# United States Patent [19]
**McHugh**

[11] **3,770,179**

[45] **Nov. 6, 1973**


[54] **SELF-PRESSURIZING SEAL FOR ROTARY SHAFTS**


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[73] Assignee: **General Electric Company**, Schenectady, N.Y.



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[21] Appl. No.: **132,485**

[52] **U.S. Cl.** .......................... **277/2**, 277/61, 277/65

[51] **Int. Cl.** ........................... **F16j 9/00**, F16j 15/38

[58] **Field of Search** ........................ 277/2, 3, 61, 65



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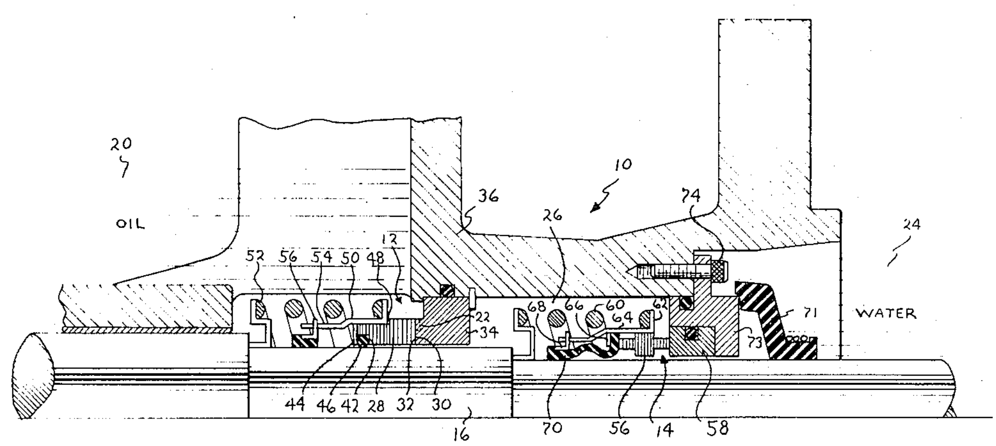
[57] **ABSTRACT**

A self-pressurizing shaft seal for an oil filled submersible motor is characterized by an inboard spiral grooved face seal and an outboard conventional face seal disposed in series relationship along the motor shaft. During operation, the inboard spiral grooved face seal pumps oil from the motor interior into a substantially confined zone between the seals to increase the oil pressure at the outboard face seal without the necessity for structurally strengthening the entire motor housing. Also disclosed is the disposition of a shoulder on the spiral grooved seal runner face remote from the pumping interface to permit the oil pressure within the confined zone to hydraulicly increase the axial force upon the runner thereby increasing the pumping pressure of the spiral grooved seal in boot strap fashion. Other disclosed seals contain means for measuring the pressure within the confined zone to actuate remote signaling devices upon a failure of the outboard seal as well as spiral grooved face seals having valving means to alter the pumping rate of the inboard seal upon a loss of pressure in the confined zone between seals.


**8 Claims, 7 Drawing Figures**



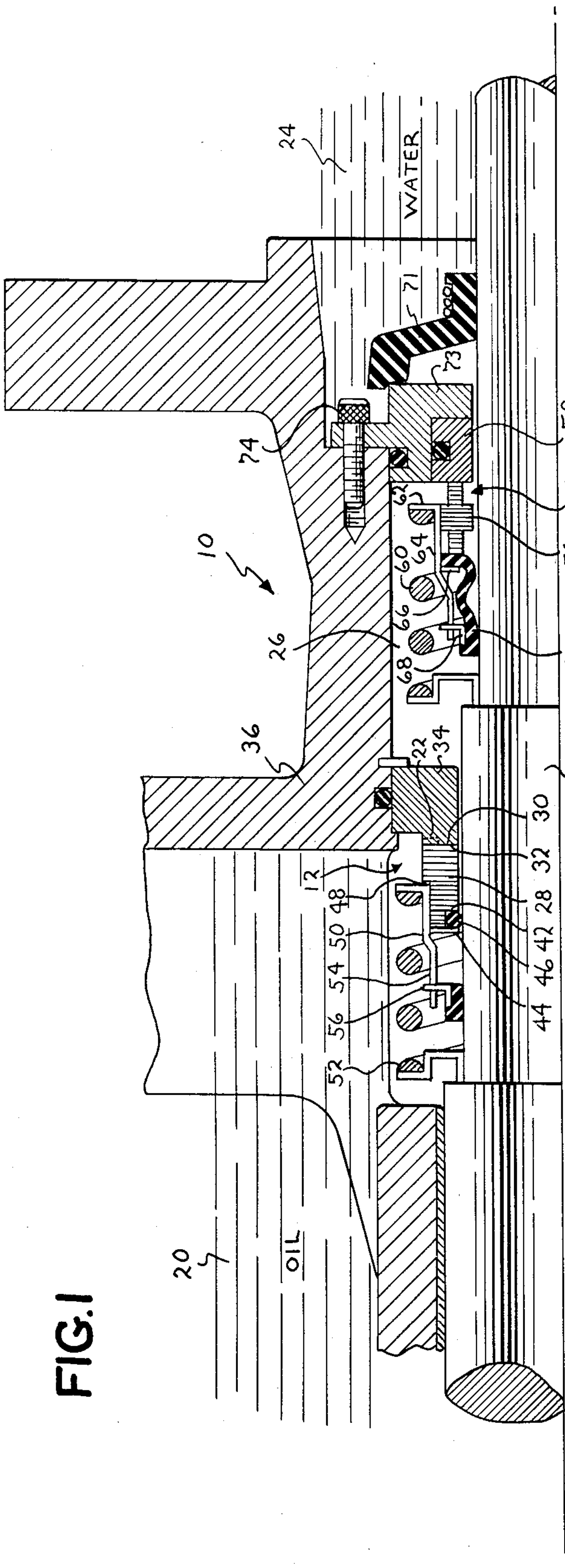



FIG.1

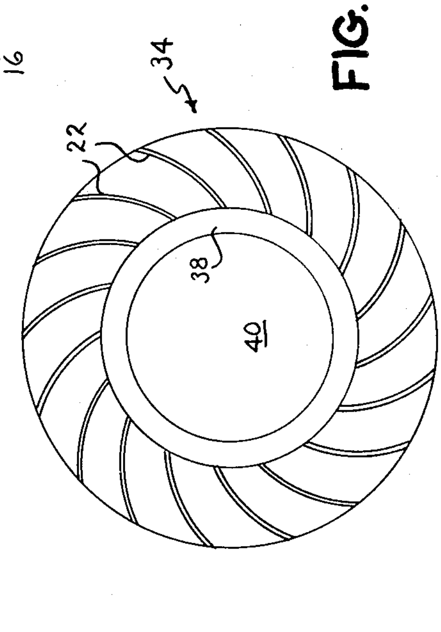


FIG.2

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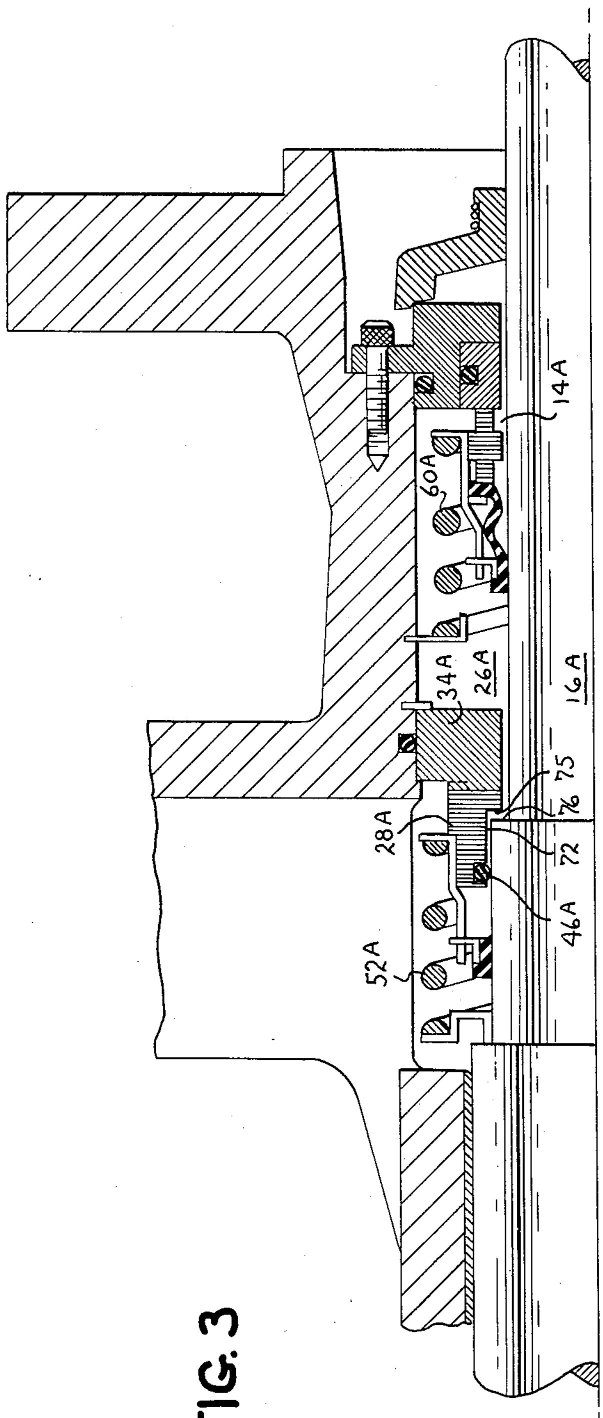
14A
60A
34A
26A
16A
75
76
72
28A
46A
52A

FIG. 3

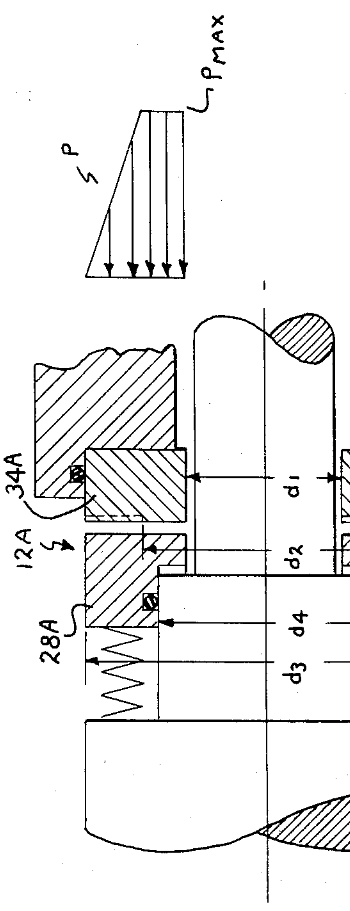
P
$P_{MAX}$
34A
12A
28A
$d_1$
$d_2$
$d_4$
$d_3$

FIG. 4

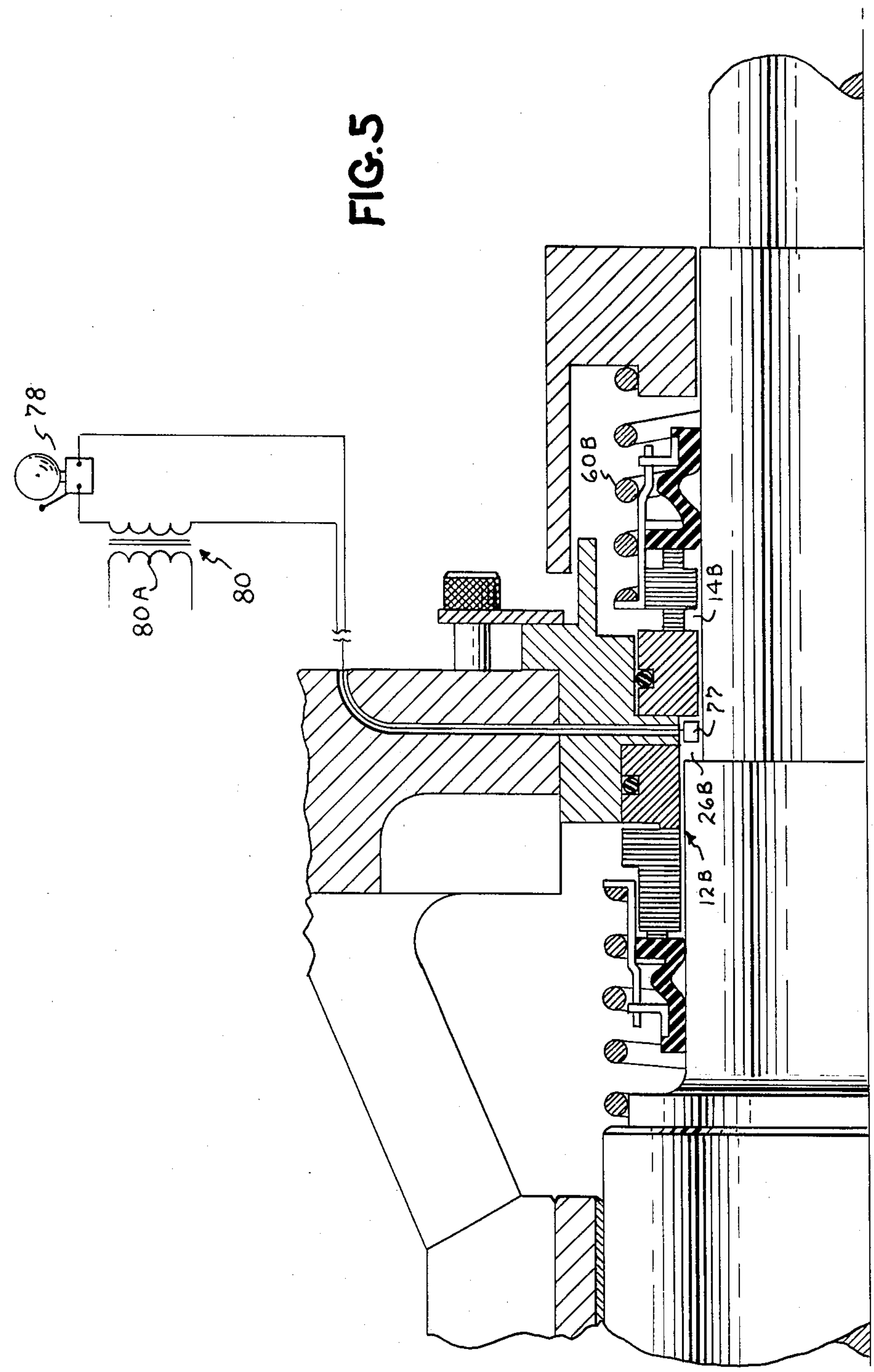


FIG.5

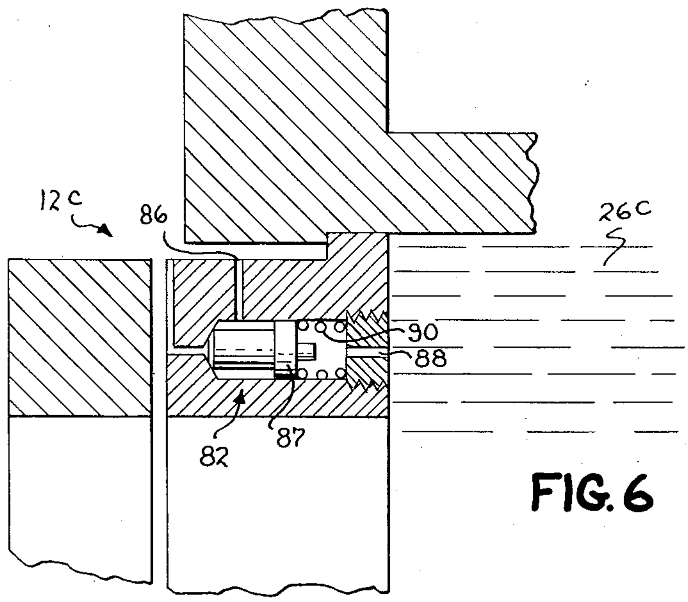


FIG. 6

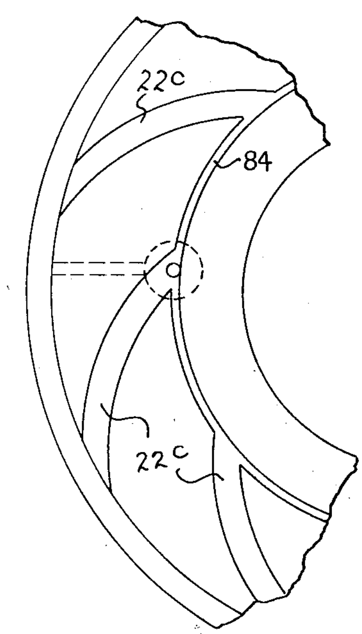


FIG. 7

## SELF-PRESSURIZING SEAL FOR ROTARY SHAFTS

This invention relates to a shaft seal for a rotatable machine and, more particularly, to a shaft seal wherein a high pressure zone of sealing fluid is formed between a spiral grooved face seal and a conventional face seal to inhibit ingress of contaminating fluid into the machine.

One of the major factors limiting the life of submersible motors is water in-pumping at the shaft seal produced by a slight eccentricity in the face seal customarily employed to assure minimum leakage at the shaft. Although water in-pumping can be overcome by substantially increasing the pressure differential between the sealing fluid, e.g., oil, typically contained within the motor and the ambient water, higher pressure differentials necessarily require reinforcement of the motor housing as well as substantial alterations in the spring biased diaphragm customarily utilized to produce the oil/water pressure differential.

Because of the difficulties associated with increaisng the oil pressure within the motor, a number of different seal configurations have been proposed to inhibit in-pumping notwithstanding a low oil/water pressure differential. For example, rotor shafts have been sealed utilizing an external pump to produce high and low pressures within sealing chambers situated at axially displaced locations along the shaft. Similarly, it has heretofore been proposed that the shaft of a centrifugal pump be sealed utilizing the rotary speed of the shaft to pump oil from an axially outboard location to an inboard seal to restrict gas leakage from the pump. I also have proposed in my co-pending U.S. Pat. No. 3,704,019, issued Nov. 28, 1972, and assigned to the assignee of the present invention utilization of a spiral grooved face seal having deep helical grooves to increase the pressure at the seal interface without increasing the outpumping rate of oil from the motor. While all these designs have certain advantages, there still remains a need for seals of different designs with differing capabilities.

It is therefore an object of this invention to provide a novel self-pressurizing seal characterized by low leakage.

It is also an object of this invention to provide a seal adaptable to monitoring at an external location to assure proper seal functioning.

It is a further object of this invention to provide a self-pressurizing seal wherein the pressure of the sealing liquid within the seal is employed to augment the mechanical bias of the seal thereby maximizing the obtainable pressure from the seal without extensive wear of the seal during start-up.

It is a still further object of this invention to provide a self-pressurizing seal wherein automatic closure of the seal is effected upon a reduction in seal pressure.

These and other objects of this invention generally are achieved by a self-pressurizing seal for a rotatable machine characterized by an inboard pumping seal having an annular running member mounted upon a rotatable shaft in juxtaposition with an annular co-planar stationary member. At least one of the juxtaposed members is provided with spiral grooves extending from the perimeter of the member to a land along the planar face of the member to pump sealing fluid from the rotatable machine into a substantially confined zone. The pumping action of the inboard seal increases the pressure of the sealing fluid within the zone relative to the sealing fluid pressure within the machine and conventional face seal means are disposed along the shaft at an axially outboard location (relative to the inboard pumping seal) to restrict the flow of sealing fluid from the high pressure zone into the ambient water. Because sealing fluid at relatively high pressure is situated only within a zone intermediate the axially displaced face seals, ingress of water into the motor is inhibited without structural reinforcement of the entire motor housing and without subjecting the necessary flexible oil expansion system to large pressure differences.

Although the features of this invention are defined with particularity in the appended claims, a more complete understanding of the invention may be obtained from the following detailed description of various specific embodiments when taken in conjunction with the appended drawings therein:

FIG. 1 is an enlarged sectional view of a self-pressurizing seal in accordance with this invention,

FIG. 2 is a plan view of one member of the inboard face seal illustrating the disposition of spiral grooves therein,

FIG. 3 is a sectional view of a self-pressurizing seal wherein the hydraulic pressure of the oil within the seal is employed to increase the obtainable seal pressure,

FIG. 4 is an enlarged sectional view illustrating the force distribution along the spiral grooved seal of FIG. 3,

FIG. 5 is an alternate seal configuration illustrating a seal monitoring device in accordance with this invention,

FIG. 6 is a sectional view of a spiral grooved seal wherein the outpumping rate of the seal is reduced upon a reduction in the outboard seal pressure, and

FIG. 7 is a plan view of the stationary member forming the seal of FIG. 6.

A self-pressurizing seal **10** in accordance with this invention is illustrated in FIG. **1** and generally includes an inboard spiral grooved face seal **12** and an outboard face seal **14** disposed in tandem upon shaft **16** of a dynamoelectric machine, e.g., the pump motor such as is described in U.S. Pat. No. 2,790,916, issued Apr. 30, 1957 to M.B. Hinman (the entire disclosure of which patent is incorporated herein by reference). Typically, the pump motor contains a sealing fluid, e.g., transformer oil **20**, biased by a flexible diaphragm to increase the pressure of the oil approximately 5 psi relative to the water **24** which forms the ambient environment for the motor during operation. The oil within the motor is in communication with the radially outer surface of inboard seal **12** and is pumped by the spiral grooves of the inboard face seal into substantially closed annular oil chamber **26** thereby increasing the oil pressure of the chamber relative to the oil pressure within the pump motor.

Spiral grooved face seal **12** generally is characterized by an annular carbon runner **28** mounted upon shaft **16** with planar face **30** of the runner being disposed in a confronting attitude with planar face **32** of ceramic stationary member **34** fixedly secured to pump motor housing **36**. One of the planar faces of face seal **12**, illustrated in FIG. **1** as face **32** of stationary member **34**, has spiral grooves **22** therein to pump oil from the motor upon rotation of runner **28** relative to stationary member **34**. The grooves, shown more clearly in FIG.

2, have a geometric configuration and density dependent upon the quantity of pumping desired by the face seal (as will be more fully explained hereinafter) and desirably extend radially from the outer circumferential edge of annular stationary member **34** to an annular land **38** separating the grooves from central aperture **40** extending axially through the member. In the event a failure of outboard face seal **14** should necessitate a shutdown of the motor, land **38** advantageously functions to block back flow of water through spiral grooved face seal **12**.

Returning again to FIG. **1**, the face of carbon runner **28** remote from planar face **30** is notched to form a lower shoulder **42** which, in association with backing plate **44**, serves to house O-Ring **46** sealing the carbon runner to shaft **16**. A second shoulder **48** also is formed along the radially outer face of carbon runner **28** to seat a generally L-shaped brass ferrule **50** biased against the runner by spring **52**. To permit axial movement of the ferrule along shaft **16** while restricting movement of the ferrule in a plane perpendicular to the shaft, elongated body **54** of the ferrule is slidably engaged within a guide **56** fixedly secured to the motor shaft.

Outboard face seal **14** is conventional in design and is mounted in tandem with spiral grooved face seal **12** so that the pressure of the oil within chamber **26** tends to close carbon runner **56** upon confronting ceramic stationary member **58**. A biasing spring **60** augments the oil pressure tending to close the face seal by providing an axial force against upper extension **62** of ferrule **64** to drive inwardly extending backing plate **66** toward carbon runner **56**. The edge of ferrule **64** proximate spiral grooved face seal **12** extends through guide **68** to limit the axial movement of the ferrule while sealing of the runner to the shaft is accomplished by a flexible bellows **70** fixedly secured between the shaft and the overlying ferrule.

To inhibit ingress of solid contamination into the motor, a sand slinger **71** is secured to motor shaft **16** at an axial location to shroud the radially outer edge of outboard seal carrier **73**. The seal carrier is fixedly mounted to the motor housing **36** by bolts **74** passing through suitable apertures in the outer flange of the seal carrier while a radially inner notch in the seal carrier serves as a seat for stationary member **58** of face seal **14**.

During operation of the motor, the rotary motion of carbon runner **28** relative to spiral grooved stationary member **34** pumps oil from the motor housing into annular oil chamber **26** to increase the oil pressure within the chamber to a predetermined level dependent primarily upon the anticipated water inpumping force at outboard face seal **14** resulting from eccentricity in the outboard seal. This predetermined pressure level can be calculated (in accordance with the teachings of an article entitled "Inward Pumping in Mechanical Face Seals, by J.A. Findlay, presented as paper No. 68 at the Lub 2 ASME-ASIE Lubrication Conference, Atlantic City, N.J., Oct. 1–10, 1968,) from the formula:

$$\Delta p/e = 3\epsilon \cos\alpha \cdot \mu\omega \; (R_o - R_i)/h^2 \; (1+1\cdot 5\epsilon^2) \tag{1}$$

wherein

$\Delta p/e$ is the required oil pressure in lbs./in.$^2$ for each inch eccentricity (e) of outboard face seal **14**,

$\epsilon$ is the maximum tilt contemplated for face seal **14**,

$\omega$ is the shaft speed in radians per second,

$R_o - R_i$ is the radial span of the juxtaposed faces forming seal **14** in inches,

$\cos\alpha$ is the maximum misalignment contemplated for face seal **14**,

$\mu$ is the viscosity of the water presumed to penetrate the seal interface, in lb-sec./in.$^2$, and

$h$ is the average oil film thickness between faces of the seal in inches. Typically, an oil pressure increase of approximately 4,000 lbs./sq. in. is required to compensate for each inch of shaft eccentricity to assure zero inpumping at the outboard face seal.

Although the seal eccentricity can vary dependent upon such factors as the amount of shaft runout under load and speed, the out-of-roundness of the carbon washer, etc., the total eccentricity generally can be estimated with a high degree of reliability for any given manufacturing procedure. Thus, if manufacturing experience has indicated that an eccentricity of approximately 0.010 inch normally is not exceeded on fabricated face seals, the pressure required for chamber **26** to prevent water inpumping is calculated by multiplying the maximum observed eccentricity by the pressure per inch of face seal eccentricity as calculated by the foregoing Findlay equation, e.g., for an empirically determined maximum eccentricity of approximatley 0.010 inch and a calculated oil pressure of 4,000 psi per inch eccentricity, a total pressure of 40 psi is required in oil chamber **26** to inhibit inpumping.

The outpumping rate at outboard face seal **14** also must be considered to assure that the oil supply within the motor is not exhausted within an unduly short time in an attempt to inhibit water ingress through the face seal. The outpumping rate for the outboard seal therefore is calculated, e.g., from the formula:

$$q = \Delta\, p\, \pi\, R_i h^3 / 6\mu\, \Delta\, R \tag{2}$$

wherein

$q$ is the outpumping rate,

$\Delta p$ is the difference in pressure across face seal **14** in psi,

$h$ is the average film thickness between juxtaposed faces of the seal in inches,

$R_i$ is the radius to the inner edge of the sealing land,

$\mu$ is the viscosity of oil in the seal interface in lb.sec.-/in.$^2$, and

$\Delta R$ is equal to the radial span of the juxtaposed faces forming the seal in inches. The optimum pressure for oil chamber **26** then is chosen as a compromise between the high oil pressure desired to overcome inpumping of water into the motor and the low oil pressure desired to limit the oil outpumping rate at the outboard face seal.

Once the pressure desired for annular oil chamber **26** has been chosen, the geometric configuration of inboard spiral groove face seal **12** required to produce this pressure can be determined in accordance with the teachings of E.A. Muijderman in an article entitled SPIRAL GROOVE BEARINGS published 1966 by Philips Technical Library. One spiral grooved face seal configuration found suitable for a 12 inch submersible motor having a 2 inch rotatable shaft was characterized by **10** equally spaced grooves notched to a depth of 0.0013 inch and extending at a spiral angle of 15° with a groove land to width ratio of 1. The inner and outer diameters of the seal measured 1.87 inches and 2.37 inches, respectively, while the groove inner diameter measured 1.95 inches. With the foregoing seal rotating at a speed of 30 revolutions per second, a maximum pressure of 67.5 psi was observed in annular oil chamber **26**.

FIG. **3** illustrates an improved embodiment of this invention whereby the force of the spiral grooved face seal biasing spring can be reduced without a reduction in the pressure obtainable from the face seal. To achieve this result, a shoulder **72** is notched in carbon runner **28A** at an outboard location relative to O-Ring **46A** thereby permitting pressurized oil within annular oil chamber **26A** to communicate with face **75** and hydraulically drive runner **28A** axially towards mutual contact with stationary member **34A** as the pressure within the oil chamber increases. Although a shoulder **76** is provided in shaft **16A** to seal notched runner **28A** and the position of the back support for spring **60A** has been changed slightly, the self-pressurizing face seal of FIG. **3** otherwise is substantially identical to the face seal illustrated in FIG. **1**. The increased axial force upon runner **28A**, however, resulting from hydraulic pressure on face **75** reduces the gap of the spiral grooved face seal tending to increase the obtainable pressure from the seal. This increased pressure, in turn, results in an increased hydraulic force upon face **75** and the pressure within annular oil chamber **26A** is increased in bootstrap fashion until an equilibrium pressure is reached.

Assuming zero net flow at outboard face seal **14A**, the pressure generated by spiral grooved face seal **12A** (illustrated by pressure diagram P in FIG. **4**) increases approximately linearly from the outer periphery of stationary member **34A** to the inner extent of the grooves in the stationary member, i.e., from $d_3$ to $d_2$ with the pressure along the ungrooved portion of the seal interface, i.e., from $d_2$ to $d_1$, remaining constant at $P_{MAX}$. For simplicity, the average pressure acting over the area between $d_2$ and $d_3$ may be assumed equal to $1/2P_{MAX}$) The maximum pressure at equilibrium therefore can be estimated from the approximate formula:

$$P_{\mathrm{MAX}}=\frac{F_s}{\frac{\pi}{4}\left[\frac{(d_2^2+d_3^2)}{2}-d_4^2\right]} \qquad (3)$$

wherein

$F_s$ is the axial load upon the seal produced by spring **52A** in pounds,

$d_2$ is the internal diameter of the spiral grooved annular portion of the face seal,

$d_3$ is the external diameter of the spiral grooved annular portion of the face seal, and

$d_4$ is the diameter of hydraulic shoulder **72** formed in carbon runner **28A**. One bootstrap seal having a seal inner diameter (i.e., $d_1$) of 1.87 inches, a groove inner diameter (i.e., $d_2$) of 1.95 inches, a seal outside diameter (i.e., $d_3$) of 2.374 inches and a seal balance diameter (i.e., $d_4$) of 2.0 inches produced a hydraulic load of 28.7 lbs. upon the face seal in addition to a bias of 42 lbs. provided by spring **52A** for a total face seal axial load of approximately 70.7 lbs. The spiral grooved runner of the face seal contained 15 equally spaced grooves disposed at a spiral angle of 15° and the runner was rotated at a speed of approximately 30 revolutions per second.

When the required pressure for the intermediate oil chamber is low, e.g., approximately 20 psi, the outboard face seal can be disposed in a back-to-back configuration with the inboard spiral grooved face seal as illustrated in FIG. **5**. The pressure within oil chamber **26B** then applies an axial force upon outboard face seal **14B** tending to separate the confronting faces of the seal requiring a biasing spring **60B** having an axial force sufficient to overcome the hydraulic pressure within chamber **26B** to maintain the desired outboard face seal opening during operation. If a failure of pressure should occur within chamber **26B**, the hydraulic force tending to maintain the outboard seal open would be removed and biasing spring **60B** would tend to close the faces of the outboard seal inhibiting ingress of water into the motor. When the back-to-back seal arrangement is utilized with relatively high seal pressures, e.g., pressures of approximately 60 psi, care must be taken to choose a biasing spring **60B** having sufficient force to inhibit excessive outpumping of oil through the outboard face seal.

A major feature of this invention is the ability to monitor seal operation at an external location by the disposition of a pressure transducer **77** within oil chamber **26B** as illustrated in FIG. **5**. The pressure transducer is connected in series with an alarm **78** and a voltage source, e.g., a transformer **80** having a primary winding **80A** connected across the motor energization leads (not shown), and functions to close the series circuit upon a reduction in pressure within oil chamber **26B** below a predetermined minimum. Alarm **78** then is sounded permitting shutdown and removal of the motor from a submerged location prior to permanent damage of the motor interior by water seepage therein. Should the pressure drop in chamber **26B** be produced by a failure of outboard face seal **14B**, seepage of water through spiral grooved face seal **12B** during shutdown is inhibited by annular land **38** of the face seal. To effectively function as a flow restricter during motor shutdown resulting from failure of outboard seal **14B**, the annular land desirably should have a radial span of at least 0.04 inches.

A self-contained motor protective device is illustrated in FIGS. **6** and **7** wherein a spring loaded pressure relief valve **82** is employed to alter the operation of inboard spiral grooved seal **12C** from a full film to a solid-solid contacting mode in the event of failure of the outboard seal. Relief valve **82** functions to restrict the flow of oil from an annular groove **84** situated at the radially inner terminus of the spiral grooves **22C** to a bypass port **86** during normal operation of the self-pressurizing seal. If the outboard face seal should fail during motor operation reducing the pressure within oil chamber **26C** confined between the face seals, the hydraulic pressure on piston **87** of valve **82** communicated to the valve through axial aperture **88** also drops and the relatively higher pressure of the oil within annular groove **84** overcomes the bias of spring **90** to relieve the pressure at the seal interface through bypass port **86**.

With valve **82** open, the operation of spiral grooved face seal **12C** shifts from a conventional thick film operation, i.e., a film in excess of approximatley 100 microinches typically produced by a conventional groove depth of 1,000 to 1,500 microinches, to a solid-solid contacting mode, i.e., a film width below approximately 50 microinches, substantially limiting the outpumping rate of the spiral grooved face seal. Thus, a portion of the oil pumped by the spiral grooved face seal is valved back to the suction side of the face seal thereby reducing both the maximum pressure generated between faces of the spiral grooved face seal and the quantity of oil pumped into oil chamber **26C**.

It will be appreciated that the spiral groove face seal will tend to close, even without operation of relief valve **82**, upon failure of the outboard seal because of increased maximum pressure at the spiral groove seal interface resulting in a changed oil distribution at the seal interface. If the maximum pressure required by the seal under conditions of leakage exceeds the maximum generating capacity of the seal, the seal will inherently change from a full film mode to a solid-solid contact mode to reduce the outpumping rate. Thus, by careful choice of spiral groove design, e.g., spiral groove width, depth and length, a seal can be fabricated wherein the desired pressure will be produced with outboard seal **14C** functioning properly in a full film mode while a substantially reduced outpumping rate is produced upon failure of the outboard seal.

The previously cited formula (3) for estimating the maximum pressure rise clearly illustrates the effect of the shoulder $d_4$ of FIG. **4** upon the pressure created. Formula (3) assumed an average pressure $P_{MAX}/2$ over that portion of the seal interface where the pressure changes. An alternate, theoretically exact formula for calculating the maximum pressure rise may be obtained by integrating the assumed linear pressure rise over the area between diameters $d_2$ and $d_3$ of FIG. **4**. If the seal diameter $d_4$ is equal to the diameter $d_1$, no shaft shoulder exists and the formula for calculating pressure rise becomes:

$$\frac{P_{MAX}}{F_s \Big/ \frac{\pi}{4} d_3^2} = \frac{3}{1+\frac{d_2}{d_3}\left(1+\frac{d_2}{d_3}\right)-3\left(\frac{d_1}{d_3}\right)^2}$$

wherein

$P_{MAX}$ is the maximum pressure generated by the seal in psi in a full film mode with zero leakage,

$F_s$ is the total force applied to the seal by the biasing spring in pounds,

$d_1$ is the span from the radially inner face of the face seal to the shaft axis,

$d_2$ is the span from the radially inward end of the spiral grooves to the shaft axis, and

$d_3$ is the span from the radially outer periphery of the spiral grooves to the shaft axis,

The ratio of the maximum pressure developed at the pumping seal with leakage at outboard seal **14** interface relative to the maximum pressure capable of being developed by the seal with zero leakage then can be calculated from the formula:

$$\frac{P_{MAX}\text{, with leakage}}{P_{MAX}\text{, no leakage}} = \frac{1+\frac{d_2}{d_3}\left(1+\frac{d_2}{d_3}\right)-3\left(\frac{d_1}{d_3}\right)^2}{1+\frac{d_2}{d_3}\left(1-\frac{d_1}{d_3}\right)-\left(\frac{d_1}{d_3}\right)^2}$$

From this ratio, the maximum pressure capable of being developed by seal **12** with no restriction in leakage at the outboard seal can be calculated to provide an indication of film thickness arising from the pressure increase. When the calculated maximum film pressure under leakage conditions exceeds the maximum generating capability of the seal (as can be calculated from the heretofore cited Muijderman publication), the seal operation changes from a full film mode to a solid-solid contact mode upon failure of the outboard seal.

It should be appreciated that very shallow (e.g., 50 microinches) or very deep (e.g., 20,000 microinches as described in my heretofore cited patent application, Ser. No. 47,824) grooves can be utilized for the inboard face seal to reduce outpumping upon failure of the outboard seal. However, because the pressures produced by these face seals during normal operation is difficult to predict due to variations in fluid viscosity at the seal interface, such seals generally are not recommended for the inboard face seal.

What I claim as new and desire to secure by Letters Patent of the United States of America is:

**1**. A shaft seal for a rotatable machine to inhibit ingress of ambient fluid into said machine, said seal comprising an inboard pumping seal characterized by an annular running member mounted upon said rotatable shaft in juxtaposition with an annular stationary member disposed in a confronting attitude relative to said running member, at least one of said juxtaposed members having viscosity grooves therein extending from a peripheral edge of said member and terminating in a land along a planar face of the member to pump sealing fluid contained within said machine into a relatively confined zone to substantially increase the pressure of said sealing fluid within said zone relative to the sealing fluid pressure within said machine, and face seal means axially mounted upon said shaft at an outboard location relative to said inboard seal to restrict the out flow of sealing fluid from said confined zone, said face seal means including a rotorary member axially mounted upon said rotatable shaft, a stationary member juxtaposed in a co-planar attitude relative to said rotary member, and means including a pressure actuated member for driving said rotary and stationary members toward mutual contact.

**2**. A seal having a rotatable shaft according to claim **1** wherein said inboard pumping seal and said face seal means are disposed in tandem along said shaft, the pressure of said sealing fluid in said confined zone tending to reduce the span between the stationary and rotary members of said face seal means.

**3**. A seal having a rotatable shaft according to claim **1** wherein said inboard pumping seal and said face seal means are disposed in back-to-back configuration along said shaft, the pressure of said sealing fluid in said confined zone tending to increase the span between the stationary and rotary members of said outboard face seal means.

**4**. A seal for a rotatable shaft according to claim **1** wherein one member of said inboard pumping seal is axially slidable along said shaft and further including a shoulder notched within the face of said axially slidable member situated remote from the stationary member forming said seal, said shoulder being in communication with said sealing fluid within said high pressure zone to provide a hydraulic force tending to bias said annular running and annular stationary members into mutual contact.

**5**. A seal for a rotatable shaft according to claim **4** further including an O-Ring seal between said rotary member and said shaft to seal said rotary member upon said shaft.

**6.** A seal having a rotatable shaft according to claim **1** further including means for measuring the pressure within said confined zone and remote signal means responsive to said pressure measuring means for indicating a reduction in the pressure of said confined zone below a predetermined minimum.

**7.** A shaft seal for a rotatable machine to inhibit ingress of ambient fluid into said machine, said seal comprising a spiral grooved face seal disposed at an inboard location along said shaft for pumping sealing fluid from said machine into a substantially confined zone to increase the sealing fluid pressure within said machine, said spiral grooved face seal comprising coaxial rotary and stationary members juxtaposed in a co-planar attitude, at least one of said members being axially slidable along said shaft to vary the span between said members, mechanical means biasing said axially slidable member toward said stationary member, a shoulder notched within said axially slidable member face remote from said stationary member, said shoulder being in communication with the sealing fluid of said confined zone to bias said axially slidable member towards said stationary member of said spiral grooved face seal with increased sealing fluid pressure in said confined zone, and pressure responsive face seal means disposed at an axially outboard location upon said rotatable shaft and operable in response to increased fluid pressure in said confined zone to restrict the flow of sealing fluid from said confined zone.

**8.** A seal for a rotatable shaft according to claim **7** wherein the grooves in said spiral grooved face seal extend from the periphery of said seal to terminate in an annular land.

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