less restrictive, but because the nozzle passage has been widened the stream of fluid now flows out at reduced pressure, covering more area but with less force compared to the more forceful stream reaching a greater distance (more pressure) which prevailed when the nozzle restriction was greater.

Whereas a preferred form of the invention has been shown and described, it will be realized that alterations may be made thereto without departing from the scope of the following claims.

## CLAIMS

1. A multiple, tandem, spiral groove seal system (10) for sealing along a rotatable shaft (16) extending through a housing (12) containing a process fluid under pressure, the system (10) comprising:

a first and a second axially spaced apart rotary mechanical end face seal module (20,22) defining with the housing (12) an intermediate chamber (21), each said seal module (20,22) having opposed generally radially extending sealing faces (28A, 32A, 28B, 32B) presented by primary rings (26A, 26B) secured to the housing (12) and mating rings (30A, 30B) rotating with the shaft (16), the sealing faces (32A, 32B) of each mating ring (30A, 30B) having a plurality of pumping spiral grooves (92) extending part-way across the face (32A, 32B) thereof inwardly from one ring circumference toward the center of the annular sealing face (32A, 32B) thereof, and means (58) for biasing the sealing faces (28A, 32A, 28B, 32B) of the respective seal modules (20, 22) toward one another;

said first seal module (20) being disposed adjacent said process fluid contained by said housing (12) and an outer annular circumference of the sealing faces (28A, 32A) being exposed to the process fluid and an inner annular circumference of said sealing faces (28A, 32A) being exposed to said intermediate chamber (21), said process fluid in said intermediate chamber (21) having a calculated pressure,

characterized in that

the process fluid passage from said housing (12) and into said intermediate chamber (21) occurs only across said sealing faces (28A, 32A) of said first seal module (20);

5 said second seal module (22) being adjacent to said intermediate chamber (21), an outer circumference of the annular sealing faces (28B, 32B) being exposed to the fluid in said intermediate chamber (21) and an inner circumference of said annular sealing faces (28B, 32B) being exposed to a venting chamber (23), and the fluid passage from said intermediate chamber (21) to the  
10 venting chamber (23) communicates only across said sealing faces (28B, 32B) of said second seal module (22).

2. A seal system as claimed in claim 1 wherein the respective primary rings (26A, 26B) of each seal module (20, 22) have substantially the same diameter and thickness geometry and the  
15 respective mating rings (30A, 30B) of each seal module (20, 22) have substantially the same diameter and thickness geometry and the spirral groove dimensions of the respective seal modules (20, 22) differ so as to produce a predetermined drop in pressure of said process fluid across said second seal module  
20 (22) which is approximately equal to a predetermined drop in fluid pressure across said first seal module (20).

3. A seal system as claimed in claim 1 wherein the spiral groove dimensions of the respective seal modules (20, 22) differ so as to produce a pressure of said fluid in said intermediate chamber  
25 (21) which is approximately equal to one-half of the process fluid pressure in said housing (12).

4. A seal system as claimed in claim 1 wherein the spiral grooves (92) of said mating ring (30B) of said second seal module (22) have a volumetric capacity which provides a reduction of about  
30 one half the fluid pressure across the sealing faces (28A, 32A)



of said first seal module (20).

- 5 5. A seal system as claimed in claim 1 wherein the venting chamber (23) is disposed between said second seal module (22) and a third seal module (24), and the venting chamber (23) is further vented to the ambient atmosphere (19) external of the housing (12) by a vent (68).
6. A seal system as claimed in any of claims 1 to 5 wherein the process fluid in said housing (12) is a gas under high pressure.
- 10 7. A multiple, tandem, spiral groove seal system substantially as hereinbefore described with reference to and as illustrated in the accompanying drawings.

F. R. KELLY & CO.,  
AGENTS FOR THE APPLICANTS.

