

SUPERINTENDENCIA DE INDUSTRIA Y COMERCIO	
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DEP: 2020 DIRECCION DE NUEVAS CREACIONES	EVE: 378 FASE NACIONAL INCLUIDO CAPITULO I
TRA: 11 PCT-PATENTE DE INVENCION CAPITULOS I Y II	FOLIOS: 6
ACT: 430 REQUERSOLICITA	

DIRECCIÓN DE NUEVAS CREACIONES

2020
Bogotá, D.C.,

Abogado(a)/Señor(a)
SILVANA MARÍA LÓPEZ DÍAZ

Asunto: Radicación: 14-120185- -4
 Trámite: 11
 Evento: 378
 Actuación: 430
 Oficio: 8965
 Folios: 6

Patente de Invención	X			Patente de Modelo de Utilidad	
Vía Nacional					
Vía PCT (Fase Nacional)	X	Capítulo I	X	Capítulo II	
No. Solicitud Internacional	PCT/IB2012/056311				
Fecha de Presentación Internacional	09/11/2012				

Como resultado del examen de fondo realizado, con fundamento en el Artículo 45 de la Decisión 486 de la Comisión de la Comunidad Andina, se comunica:

1. Documentos estudiados

Descripción:	Rad. N° 14-120185-00000-0000	04/junio/2014
Reivindicaciones:	Rad. N° 14-120185-00000-0000	04/junio/2014
Dibujos:	Rad. N° 14-120185-00000-0000	04/junio/2014
Prioridad:	PT 10600	11/noviembre/2011

2. Análisis de la Prioridad

Se acepta la reivindicación de prioridad porque ésta solicitud fue presentada dentro del plazo establecido (Art. 9 D 486) y su contenido técnico corresponde con el de la prioridad reivindicada.

3. Objeto de la invención

De acuerdo con lo establecido en el numeral 1.2.2.2 del Capítulo Primero del Título X de la Circular Única para efectos del examen de patentabilidad, se tendrá en cuenta la tasa pagada por el solicitante al momento de comenzar el examen para las reivindicaciones

adicionales; esto es que cuando la solicitud contenga más de diez (10) reivindicaciones y no se haya pagado la tasa complementaria, sólo se estudiarán las cubiertas por la tasa pagada. Por lo tanto, se estudiarán las reivindicaciones 1 a 11.

Título de la solicitud: “Sistema de enfriamiento interno para sellos mecánicos y uso del mismo” (Fl. 1).

El problema planteado en esta solicitud, consiste en lograr un sistema de enfriamiento de sellos mecánicos, con el que se logre aumentar la eficiencia de los equipos de bombeo en donde están instalados, de tal manera que se eviten tiempos de inactividad, servicios de mantenimiento y los problemas medioambientales originados por las fugas provocadas.

Para resolver este problema técnico, la solicitud en estudio propone un sistema de enfriamiento interno para sellos mecánicos que comprende un circuito interno en el cuerpo del sello mecánico, un punto de entrada del fluido, un punto de salida del fluido y dos anillos raspadores protectores.

El objeto de la presente solicitud se relaciona con la Clasificación Internacional de Patentes (CIP): F 16 J 15/34.

4. Reivindicaciones relativas a un uso o a un segundo uso (Art 14 y 21 D 486)

Las reivindicaciones 10 y 11, al mencionar que el sistema de enfriamiento se aplica a determinados sellos y a bombas centrífugas, hace referencia a la forma de utilizar la invención. De acuerdo al artículo 14, los usos no pueden ser objeto de protección. En consecuencia, las reivindicaciones citadas no comprenden materia patentable y por lo tanto, se sugiere eliminarlas.

5. De la solicitud de patente

Reivindicaciones (Art 30 D 486, Art 22 PCT)

El capítulo reivindicatorio no es claro porque:

- La reivindicación 2 y las expresiones: “...donde circula un fluido de enfriamiento...” reivindicación 1) “...para la zona de sello de anillo estacionario y sello de tapa.” (reivindicación 7), “...donde el fluido de enfriamiento se proyecta por pulverización al área del sello secundario desde el eje de la manga.” (reivindicación 9); describen funciones, efectos, ventajas o resultados a alcanzar, los cuales no son aspectos susceptibles de patente. Para mayor claridad, el producto reclamado se debe definir técnicamente mediante sus características estructurales. Se sugiere al solicitante eliminar la reivindicación y las expresiones citadas.
- La expresión: “al menos” (reivindicación 1), es impreciso, por lo cual no permite distinguir la invención del estado de la técnica. Se sugiere sustituirlo por un término preciso o un rango concreto.

6. Determinación del Estado de la Técnica

El estado de la técnica se determinó con base en la fecha de presentación de la solicitud de prioridad: noviembre 11 de 2011, y se encontraron los siguientes documentos que

anticipan el objeto de esta solicitud:

Ítem	Documento	Título	Fecha de Publicación
D1	DE 20217585	'Sperrdruckdichtung mit Pumfunktion'	06/Mayo/2004
D2	EP 447.898	'Improvements in barrier seal systems'	25/Septiembre/1991
D3	US 4'872.689	'Mechanical seal with heat exchanger'	10/Octubre/1989

El documento D1¹ divulga un sello con acción de bombeo, con el cual se sella un eje rotativo con un alojamiento que comprende unas copas acoplables (15) que tiene un canal de alimentación (32) y un canal de retorno (38) (reivindicación 1)

El documento D2² divulga un sello barrera para sellar un espacio anular entre un alojamiento y un eje rotativo que se extiende por el agujero del alojamiento (resumen).

El documento D2³ divulga sellos que incluyen un par de miembros de sellado dispuestos en proximidad relativa respecto a un eje rotatorio (resumen).

7. Examen de Patentabilidad (Art 14 D486)

Novedad (Art 16 D486)

La reivindicación 1 consiste en: “Un sistema de enfriamiento interno para sellos mecánicos, que comprende: - un circuito interno en el que el cuerpo del sello mecánico que consiste de una conexión...” (Ver página 14 del radicado N° 14-120185-00000-0000).

Comparación de las características técnicas esenciales, con las del estado de la técnica:

Características esenciales	D1	D2	D3
Un sistema de enfriamiento interno para sellos mecánicos que comprende:	Sello de presión con acción de bombeo caracterizado por:	Sello mecánico con intercambiador de calor caracterizado por:	Mejoras en sistemas de sellos caracterizadas por:
Un circuito interno en el cuerpo del sello mecánico que consiste de una conexión efectuada en el diámetro más grande del sello con un diámetro de entre 2.5 y 6.0 mm que se conectara a una cámara previamente implementada dentro del cuerpo, que involucra en su periferia el área donde se aplicara el sello secundario.	Un eje (3) al que lo rodea un alojamiento (1), copas de acoplamiento (15) que tienen un canal de alimentación (32) y un canal de retomo (38) que conectan con los sellos (11) y (23) (reivindicación 1). El conducto de circulación (32), (36), (37) y (38) conduce el fluido para refrigerar o lubricar los sellos (11) y (23) (Transducción de la Descripción, Pág. 16, Párr. 2).	Un intercambiador de calor (12) que hace parte del sello y que consiste en un cuerpo hueco que incluye las paredes (43), (44), (45) y (46). Dicho cuerpo (12) se extiende axialmente hacia posiciones axiales espaciadas en lados opuestos entre los anillos (28) y (32) para que ambos anillos estén expuestos a una zona anular que existe entre sus superficies y la superficie interior del intercambiador de calor	Un sello interno (66) está espaciado axialmente del sello exterior (46). Los sellos interno y externo junto con la camisa (36) y el plato (18) forman una cámara (68). La carama esta en comunicación con el conducto (28) en el plato por donde se suministra el fluido a la cámara desde el un reservorio (Col. 4, líneas 11 a 17). El anillo primario tiene una pluralidad de ranuras (82) formadas en el diámetro interior, las cuales facilitan el

1 http://worldwide.espacenet.com/publicationDetails/biblio?CC=DE&NR=20217585U&KC=&locale=en_EP&FT=E

2 http://worldwide.espacenet.com/publicationDetails/description?CC=EP&NR=0447898A2&KC=A2&FT=D&ND=&date=19910925&DB=&locale=en_EP

3 <https://docs.google.com/viewer?url=patentimages.storage.googleapis.com/pdfs/US4872689.pdf>

		(12) (Col. 4, líneas 35 a 55; Col. 5, líneas 1 a 4).	movimiento del fluido a través de la cámara (68) a las caras de los sellos (Fig. 2; Col. 4, líneas 31 a 35).
Un punto de entrada del fluido de enfriamiento.		Una entrada (52) para el medio de intercambio de calor (Col. 5, líneas 15 a 18).	Una línea de suministro de fluido desde un reservorio conecta con alguna de las entradas (24), (26), (19) o (31) (Col. 2, líneas 26 a 42).
Un punto de salida del fluido de enfriamiento.		Una salida (54) para el medio de intercambio de calor (Col. 5, líneas 15 a 18).	Una salida (128) de vuelta al reservorio (Col. 5, líneas 49 a 51).
Dos anillos raspadores protectores.	Anillos (30), (18) y (16) y los sellos dinámicos (13 y 25) (Fig. 1).	Un par de sellos anulares (Col. 3, líneas 58 a 60). O-ring (24) (Col. 3, líneas 66 y 67)	O-ring (40), anillo de retención (42) (Col. 3, líneas 43 a 52).

De acuerdo con la tabla anterior, las características esenciales del sistema de la reivindicación 1 habían sido divulgadas previamente en los documentos D1, D2 y D3, por lo tanto, la materia reivindicada no es nueva.

Asimismo, los documentos antecedentes divulgan las características de las reivindicaciones dependientes así:

- Reivindicación 3: Una salida (128) de vuelta al reservorio. Una línea de suministro de fluido desde un reservorio conecta con alguna de las entradas (24), (26), (19) o (31) (D3: Figs. 4 y 5; Col. 2, líneas 26 a 42; Col. 5, líneas 49 a 51).
- Reivindicación 4: Los puntos de entrada y salida del fluido de enfriamiento tienen el mismo diámetro (D3: Figs. 4 y 5)
- Reivindicación 6: Se aplica un separador entre los puntos de entrada y salida del fluido de enfriamiento (D3: Fig. 5).
- Reivindicación 7: El conducto de circulación (32), (36), (37) y (38) conduce el fluido para refrigerar o lubricar los sellos (11) y (23) (D1: Transducción de la descripción, Pág. 16, Párr. 2).
- Reivindicación 8: Anillos (30), (18) y (16) y los sellos dinámicos (13 y 25) (D1: Fig. 1). Un par de sellos anulares (D2: Col. 3, líneas 58 a 60). O-ring (24) (D2: Col. 3, líneas 66 y 67). O-ring (40), anillo de retención (42) (D3: Col. 3, líneas 43 a 52).
- Reivindicación 9: Los sistemas tipo aspersor para lubricación y refrigeración no pueden dar novedad a la invención, por cuanto son ampliamente conocidos y utilizados en sellos instalados en sistemas de bombeo⁴.

Las reivindicaciones 2, 10 y 11 fueron afectadas en el numeral 4 de la presente comunicación.

4 BLOCH, Heinz y BUDRIS, Allan. Pump User's Handbook: Life Extension. Cuarta Edicion. Figura 9- 11. Disponible en: https://books.google.com.co/books?id=6TZSCAAQBAJ&pg=PT1159&dq=spray+for+lubrication++for+seals+of+pumps&hl=es&sa=X&ei=YiydVeGCilMvNo_yiOgJ&ved=0CCQQ6AEwAA#v=onepage&q=spray%20for%20lubrication%20%20for%20seals%20of%20pumps&f=false

8. Conclusión

Los requisitos de Patentabilidad de la presente solicitud están afectados así:

Ítem	Documento	Reivindicaciones afectadas	Requisito que afecta
D1	DE 20217585	1, 7 y 8	Novedad/Nivel Inventivo
D2	EP 447.898	1, 6 y 8	Novedad/Nivel Inventivo
D3	US 4'872.689	1, 3, 4 y 8	Novedad/Nivel Inventivo

En cumplimiento del artículo 45 de la Decisión 486 de la Comisión de la Comunidad Andina, se le indica que debe dar respuesta dentro del término de sesenta (60) días contados a partir del día siguiente de la fecha de notificación de la presente comunicación. En caso de no dar respuesta al presente requerimiento, o si a pesar de la respuesta subsistieran los impedimentos para la concesión, se denegará la patente.

Se invita al solicitante se sirva indicar, si así lo considera, su renuncia al término faltante de sesenta (60) días para dar respuesta a este requerimiento y presentarla por escrito.

Finalmente, se le recuerda al solicitante que si como respuesta a este requerimiento llegara a modificar el capítulo reivindicatorio y éste contenga más de diez (10) reivindicaciones, no pagará la tasa de modificación voluntaria pero sí la correspondiente al pago por reivindicación adicional.

Notifíquese el presente oficio conforme a lo dispuesto en el numeral 5.2, literal e), del capítulo quinto del título primero de la Circular Única No. 10 de 2001.



JOSÉ LUIS SALAZAR LÓPEZ
Director de Nuevas Creaciones (c)

El anterior requerimiento fue notificado por fijación en lista, de fecha 2015-08-04

Elaboró: Monica Andrea Gutierrez Almonacid
Revisó: Sonia Andrea Diaz Pirazan



(19)
Bundesrepublik Deutschland
Deutsches Patent- und Markenamt

(10) **DE 202 17 585 U1** 2004.05.06

(12)

Gebrauchsmusterschrift

(22) Anmeldetag: 14.11.2002

(47) Eintragungstag: **01.04.2004**

(43) Bekanntmachung im Patentblatt: **06.05.2004**

(51) Int Cl.⁷: **F16J 15/34**

(71) Name und Wohnsitz des Inhabers:

M.E.F. Financing Establishment, Ruggell, LI

(74) Name und Wohnsitz des Vertreters:

**Patentanwälte von Kreisler, Selting, Werner et col.,
50667 Köln**

(56) Recherchenergebnisse nach § 7 Abs. 2 GbrMG:

DE 33 20 095 A1

DE 298 00 616 U1

DE 297 13 603 U1

DE 66 08 692 U

DD 95 726

US 64 28 011 B1

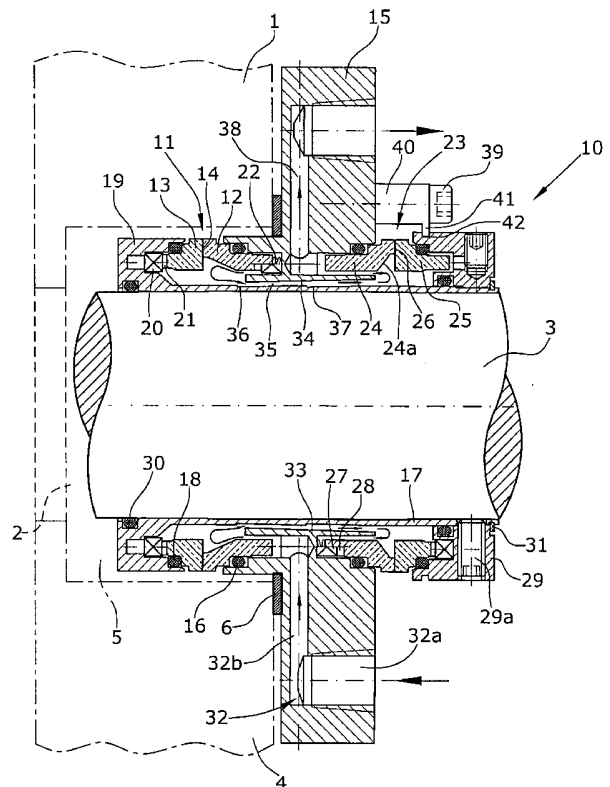
US 30 88 744

EP 12 88 542 A2

Die folgenden Angaben sind den vom Anmelder eingereichten Unterlagen entnommen

(54) Bezeichnung: **Sperrdruckdichtung mit Pumpfunktion**

(57) Hauptanspruch: Sperrdruckdichtung mit Pumpwirkung, die zur Abdichtung einer rotierenden Welle (3) gegen ein Gehäuse (1) an diesem befestigbar ist, mit einer die Welle (3) umgebenden, an dem Gehäuse (1) anbringbaren Brille (15), die einen Zulaufkanal (32) und einen Rücklaufkanal (38) für ein Sperrmedium aufweist, einer die Welle (3) gegen den Gehäuseinnenraum abdichtenden inneren Gleitringdichtung (11) und einer die Welle (3) gegen die Umgebung abdichtenden äußeren Gleitringdichtung (23), wobei jede Gleitringdichtung (11, 23) zwei gegeneinander vorgespannte Gleitringe (12, 13; 24, 25) aufweist und wobei ein Zirkulationsweg (32, 36, 35, 37, 38) für das Sperrmedium zwischen den Kanälen (32, 38) und entlang der Gleitringdichtungen (11, 23) verläuft, dadurch gekennzeichnet, daß die Brille (15) im Zentralbereich (33) zwischen den beiden Gleitringdichtungen (11, 23) eine sich in Strömungsrichtung erweiternde glatte konische Pumpkanalwand (34) zur Förderung des Sperrmediums aufweist.



Beschreibung

[0001] Eine Gleitringdichtung weist im allgemeinen einen stationären Gleitring auf, der z.B. mit einer flanschartigen Brille drehfest an dem Gehäuse befestigt wird, und einen dynamischen Gleitring, der drehfest mit der Welle befestigt wird und mit dieser rotiert. An einer Gleitfläche liegen die beiden Gleitringe aneinander an und bewirken dort, die Abdichtung zwischen den stationären und den dynamischen Teilen der Gleitringdichtung. Die Gleitringe sind beispielsweise mit Federn gegeneinander vorgespannt, um einen kräftigen Andruck und damit eine gute Dichtwirkung zu erzielen.

[0002] Bei Sperrdruckdichtungen wird innerhalb der Gleitringdichtung ein Druck mit Hilfe eines Sperrmediums, wie beispielsweise Öl, aufgebaut, der einem auf der Gleitringdichtung lastenden Druck entgegenwirkt. Gleichzeitig dient das Sperrmedium zur Wärmeabfuhr und Schmierung der Gleitringdichtung. Üblicherweise sind Sperrdruckdichtungen als sogenannte Doppel-Gleitringdichtungen mit einer inneren und einer äußeren Gleitringdichtung ausgeführt.

[0003] Eine derartige Doppel-Gleitringdichtung ist in DE 29800616 offenbart. Sie weist zwei Gleitringdichtungen auf, wobei eine innere Gleitringdichtung den Gehäuseinnenraum gegenüber der Welle abdichtet und eine äußere Gleitringdichtung die Umgebung gegen die Welle abdichtet. Der Zulauf und Rücklauf des Sperrmediums findet in der Brille zwischen den beiden Gleitringdichtungen statt. Das Sperrmedium wird im Bereich der äußeren Gleitringdichtung mit Hilfe einer an einer Hülse ausgebildeten schraubenförmigen Pumpnut und einer die Pumpnut umgebenden Pumpkanalwand gefördert.

[0004] Der neuen Erfindung liegt die Aufgabe zugrunde, eine für schnell-laufende Wellen geeignete Doppelgleitringdichtung mit einer verbesserten Förderungsmöglichkeit des Sperrdruckmediums bei gleichzeitiger vereinfachter Konstruktion zu schaffen.

[0005] Diese Aufgabe wird erfindungsgemäß mit den Merkmalen des Anspruchs 1 gelöst.

[0006] Die Erfindung schlägt eine Doppel-Gleitringdichtung vor, deren Brille im Bereich zwischen den beiden Gleitringdichtungen eine konische Pumpkanalwand zur Förderung des Sperrmediums aufweist. Mit dieser konischen Fläche wird innerhalb der Dichtung eine Pumpwirkung erzielt, die sich die Zentrifugalkraft zu Nutze macht. Durch die Zentrifugalkraft der umlaufenden Welle wird das Sperrmedium nach außen abgeschleudert, wo es auf die Pumpkanalwand auftrifft. Dort wird das Sperrmedium aufgrund der konischen Form der Pumpkanalwand weiterbefördert. Somit wird auf einfache Art und Weise eine Pumpwirkung vom Zulauf zum Ablass der Gleitringdichtung erzielt. Die erfindungsgemäße Dichtung eignet sich für schnell-laufende Pumpen. Trotz einer sehr kleinen Bauweise, welche es der Dichtung ermöglicht, auch in kleinen Räumen mit kleinen Durchmessern eingesetzt werden zu können, benötigt die-

se Dichtung keine Schneckenförderung oder separate mechanische Förderung im Zirkulationsweg. Sie eignet sich auch für sogenannte Thermosyphonsysteme, die für Sperrmedien häufig eingesetzt werden, da die gewissermaßen natürliche Zirkulation des Sperrmediums von der Dichtung unterstützt wird.

[0007] Vorzugsweise ist der Durchmesser der Pumpkanalwand im Bereich der äußeren Gleitringdichtung größer als im Bereich der inneren Gleitringdichtung, so daß das Sperrmedium von innen nach außen gefördert wird. Dies hat den Vorteil, daß zunächst die kühlere innere Gleitringdichtung von dem Sperrmedium gekühlt wird und erst dann die äußere, die meistens heißer wird, da sie nicht von außen gekühlt werden kann. So kann das auf der Innenseite der Doppelgleitringdichtung geringfügig erwärmte Sperrmedium auch noch die äußere Gleitringdichtung effizient kühlen.

[0008] Gemäß einer bevorzugten Ausführungsform der Erfindung beginnt die Pumpkanalwand in Höhe des statischen Gleitrings der inneren Gleitringdichtung, wodurch das Sperrmedium unbehindert an der innen liegenden Seite der inneren Gleitringdichtung zirkulieren kann. Das andere Ende der Pumpkanalwand erstreckt sich hingegen bis etwa zur Gleitfläche der äußeren Gleitringdichtung, so daß das Sperrmedium direkt zu der Gleitfläche gefördert wird. Oftmals weist der stationäre Gleitring einen Hinterschnitt auf, dann kann sich die Pumpkanalwand auch nur bis zum inneren, das heißt der Brille zugewandten, Anfang des Hinterschnitts erstrecken, was für eine Zirkulation des Sperrmedium im Bereich der äußeren Gleitringdichtung ausreichend ist.

[0009] Die dynamischen Gleitringe können an einer auf die Welle aufschiebbarer Hülse befestigt sein, was die Montage erheblich vereinfacht. Des weiteren kann die Hülse im Bereich der inneren Gleitringdichtung eine Einlaßausnehmung aufweisen, die außerhalb, das heißt vor dem inneren Ende der Pumpkanalwand beginnt und sich bis unter die Pumpkanalwand fortsetzen kann. Diese Einlaßausnehmung verbessert das Einströmen des Sperrmedium in einen zwischen Pumpkanalwand und Hülse gebildeten Pumpkanal. Es kann auch noch eine Auslaßausnehmung in der Hülse vorgesehen sein, die innerhalb der Pumpkanalwand beginnt und sich auch bis aus der Pumpkanalwand heraus fortsetzen kann. Die Auslaßausnehmung erleichtert das Ausströmen des Sperrmediums aus dem Pumpkanal. Die beiden Ausnehmungen erleichtern zusätzlich die Montage der Hülse innerhalb der Brille.

[0010] Zur Verbesserung des Förderverhaltens können die Ausnehmungen konisch ausgebildet sein, vorzugsweise parallel zu der Pumpkanalwand. Dadurch kann das Abschleudern des Sperrmediums verbessert werden. Mittels der Auslaßausnehmung oder gegebenenfalls noch weiterer zwischen der Einlaßausnehmung und der Auslaßausnehmung angeordneten Ausnehmungen kann die Breite des Pumpkanals abschnittsweise angepaßt werden, um die

Pumpwirkung zu beeinflussen.

[0011] Im folgenden wird ein Ausführungsbeispiel der Erfindung anhand der Zeichnung näher erläutert, wobei **Fig. 1** einen Längsschnitt durch die Sperrdruckdichtung zeigt.

[0012] Gemäß **Fig. 1** weist ein druckdichtes Gehäuse **1** einen Wellendurchgang **2** auf, durch das eine rotierende Welle **3** verläuft. Ausgehend von der Außenfläche **4** des Gehäuses **1** verläuft ein Aufnahmeraum **5**, der die Welle **3** ringförmig umgibt. Eine Sperrdruckdichtung **10** dichtet den Gehäuseinnenraum und die Umgebung gegen die Welle **3** ab. Die Gehäuseaußenwand **4** ist mit einem Dichtring **6** gegenüber der Sperrdruckdichtung **10** abgedichtet.

[0013] Die Sperrdruckdichtung **10** ist hier als Doppel-Gleitringdichtung ausgeführt; es können auch andere Dichtungstypen verwendet werden, solange diese als Sperrdruckdichtungen betrieben werden können.

[0014] Eine die Welle **3** umgebende innere Gleitringdichtung **11** der Sperrdruckdichtung **10** befindet sich in dem Aufnahmeraum **5**, um den Gehäuseinnenraum gegen die Welle **3** abzudichten. Die innere Gleitringdichtung **11** besteht aus einem stationären Gleitring **12** und einem dynamischen Gleitring **13**; die beiden Gleitringe **12**, **13** treffen in einer Gleitfläche **14** aufeinander, die die Dichtfläche zwischen ortsfesten und rotierenden Teilen bildet.

[0015] Der stationäre Gleitring **12** greift in eine die Welle **3** umgebende Brille **15** ein, die mittels mehrerer nicht gezeigter Schrauben mit der Außenfläche **4** des Gehäuses **1** verbunden ist. Der stationäre Gleitring **12** ist axial beweglich in der Brille **15** gelagert und mit einem Dichtring **16** gegen die Brille **15** abgedichtet. Zur Drehsicherung weist der stationäre Gleitring **12** einen Schlitz auf, in den ein Vorsprung der Brille **15** eingreift.

[0016] Der dynamische Gleitring **13** ist an einer auf der Welle **3** sitzenden Hülse **17** mit geringem radialen Spiel befestigt und mit einem Dichtring **18** gegen die Hülse **17** abgedichtet. Zur Drehsicherung des dynamischen Gleitrings **13** gegenüber der Welle **3** greift ein in einem radial abstehenden Bereich **19** der Hülse **17** eingesetzter Bolzen **20** in einen Schlitz **21** des dynamischen Gleitrings **13** ein. Mehrere über den Umfang verteilte Federn **22** stützen sich an der Brille **15** ab und pressen den stationären Gleitring **12** an der Gleitfläche **14** mit dem dynamischen Gleitring **13** zusammen.

[0017] An der Außenseite der Brille **15** ist eine äußere Gleitringdichtung **23** angeordnet, die im Aufbau im wesentlichen symmetrisch zu der inneren Gleitringdichtung **11** ist. So weist auch die äußere Gleitringdichtung **23** einen stationären Gleitring **24** und einen dynamischen Gleitring **25** auf, die zusammen eine Gleitfläche **26** bilden.

[0018] Der stationäre Gleitring **24** der äußeren Gleitringdichtung **23** ist ebenso wie der stationäre Gleitring **12** der inneren Gleitringdichtung **11** an der Brille **15** befestigt. Zur Verdrehsicherung greift ein an der

Brille **15** befestigter Bolzen **27** in einen Schlitz **28** des stationären Gleitrings **24** ein. Die beiden Gleitringe **24**, **25** der äußeren Gleitringdichtung **23** sind ebenfalls mit Federn gegeneinander vorgespannt.

[0019] Der dynamische Gleitring **25** der äußeren Gleitringdichtung **23** ist an einem ringförmigen Aufsatz **29** befestigt, der mit der Hülse **17** verbunden ist. Die Hülse **17** erstreckt sich durchgängig über die gesamte Länge der Gleitringdichtung **10**. An der Innenseite ist die Hülse **17** mit einem Dichtring **30** gegen den Gehäuseinnenraum abgedichtet. An der Außenseite weist die Hülse **17** mehrere über den Umfang verteilte Löcher auf, durch die jeweils eine Setzschraube **29a** des ringförmigen Aufsatzes **29** gegen die Welle verschraubt wird, so daß der ringförmige Aufsatz **29** auf der Welle **3** fixiert ist. Mit einem Sprengling **31**, der den ringförmigen Aufsatz **29** mit der Hülse **17** verbindet, wird die Hülse **17** axial an der Welle **3** gesichert.

[0020] Die Brille **15** weist einen Zulaufkanal **32** für das Sperrmedium auf. Der Zulaufkanal **32** hat einen mit einem Gewinde versehenen Bereich **32a**, der sich von der dem Gehäuse **1** abgewandten Stirnseite der Brille **15** ausgehend, in bezüglich der Welle **3** axialer Richtung in die Brille **15** erstreckt. Dieser Bereich geht in einen radial verlaufenden Bereich **32b** des Zulaufkanals **32** über, dessen anderes Ende sich im Zentralbereich **33** der Brille befindet, wobei der Zentralbereich **33** sich axial in der Mitte zwischen den beiden stationären Gleitrings **12**, **24** befindet. Von dort gelangt das Sperrmedium in einen Spalt, der von der inneren Gleitringdichtung **11** und der Hülse **17** begrenzt wird. Weiterhin bewegt sich das Sperrmedium zwischen dem zentralen Bereich **33** der Brille und der Hülse **17** hindurch.

[0021] In dem zentralen Bereich **33** ist in der Brille **15** eine Pumpkanalwand **34** ausgebildet, die sich in axialer Richtung etwa von der Mitte des stationären Gleitrings **12** der inneren Gleitringdichtung **11** bis zu einem Hinterschnitt **24a** des stationären Gleitrings **24** der äußeren Gleitringdichtung **23** erstreckt. Die Pumpkanalwand **34** ist konisch ausgebildet, wobei sich der Durchmesser in Richtung der äußeren Gleitringdichtung **23** vergrößert.

[0022] Zwischen der Pumpkanalwand **34** und der Hülse **17** wird ein Pumpkanal **35** gebildet, durch den das Sperrmedium von der inneren Gleitringdichtung **11** zu der äußeren Gleitringdichtung **23** in Pfeilrichtung gefördert wird.

[0023] In der Hülse **17** ist vor dem inneren Ende der Pumpkanalwand **34** eine Einlaßausnehmung **36** ausgebildet. Diese Einlaßausnehmung **36** beginnt etwa in der Mitte des stationären Gleitrings **12** der inneren Gleitringdichtung **11** und setzt sich bis etwa zur Mitte der Pumpkanalwand **34** fort. Am Ende der Einlaßausnehmung **36** erreicht die Hülse **17** wieder ihren ursprünglichen Durchmesser. Die Einlaßausnehmung **36** ist wie die Pumpkanalwand **34** konisch ausgebildet und zwar mit dem gleichen Winkel, so daß die Flächen der Pumpkanalwand **34** und der Einlaßaus-

nehmung **36** parallel verlaufen.

[0024] Am Ende der Einlaßausnehmung **36** beginnt eine Auslaßausnehmung **37**, die fast bis zum äußeren Ende der Pumpkanalwand **34** verläuft. Die Fläche der Auslaßausnehmung **37** ist ebenfalls parallel zu der Fläche der Pumpkanalwand **34**.

[0025] Nachdem das Sperrmedium aus dem Zulaufkanal **32** ausgetreten ist, zirkuliert es zunächst in Pfeilrichtung an der inneren Gleitringdichtung **11** entlang. Durch die umlaufende Welle **3** wird das Sperrmedium mitgenommen, so daß sich eine schraubenförmige Bewegung des Sperrmediums ergibt. Nun gelangt das Sperrmedium über die Einlaßausnehmung **36** in den Pumpkanal **35**. Dort wird es aufgrund der Zentrifugalkraft der Welle **3** von dieser abgeschleudert und trifft auf die konische Pumpkanalwand **34** auf. Hier wird das Sperrmedium nach außen, also in Richtung der äußeren Gleitringdichtung **23** gedrückt. Im Bereich der Auslaßausnehmung **37** ist der Querschnitt des Pumpkanals **35** erweitert, so daß das Sperrmedium ungehindert aus dem Pumpkanal **35** unter die äußere Gleitringdichtung **23** austreten kann, wo das Sperrmedium zur Kühlung zirkuliert, bevor es über einen Rücklaufkanal **38**, der wie der Zulaufkanal **32** ausgebildet ist, die Sperrdruckdichtung **10** verläßt. In dem beschriebenen Zirkulationsweg **32, 36, 35, 37** und **38** strömt das Sperrmedium zur Abführung von Wärme und zur Schmierung der Gleitringdichtungen **11** und **23**.

[0026] Durch die Pumpkanalwand **34** und die Einlaß- und Auslaßausnehmungen **36, 37** wird eine integrierte Pumpwirkung erzielt, die das Sperrmedium durch die Sperrdruckdichtung zirkulieren läßt; so kann auf externe Pumpmittel zumindest weitgehend verzichtet werden. Die erzielte Pumpwirkung hängt neben der Drehzahl der Welle **3** und der Größe der Dichtung auch von der Größe des Pumpkanals **35** ab. Die Größe kann durch den Abstand der Pumpkanalwand **34** zu der Hülse **17** und durch die Ausgestaltung der Ausnehmungen **36, 37** variiert werden.

[0027] Bei der Montage der Sperrringdichtung **10** wird diese einfach auf die Welle **3** aufgeschoben und die stationären Teile der Sperrdruckdichtung **10** mit der Brille **15** werden an dem Gehäuse **1** befestigt. Zur Befestigung der dynamischen Teile der Sperrdruckdichtung **10** werden die Setzschrauben **30** angezogen. Um während des Transports und während der Montage eine Dejustierung der Sperrdruckdichtung **10** zu verhindern, sind an der Außenseite der Brille **15** Justierschrauben **39** angebracht. Jede Justierschraube **39** sichert eine Justierhülse **40**, die einen radial abstehenden Vorsprung **41** aufweist. Ist die Justierschraube **39** gelöst, kann die Justierhülse **40** gedreht werden, so daß der Vorsprung **41** in eine ringförmige Nut **42** des ringförmigen Aufsatzes **29** eingreift und somit die innere Gleitringdichtung **11** und die äußere Gleitringdichtung **23** unter der gewünschten Vorspannung sichert. Nachdem die Sperrdruckdichtung **10** sowohl an dem Gehäuse **1** als auch an der Welle **3** befestigt ist, können die Jus-

tierschrauben **39** gelöst und die Vorsprünge **41** aus dem Eingriff herausgedreht werden. Die Justierhülsen **40** werden zur späteren Wiederverwendung mit dem Vorsprung **41** von der Nut **42** abgewandt wieder an der Brille **15** festgeschraubt.

Schutzansprüche

1. Sperrdruckdichtung mit Pumpwirkung, die zur Abdichtung einer rotierenden Welle (**3**) gegen ein Gehäuse (**1**) an diesem befestigbar ist, mit einer die Welle (**3**) umgebenden, an dem Gehäuse (**1**) anbringbaren Brille (**15**), die einen Zulaufkanal (**32**) und einen Rücklaufkanal (**38**) für ein Sperrmedium aufweist, einer die Welle (**3**) gegen den Gehäuseinnenraum abdichtenden inneren Gleitringdichtung (**11**) und einer die Welle (**3**) gegen die Umgebung abdichtenden äußeren Gleitringdichtung (**23**), wobei jede Gleitringdichtung (**11, 23**) zwei gegeneinander vorgespannte Gleitringe (**12, 13; 24, 25**) aufweist und wobei ein Zirkulationsweg (**32, 36, 35, 37, 38**) für das Sperrmedium zwischen den Kanälen (**32, 38**) und entlang der Gleitringdichtungen (**11, 23**) verläuft, **dadurch gekennzeichnet**, daß die Brille (**15**) im Zentralbereich (**33**) zwischen den beiden Gleitringdichtungen (**11, 23**) eine sich in Strömungsrichtung erweiternde glatte konische Pumpkanalwand (**34**) zur Förderung des Sperrmediums aufweist.

2. Sperrdruckdichtung nach Anspruch 1, dadurch gekennzeichnet, daß der Durchmesser der Pumpkanalwand (**34**) in Richtung der äußeren Gleitringdichtung (**23**) zunimmt.

3. Sperrdruckdichtung nach Anspruch 1 oder 2, dadurch gekennzeichnet, daß die Pumpkanalwand (**34**) sich von dem statischen Gleitring (**12**) der inneren Gleitringdichtung (**11**) bis zu einer zwischen dem statischen Gleitring (**24**) und dem dynamischen Gleitring (**25**) gebildeten Gleitfläche (**26**) der äußeren Gleitringdichtung (**23**) erstreckt.

4. Sperrdruckdichtung nach einem der Ansprüche 1 bis 3, dadurch gekennzeichnet, daß die Pumpkanalwand (**34**) sich bis zu einem Hinterschnitt (**24a**) des stationären Gleitrings (**24**) der äußeren Gleitringdichtung (**23**) erstreckt.

5. Sperrdruckdichtung nach einem der Ansprüche 1–4, dadurch gekennzeichnet, daß die an der Welle (**3**) anbringbaren dynamischen Gleitringe (**13, 25**) an einer auf die Welle (**3**) aufschiebbarer Hülse (**17**) angebracht sind, die durch die Brille (**15**) hindurchragt und eine der Pumpkanalwand (**34**) gegenüberliegende generell zylindrische Außenfläche hat.

6. Sperrdruckdichtung nach Anspruch 5, dadurch gekennzeichnet, daß die Hülse (**17**) im Bereich der inneren Gleitringdichtung (**11**) eine Einlaßausnehmung (**36**) aufweist, die außerhalb der Pumpkanal-

wand (34) beginnt.

7. Sperrdruckdichtung nach Anspruch 5 oder 6, dadurch gekennzeichnet, daß die Hülse (17) eine Auslaßausnehmung (37) aufweist, die innerhalb der Pumpkanalwand (34) beginnt.

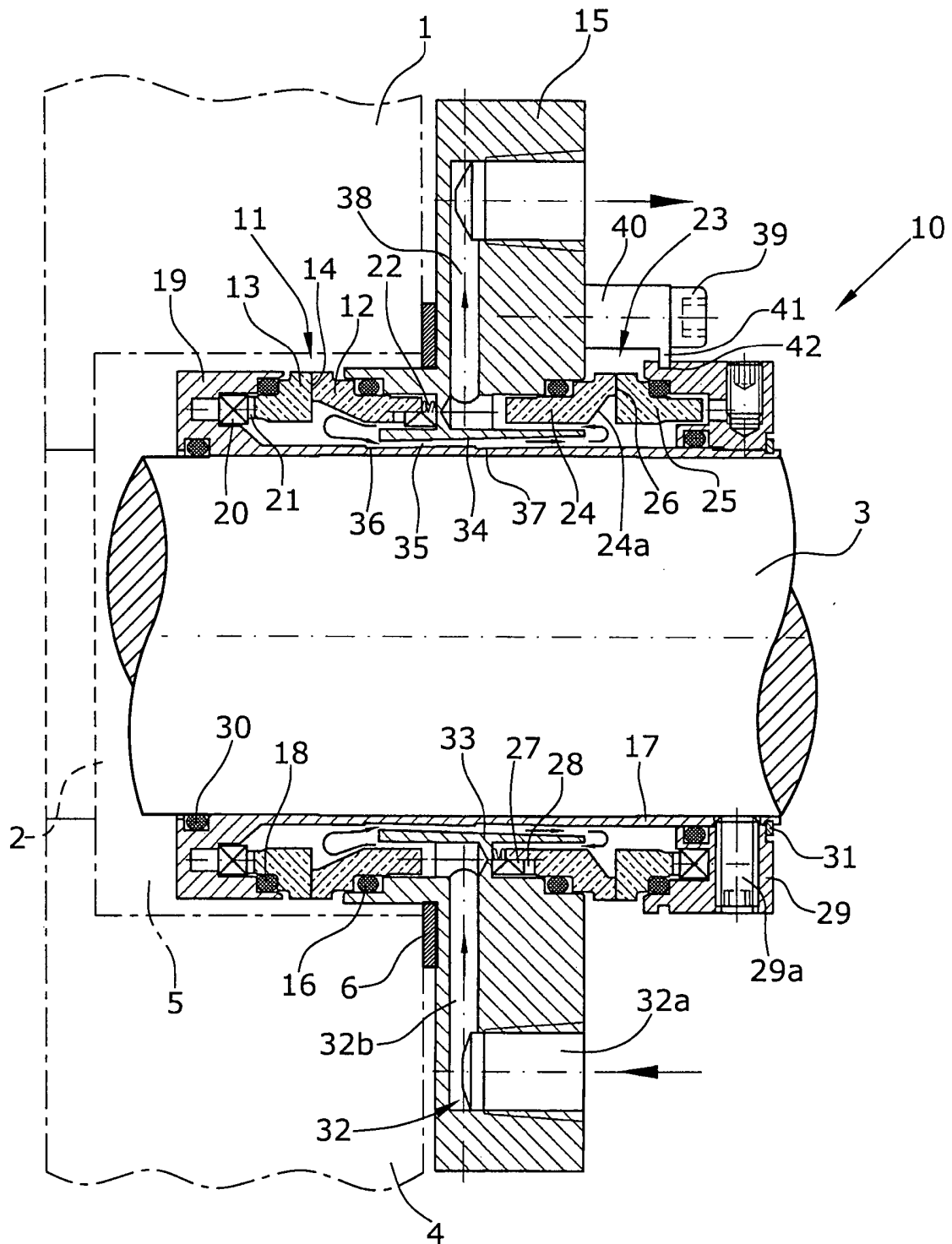
8. Sperrdruckdichtung nach Anspruch 6 oder 7, dadurch gekennzeichnet, daß die Einlaßausnehmung (36) und/oder die Auslaßausnehmung (37) konisch ausgebildet sind.

9. Sperrdruckdichtung nach Anspruch 8, dadurch gekennzeichnet, daß die Flächen der Pumpkanalwand (34) und der Einlaßausnehmung (36) und/oder der Auslaßausnehmung (37) parallel verlaufen.

10. Sperrdruckdichtung nach einem der Ansprüche 1–9, dadurch gekennzeichnet, dass die Pumpkanalwand (34) einen Durchmesser hat, der kleiner ist als die Innendurchmesser der Gleitringe.

Es folgt ein Blatt Zeichnungen

Anhängende Zeichnungen



Notice

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DESCRIPTION DE20217585

A mechanical seal has a stationary Gleitringauf, the z generally.

Eine Gleitringdichtung weist im allgemeinen einen stationären Gleitringauf, der z.

B. with a flange-glasses-rotatably attached to the housing is, and a dynamic seal ring, the wird und rotatably attached to the shaft for rotation therewith.

B. mit einer flanschartigen Brille drehfest an dem Gehäuse befestigt wird, und einen dynamischen Gleitring, der drehfest mit der Welle befestigt wird und mit dieser rotiert.

Lie on a sliding surface aneinanderan the two sliding rings and cause there, the seal between the stationary and the dynamic parts of the mechanical seal.

An einer Gleitfläche liegen die beiden Gleitringe aneinanderan und bewirken dort, die Abdichtung zwischen den stationären und den dynamischen Teilen der Gleitringdichtung.

The sliding rings are for example biased together with springs in order to achieve a strong contact pressure and thus a good sealing effect.

Die Gleitringe sind beispielsweise mit Federn gegeneinandervorgespannt, um einen kräftigen Andruck und damit eine gute Dichtwirkung zu erzielen.

When barrier pressure seals of the mechanical seal einemauf burdensome pressure is within the mechanical seal impression by means of a sealing medium, such as oil, up, counteracts.

Bei Sperrdruckdichtungen wird innerhalb der Gleitringdichtung ein Druck mit Hilfe eines Sperrmediums, wie beispielsweise Öl, aufgebaut, der einemauf der Gleitringdichtung lastenden Druck entgegenwirkt.

At the same time serves dasSperrmedium for heat dissipation and lubrication of the mechanical seal.

Gleichzeitig dient dasSperrmedium zur Wärmeabfuhr und Schmierung der Gleitringdichtung.

Common enough, are blocking pressure seals as so-called double mechanical seals run with a innerenund an external mechanical seal.

Üblicherweise sind Sperrdruckdichtungen als sogenannte Doppel-Gleitringdichtungen mit einer innerenund einer äußeren Gleitringdichtung ausgeführt.

Such a double mechanical seal is disclosed in.

Eine derartige Doppel-Gleitringdichtung ist in offenbart.

It has two seals, an inner mechanical seal the housing interior relative to the shaft seals and an outer mechanical seal seals the environment against the shaft.

Sie weist zwei Gleitringdichtungen auf, wobei eine innere Gleitringdichtung den Gehäuseinnenraum gegenüber der Welle abdichtet und eine äußere Gleitringdichtung die Umgebung gegen die Welle abdichtet.

The Zulauf und return of the barrier medium is in the gap between the two Gleitringdichtungen statt.

Der Zulauf und Rücklauf des Sperrmediums findet in der Brille zwischen den beiden

Gleitringdichtungenstatt.

The locking medium is conveyed in the field of external mechanical seal with the aid of a sleeve
eineran trained helical pump groove and the pump groove-surrounding pump duct wall.

Das Sperrmedium wird im Bereich der äußeren Gleitringdichtung mit Hilfe eineran einer Hülse
ausgebildeten schraubenförmigen Pumpnut und einer die Pumpnut umgebendenPumpkanalwand
gefördert.

The new invention has for its object to provide a suitable for fast-running double mechanical
waves with an improved locking Förderungsmöglichkeitdes print medium while simplified
construction.

Der neuen Erfindung liegt die Aufgabe zugrunde, eine für schnell-laufendeWellen geeignete
Doppelgleitringdichtung mit einer verbesserten Förderungsmöglichkeitdes Sperrdruckmediums bei
gleichzeitiger vereinfachter Konstruktion zu schaffen.

This object is achieved by the features of Anspruchs1.

Diese Aufgabe wird erfindungsgemäß mit den Merkmalen des Anspruchs1 gelöst.

The invention proposes a double mechanical seal, whose Brilleim area between the two seals has
a conical pump channel wall zurFörderung the barrier medium.

Die Erfindung schlägt eine Doppel-Gleitringdichtung vor, deren Brilleim Bereich zwischen den
beiden Gleitringdichtungen eine konische Pumpkanalwand zurFörderung des Sperrmediums
aufweist.

This conical surface a pumping action is achieved innerhalbder seal that makes use of the
centrifugal force advantage.

Mit dieser konischen Fläche wird innerhalb der Dichtung eine Pumpwirkung erzielt, die sich die Zentrifugalkraft zu Nutze macht.

By the centrifugal force of the rotating shaft, the barrier medium is thrown outwards where it impinges on the pump channel wall.

Durch die Zentrifugalkraft der umlaufenden Welle wird das Sperrmedium nach außenabgeschleudert, wo es auf die Pumpkanalwand auftrifft.

There, the barrier medium Due to the conical shape of the pump channel wall is further conveyed.

Dort wird das Sperrmedium aufgrund der konischen Form der Pumpkanalwand weiterbefördert.

Thus, a pumping action from the inlet to the outlet of the mechanical seal is achieved on simple ways.

Somit wird auf einfache Art und Weise eine Pumpwirkung vom Zulauf zum Ablass der Gleitringdichtung erzielt.

The seal according to the invention is suitable for fast-running pumps.

Die erfindungsgemäße Dichtung eignet sich für schnell-laufende Pumpen.

To be used despite a very small design, which allows the seal, even in small rooms with small diameters, require this seal no auger or separate mechanical conveying in circulation.

Trotz einer sehr kleinen Bauweise, welche es der Dichtung ermöglicht, auch in kleinen Räumen mit kleinen Durchmessern eingesetzt werden zu können, benötigt diese Dichtung keine Schneckenförderung oder separate mechanische Förderung im Zirkulationsweg.

You are also suitable for so-called thermosyphons, which are often used for blocking media, as it were, natural circulation of the barrier medium of poetry supports will.

Sie eignen sich auch für sogenannte Thermosyphonsysteme, die für Sperrmedien häufig

eingesetzt werden, da die gewissermaßen natürliche Zirkulation des Sperrmediums von der Dichtung unterstützt wird.

Preferably, the diameter of the pump channel wall in the region of Outer mechanical seal is greater than in the region of the inner face seal, so that the sealing medium is conveyed from the inside to the outside.

Vorzugsweise ist der Durchmesser der Pumpkanalwand im Bereich der äußeren Gleitringdichtung größer als im Bereich der inneren Gleitringdichtung, sodaß das Sperrmedium von innen nach außen gefördert wird.

This has the advantage, daß zunächst die kühlere innere mechanische Dichtung durch das Sperrmedium abgekühlt wird, bevor die äußere, die meistens heißer wird, da sie nicht von außen abgekühlt werden kann.

Dies hat den Vorteil, daß zunächst die kühlere innere Gleitringdichtung von dem Sperrmedium gekühlt wird, bevor dann die äußere, die meistens heißer wird, da sie nicht von außen gekühlt werden kann.

Thus, the erwärmte Sperrmedium also cools the inside of the double mechanical seal slightly, the outer mechanical seal efficiently.

So kann das auf der Innenseite der Doppelgleitringdichtung geringfügig erwärmte Sperrmedium auch noch die äußere Gleitringdichtung effizient kühlen.

According to a preferred embodiment of the invention, the pump channel wall starts unimpeded in the amount of static seal ring of the inner mechanical seal, so that the barrier medium on the inboard side of the inner Gleitringdichtung circulate can.

Gemäß einer bevorzugten Ausführungsform der Erfindung beginnt die Pumpkanalwand in Höhe des statischen Gleitrings der inneren Gleitringdichtung, so daß das Sperrmedium ungehindert an der innen liegenden Seite der inneren Gleitringdichtung zirkulieren kann.

The other end of the pump channel wall extends, however bisetwa to the sliding surface of the external mechanical seal, so that the blocking medium is conveyed direktzu the sliding surface.

Das andere Ende der Pumpkanalwand erstreckt sich hingegen bisetwa zur Gleitfläche der äußeren Gleitringdichtung, so daß das Sperrmedium direktzu der Gleitfläche gefördert wird.

Oftentimes, the stationary seal ring has an undercut, then the pump channel wall can also facing only up to the inner dasheißt of the spectacles, the top of the undercut extending what ausreichendist for a circulation of the barrier medium in the region of the outer mechanical seal.

Oftmals weist der stationäre Gleitring einenHinterschnitt auf, dann kann sich die Pumpkanalwand auch nur bis zum inneren, dasheißt der Brille zugewandten, Anfang des Hinterschnitts erstrecken, was für eineZirkulation des Sperrmedium im Bereich der äußeren Gleitringdichtung ausreichendist.

The dynamic seal rings can be attached to one of the shaft aufschiebbarenHülse, which greatly simplifies installation.

Die dynamischen Gleitrings können an einer auf die Welle aufschiebbarenHülse befestigt sein, was die Montage erheblich vereinfacht.

Furthermore dieHülse may have an inlet recess on internal mechanical seal, the outside, that is, begins at the inner end of the pump channel wall and sichbis can continue under the pump channel wall.

Des weiteren kann dieHülse im Bereich der inneren Gleitringdichtung eine Einlaßausnehmung aufweisen, die außerhalb, das heißt vor dem inneren Ende der Pumpkanalwand beginnt und sichbis unter die Pumpkanalwand fortsetzen kann.

This inlet recess improves life flow of blocking medium in a pump channel between wall and sleeve formed pump channel.

Diese Einlaßausnehmung verbessert dasEinströmen des Sperrmedium in einen zwischen Pumpkanalwand und Hülse gebildetenPumpkanal.

It can also be provided an outlet recess in the sleeve, which begins within the pump channel wall and can also continue to pump from the duct wall out.

Es kann auch noch eine Auslaßausnehmung in der Hülse vorgesehen sein, die innerhalb der Pumpkanalwand beginnt und sich auch bis aus der Pumpkanalwand heraus fortsetzen kann.

The outlet recess facilitates the outflow of Sperrmediums aus the pump channel.

Die Auslaßausnehmung erleichtert das Ausströmen des Sperrmediums aus dem Pumpkanal.

The two recesses facilitate additional mounting der Hülse inside the goggles.

Die beiden Ausnehmungen erleichtern zusätzlich die Montage der Hülse innerhalb der Brille.

In order to improve the conveying behavior, the recesses may be konisch ausgebildet, preferably parallel to the pump channel wall.

Zur Verbessern des Förderverhaltens können die Ausnehmungen konisch ausgebildet sein, vorzugsweise parallel zu der Pumpkanalwand.

This allows the Abschleuderndes barrier medium can be improved.

Dadurch kann das Abschleuderndes Sperrmediums verbessert werden.

Can angeordneten Ausnehmungen means the outlet recess or optionally, still further between the inlet recess and the outlet recess the width of the pump channel sections are adapted to influence um die pumping action.

Mittels der Auslaßausnehmung oder gegebenenfalls noch weiterer zwischen der Einlaßausnehmung und der Auslaßausnehmung angeordneten Ausnehmungen kann die Breite des Pumpkanals abschnittsweise angepaßt werden, um die Pumpwirkung zu beeinflussen.

In the following, an embodiment of the invention is explained in more detail with reference to the drawing, wherein Fig. 1 shows a longitudinal section through the locking pressure seal.

Im folgenden wird ein Ausführungsbeispiel der Erfindung anhand der Zeichnung näher erläutert, wobei Fig. 1 einen Längsschnitt durch die Sperrdruckdichtung zeigt.

As shown in FIG. 1, a pressure-tight housing 1 has a shaft passage 2, through which a rotating shaft 3 runs.

Gemäß Fig. 1 weist ein druckdichtes Gehäuse 1 einen Wellendurchgang 2 auf, durch den eine rotierende Welle 3 verläuft.

Proceeding from the outer surface 4 of the housing 1 extends a receiving space 5, the ring-shaped space surrounds the shaft 3.

Ausgehend von der Außenfläche 4 des Gehäuses 1 verläuft ein Aufnahmeraum 5, der die Welle 3 ringförmig umgibt.

A locking pressure seal 10 seals the housing interior and the environment from against the shaft 3.

Eine Sperrdruckdichtung 10 dichtet den Gehäuseinnenraum und die Umgebung gegen die Welle 3 ab.

The outer housing wall 4 is with a seal 6 sealed to the barrier pressure seal 10.

Die Gehäuseaußenwand 4 ist mit einem Dichtring 6 gegenüber der Sperrdruckdichtung 10 abgedichtet.

The barrier pressure seal 10 is here as a double-lip seal executed; it can also be used other types of seals, while the locking pressure seals can be operated.

Die Sperrdruckdichtung 10 ist hier als Doppel-Gleitringdichtung ausgeführt; es können auch andere Dichtungstypen verwendet werden, solange diese als Sperrdruckdichtungen betrieben werden können.

A 3 surrounding the inner shaft mechanical seal 11 of the sealing pressure seal 10 is located in the receiving space 5 to seal the housing interior against the shaft 3.

Eine die Welle 3 umgebende innere Gleitringdichtung 11 der Sperrdruckdichtung 10 befindet sich in dem Aufnahmeraum 5, um den Gehäuseinnenraum gegen die Welle 3 abzudichten.

The inner mechanical seal 11 comprises a stationary seal ring 12 and a dynamic seal ring 13; the two slip rings 12, 13 come together in a sliding surface 14, the sealing surface forms between stationary and rotating parts.

Die innere Gleitringdichtung 11 besteht aus einem stationären Gleitring 12 und einem dynamischen Gleitring 13; die beiden Gleitringe 12, 13 treffen in einer Gleitfläche 14 aufeinander, die die Dichtfläche zwischen ortsfesten und rotierenden Teilen bildet.

The stationary seal ring 12 engages a shaft surrounding a 3 glasses 15, which is not gezeigter Schrauben means of several connected to the outer surface 4 of the housing 1.

Der stationäre Gleitring 12 greift in eine die Welle 3 umgebende Brille 15 ein, die mittels mehrerer nicht gezeigter Schrauben mit der Außenfläche 4 des Gehäuses 1 verbunden ist.

The stationary seal ring 12 is axially movably mounted in the spectacles 15 and sealed with a sealing ring 16 against the glasses 15.

Der stationäre Gleitring 12 ist axial beweglich in der Brille 15 gelagert und mit einem Dichtring 16 gegen die Brille 15 abgedichtet.

For rotation, the stationary seal ring 12 has a slot which is engaged by a projection of 15 glasses.

Zur Drehsicherung weist der stationäre Gleitring 12 einen Schlitz auf, in den ein Vorsprung der Brille 15 eingreift.

The dynamic seal ring 13 is attached to a fitting on the shaft 3 sleeve 17 with slight radial play and with a sealing ring 18 is sealed against the sleeve 17.

Der dynamische Gleitring 13 ist an einer auf der Welle 3 sitzenden Hülse 17 mit geringem radialen Spiel befestigt und mit einem Dichtring 18 gegen die Hülse 17 abgedichtet.

To Drehsicherung des dynamic seal ring 13 relative to the shaft 3 engages ein in a radially projecting portion 19 of the sleeve 17 eingesetzter Bolzen 20 in a slot 21 of a dynamic seal ring 13.

Zur Drehsicherung des dynamischen Gleitringes 13 gegenüber der Welle 3 greift ein in einem radial abstehenden Bereich 19 der Hülse 17 eingesetzter Bolzen 20 in einen Schlitz 21 des dynamischen Gleitringes 13 ein.

Support several circumferentially distributed springs 22 sich an der Brille 15 ab und pressen den stationären Gleitring 12 auf der Gleitfläche 14 mit dem dynamischen Gleitring 13 zusammen.

Mehrere über den Umfang verteilte Federn 22 stützen sich an der Brille 15 ab und pressen den stationären Gleitring 12 an der Gleitfläche 14 mit dem dynamischen Gleitring 13 zusammen.

On the outer side of the spectacles 15 has an outer face seal 23 is arranged, which is to the inneren Gleitringdichtung 11 in the structure is substantially symmetrical.

An der Außenseite der Brille 15 ist eine äußere Gleitringdichtung 23 angeordnet, die im Aufbau im wesentlichen symmetrisch zu der inneren Gleitringdichtung 11 ist.

Thus, the mechanical seal 23 to an outer stationary seal ring 24 and a dynamic seal ring 25, which together form a sliding surface 26.

So weist auch die äußere Gleitringdichtung 23 einen stationären Gleitring 24 und einen dynamischen Gleitring 25 auf, die zusammen eine Gleitfläche 26 bilden.

The stationary seal ring 24 of the outer mechanical seal 23 is the same as the stationary seal ring 12 of the inner shaft seal 11 attached to the goggles 15.

Der stationäre Gleitring 24 der äußeren Gleitringdichtung 23 ist ebenso wie der stationäre Gleitring 12 der inneren Gleitringdichtung 11 an der Brille 15 befestigt.

To twist grips Einan the glasses 15 attached pin 27 in a slot 28 of the stationary seal ring 24 a.

Zur Verdrehsicherung greift einan der Brille 15 befestigter Bolzen 27 in einen Schlitz 28 des stationären Gleitrings 24 ein.

The two slip rings 24, 25 of the external mechanical seal 23 are ebenfalls mit springs biased against each other.

Die beiden Gleitrings 24 , 25 der äußeren Gleitringdichtung 23 sind ebenfalls mit Federn gegeneinander vorgespannt.

The dynamic seal ring 25 of the outer mechanical seal 23 is attached to an annular attachment 29, which is connected to derHülse 17.

Der dynamische Gleitring 25 der äußeren Gleitringdichtung 23 ist an einem ringförmigen Aufsatz 29 befestigt, der mit derHülse 17 verbunden ist.

The sleeve 17 extends continuously along the entire length of the mechanical seal 10th

Die Hülse 17 erstreckt sich durchgängig über die gesamte Länge der Gleitringdichtung 10 .

On the inside isthe sleeve 17 with a sealing ring 30 against the Gehäuseinnenraum abgedichtet.

An der Innenseite ist die Hülse 17 mit einem Dichtring 30 gegen den Gehäuseinnenraum abgedichtet.

On the outside of the sleeve has a plurality of circumferentially distributed 17 holes through each of which a set screw is screwed 29a of the annular attachment 29 against the shaft so that the annular attachment 29 is fixed to the shaft 3.

An der Außenseite weist die Hülse 17 mehrere über den Umfang verteilte Löcher auf, durch die jeweils eine Setzschraube 29a des ringförmigen Aufsatzes 29 gegen die Welle verschraubt wird, so daß der ringförmige Aufsatz 29 auf der Welle 3 fixiert ist.

With a snap ring 31, which connects the annular attachment 29 to the sleeve 17, the sleeve 17 is axially secured to the shaft 3.

Mit einem Sprengring 31, der den ringförmigen Aufsatz 29 mit der Hülse 17 verbindet, wird die Hülse 17 axial an der Welle 3 gesichert.

The glasses 15 has an inlet 32 for the sealing medium.

Die Brille 15 weist einen Zulaufkanal 32 für das Sperrmedium auf.

The inlet 32 has a, extends with a threaded portion 32a extends from the housing 1 facing away from the front side of the glasses 15 in respect to the shaft 3 axial direction in glasses 15.

Der Zulaufkanal 32 hat einen mit einem Gewinde versehenen Bereich 32a, der sich von der dem Gehäuse 1 abgewandten Stirnseite der Brille 15 ausgehend, in bezüglich der Welle 3 axialer Richtung in die Brille 15 erstreckt.

This area is in a radially extending portion 32b of the inlet channel 32 on the other end side in a central region 33 of the spectacle is, where the central region 33 axially in the middle between the two stationary seal rings 12, 24 is.

Dieser Bereich geht in einen radial verlaufenden Bereich 32b des Zulaufkanals 32 über, dessen

anderes Ende sich im Zentralbereich 33 der Brille befindet, wobei der Zentralbereich 33 sich axial in der Mitte zwischen den beiden stationären Gleitringen 12, 24 befindet.

From there, the barrier medium enters a gap which is bounded by the inner face seal 11 and the sleeve 17.

Von dort gelangt das Sperrmedium in einen Spalt, der von der inneren Gleitringdichtung 11 und der Hülse 17 begrenzt wird.

Further moves the barrier medium between the central portion 33 of the spectacle and the sleeve 17 passes.

Weiterhin bewegt sich das Sperrmedium zwischen dem zentralen Bereich 33 der Brille und der Hülse 17 hindurch.

In the central portion 33 is a pump channel wall 34 is formed in the eyeglass 15 which extends in the axial direction approximately from the center of the stationary seal ring 12 of the inner face seal 11 to a rear section 24a of the stationary seal ring 24 of the outer mechanical seal 23.

In dem zentralen Bereich 33 ist in der Brille 15 eine Pumpkanalwand 34 ausgebildet, die sich in axialer Richtung etwa von der Mitte des stationären Gleitrings 12 der inneren Gleitringdichtung 11 bis zu einem Hinterschnitt 24a des stationären Gleitrings 24 der äußeren Gleitringdichtung 23 erstreckt.

The pump channel wall 34 is conical, which increases the diameter in the direction of the outer mechanical seal 23.

Die Pumpkanalwand 34 ist konisch ausgebildet, wobei sich der Durchmesser in Richtung der äußeren Gleitringdichtung 23 vergrößert.

Formed between the pump channel wall 34 and the sleeve 17 is a pumping channel 35, through which the barrier medium is transported from the inner mechanical seal 11 to the external

mechanical seal 23 in the arrow direction.

Zwischen der Pumpkanalwand 34 und der Hülse 17 wird ein Pumpkanal 35 gebildet, durch den das Sperrmedium von der inneren Gleitringdichtung 11 zu der äußeren Gleitringdichtung 23 in Pfeilrichtung gefördert wird.

In the sleeve 17 an inlet recess 36 is formed in front of the inner end of the pump channel wall 34.

In der Hülse 17 ist vor dem inneren Ende der Pumpkanalwand 34 eine Einlaßausnehmung 36 ausgebildet.

This inlet recess 36 starts approximately in the middle of the stationary seal ring 12 der inneren mechanical seal 11 and continues until the middle of the pump channel wall 34.

Diese Einlaßausnehmung 36 beginnt etwa in der Mitte des stationären Gleitrings 12 der inneren Gleitringdichtung 11 und setzt sich bis etwa zur Mitte der Pumpkanalwand 34 fort.

At the end of the inlet recess 36, the sleeve 17 reaches again its original diameter.

Am Ende der Einlaßausnehmung 36 erreicht die Hülse 17 wieder ihren ursprünglichen Durchmesser.

The inlet recess 36 is as the pumping channel wall 34 conically namely WITH THE same angle, so that the surfaces of the pump channel wall 34 and the recess 36 Einlaßaus parallel.

Die Einlaßausnehmung 36 ist wie die Pumpkanalwand 34 konisch ausgebildet und zwar mit dem gleichen Winkel, so daß die Flächen der Pumpkanalwand 34 und der Einlaßausnehmung 36 parallel verlaufen.

At the end of the inlet recess 36 an outlet recess 37, 34 which extends almost to the outer end of the pump channel wall begins.

Am Ende der Einlaßausnehmung 36 beginnt eine Auslaßausnehmung 37 , die fast bis zum äußeren Ende der Pumpkanalwand 34 verläuft.

The area of the outlet recess 37 is also parallel to the duct wall Flächeder pumping 34th

Die Fläche der Auslaßausnehmung 37 ist ebenfalls parallel zu der Flächeder Pumpkanalwand 34

After the barrier medium ausgetretenist from the feed channel 32, first it circulates in the arrow direction on the inner face seal 11 along.

Nachdem das Sperrmedium aus dem Zulaufkanal 32 ausgetretenist, zirkuliert es zunächst in Pfeilrichtung an der inneren Gleitringdichtung 11 entlang.

By rotating shaft 3 is the Sperrmediummitgenommen, thus resulting in a helical motion of the barrier medium.

Durch die umlaufende Welle 3 wird das Sperrmediummitgenommen, so daß sich eine schraubenförmige Bewegung des Sperrmediums ergibt.

Now the barrier medium passes via the inlet recess 36 in the pump channel 35th

Nun gelangt das Sperrmedium über die Einlaßausnehmung 36 in den Pumpkanal 35 .

There, it is spun off vondieser 3 due to centrifugal force of the shaft and impinges on the conical wall 34 of the pump channel.

Dort wird es aufgrund der Zentrifugalkraft der Welle 3 vondieser abgeschleudert und trifft auf die konische Pumpkanalwand 34 auf.

Here, the barrier fluid is outward, so pushed toward the outer mechanical seal 23.

Hier wird das Sperrmedium nach außen, also in Richtung der äußeren Gleitringdichtung 23

gedrückt.

In the area of outlet recess 37 of Querschnittdes pumping channel 35 is expanded, so that the barrier medium without hindrance from demPumpkanal 35 can emerge under the external mechanical seal 23 where the barrier medium for cooling circulates before it, via a return channel 38 which is formed as the inlet 32 the confining pressure seal 10 leaves.

Im Bereich der Auslaßausnehmung 37 ist der Querschnittdes Pumpkanals 35 erweitert, so daß das Sperrmedium ungehindert aus demPumpkanal 35 unter die äußere Gleitringdichtung 23 austreten kann,wo das Sperrmedium zur Kühlung zirkuliert, bevor es über einen Rücklaufkanal 38 , der wie der Zulaufkanal 32 ausgebildet ist, die Sperrdruckdichtung 10 verläßt.

In the described circulation path 32, 36, 35, 37 and 38 flows through the barrier medium zurAbführung of heat and to lubricate the mechanical seals 11 and 23rd

In dem beschriebenen Zirkulationsweg 32 , 36 , 35 , 37 und 38 strömt das Sperrmedium zurAbführung von Wärme und zur Schmierung der Gleitringdichtungen 11 und 23 .

By the pumping channel wall 34 and the inlet and outlet recesses 36, 37 an integrated pumping action is obtained which leaves the barrier medium by circulating the sealing pressure seal; can be dispensed with at least largely to external pumping means.

Durch die Pumpkanalwand 34 und die Einlaß- und Auslaßausnehmungen 36 , 37 wird eine integrierte Pumpwirkung erzielt, die das Sperrmediumdurch die Sperrdruckdichtung zirkulieren läßt; so kann auf externe Pumpmittel zumindestweitgehend verzichtet werden.

The pumping action generated depends not only on the Drehzahl der shaft 3 and the size of the seal also depends on the size of the pumping channel 35.

Die erzielte Pumpwirkung hängt neben der Drehzahl der Welle 3 und der Größe der Dichtung auch von der Größe des Pumpkanals 35 ab.

The size can be varied by the distance between the pump channel wall 34 to the sleeve 17 and through the design of the recesses 36, 37.

Die Größe kann durch den Abstand der Pumpkanalwand 34 zuder Hülse 17 und durch die Ausgestaltung der Ausnehmungen 36 , 37 variiert werden.

When mounting the locking ring 10 will seal these einfach auf the shaft 3 is pushed and the stationary parts of the locking pressure seal 10 with the glasses 15 are secured to the housing 1.

Bei der Montage der Sperrringdichtung 10 wird diese einfach auf die Welle 3 aufgeschoben und die stationären Teile der Sperrdruckdichtung 10 mit der Brille 15 werden an dem Gehäuse 1 befestigt.

To secure the dynamic parts of the barrier pressure seal 10 werden die set screws 30 tightened.

Zur Befestigung der dynamischen Teile der Sperrdruckdichtung 10 werden die Setzschrauben 30 angezogen.

To and during der Montage to prevent misalignment of the locking pressure seal 10 during transport, the outside of the glasses sind an mounted 15 adjustment screws 39.

Um während des Transports und während der Montage eine Dejustierung der Sperrdruckdichtung 10 zu verhindern, sind an der Außenseite der Brille 15 Justierschrauben 39 angebracht.

Each adjustment screw 39 secures an adjustment sleeve 40 that has a radially projecting projection 41.

Jede Justierschraube 39 sichert eine Justierhülse 40 , die einen radial abstehenden Vorsprung 41 aufweist.

The adjusting screw is released 39, the adjustment sleeve 40 can be rotated so that der Vorsprung 41 engages in an annular groove 42 of the annular attachment 29 and thus ensures that the inner seal 11 and die äußere mechanical seal 23 under the desired prestress.

Ist die Justierschraube 39 gelöst, kann die Justierhülse 40 gedreht werden, so daß der Vorsprung 41 in eine ringförmige Nut 42 des ringförmigen Aufsatzes 29 eingreift und somit die innere

Gleitringdichtung 11 und dieäußere Gleitringdichtung 23 unter der gewünschten Vorspannung sichert.

After the sealing pressure seal 10 is attached to both the housing 1 alsoach to the shaft 3, the adjusting screws 39 can be loosened and unscrewed, the projections 41 out of engagement.

Nachdem die Sperrdruckdichtung 10 sowohl an dem Gehäuse 1 alsoach an der Welle 3 befestigt ist, können die Justierschrauben 39 gelöst und die Vorsprünge 41 aus dem Eingriff herausgedrehtwerden.

The Justierhülsen 40 are turned away for later use with demVorsprung 41 of the groove 42 bolted back to the spectacles 15.

Die Justierhülsen 40 werden zur späteren Wiederverwendung mit demVorsprung 41 von der Nut 42 abgewandt wieder an der Brille 15 festgeschraubt.

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CLAIMS DE20217585

Confining pressure seal with pumping action, which is for sealing a rotating shaft (3) against a housing (1) at this fastened, comprising: the shaft (3) surrounding on the housing (1) anbringbarenBrille (15) having a feed channel (32) and a return channel (38) for a sealing medium, a shaft (3) against denGehäuseinnenraum sealing internal mechanical seal (11) and einerdie shaft (3) from the environment sealing the external mechanical seal (23), each mechanical seal (11,23) two oppositely biased sliding rings (12,13; 24,25), whereinthe a circulation path (32,36,35,37,38) for the barrier medium between the channels (32,38) and along the seals (11,23) extends, characterized in that the spectacles (15) in the central region (33) between the two mechanical seals (11,23) has a smooth conical widening inStrömungsrichtung pump channel wall (34) for Förderungdes barrier medium.

Sperrdruckdichtung mit Pumpwirkung, die zur Abdichtung einer rotierendenWelle (3) gegen ein Gehäuse (1) an diesem befestigbar ist, miteiner die Welle (3) umgebenden, an dem Gehäuse (1) anbringbarenBrille (15), die einen Zulaufkanal (32) und einen Rücklaufkanal(38) für ein Sperrmedium aufweist, einer die Welle (3) gegen denGehäuseinnenraum abdichtenden inneren Gleitringdichtung (11) und einerdie Welle (3) gegen die Umgebung abdichtenden äußeren Gleitringdichtung(23), wobei jede Gleitringdichtung (11,23) zwei gegeneinandervorgespannte Gleitringe (12,13;24,25) aufweistund wobei ein Zirkulationsweg (32,36,35,37,38) für das Sperrmedium zwischen den Kanälen (32,38) und entlang der Gleitringdichtungen (11,23) verläuft,dadurch gekennzeichnet, daß die Brille (15) im Zentralbereich (33)zwischen den beiden Gleitringdichtungen (11,23) eine sich inStrömungsrichtung erweiternde glatte konische Pumpkanalwand (34) zur Förderungdes Sperrmediums aufweist.

Confining pressure seal according to claim 1, characterized in that the pumping derDurchmesser channel wall (34) increases in the direction of the outer seal (23).

Sperrdruckdichtung nach Anspruch 1, dadurch gekennzeichnet, daß der Durchmesser der Pumpkanalwand (34) in Richtung der äußeren Gleitringdichtung (23) zunimmt.

Confining pressure seal according to claim 1 or 2, characterized in that the pump channel wall (34) extends from the static seal ring (12) of the inner seal (11) to a between the statischen Gleitring (24) and the dynamic seal ring (25) formed sliding surface (26) of the outer mechanical seal (23).

Sperrdruckdichtung nach Anspruch 1 oder 2, dadurch gekennzeichnet, daß die Pumpkanalwand (34) sich von dem statischen Gleitring (12) der inneren Gleitringdichtung (11) bis zu einer zwischen dem statischen Gleitring (24) und dem dynamischen Gleitring (25) gebildeten Gleitfläche (26) der äußeren Gleitringdichtung (23) erstreckt.

Confining pressure seal according to one of claims 1 to 3, characterized in that the pump channel wall (34) (24) of the outer mechanical seal (23) extends to a rear section (24a) of the stationary seal ring.

Sperrdruckdichtung nach einem der Ansprüche 1 bis 3, dadurch gekennzeichnet, daß die Pumpkanalwand (34) sich bis zu einem Hinterschnitt (24a) des stationären Gleitrings (24) der äußeren Gleitringdichtung (23) erstreckt.

Confining pressure seal according to one of claims 1-4, characterized in that to the shaft (3) attachable dynamic seal rings (13,25) can be pushed at one of the shaft (3) sleeve (17) are mounted, the through the lens (15) protrudes, the pump channel and a wall (34) has opposed generally cylindrical outer surface.

Sperrdruckdichtung nach einem der Ansprüche 1-4, dadurch gekennzeichnet, daß die an der Welle (3) anbringbaren dynamischen Gleitringe (13,25) an einer auf die Welle (3) aufschiebbarer Hülse (17) angebracht sind, die durch die Brille (15) hindurchragt und eine der Pumpkanalwand (34) gegenüberliegende generell zylindrische Außenfläche hat.

Confining pressure seal according to claim 5, characterized in that die Hülse (17) in the region of the inner seal (11) eine Einlaßausnehmung (36), which begins outside the pump channel wall (34).

Sperrdruckdichtung nach Anspruch 5, dadurch gekennzeichnet, daß die Hülse (17) im Bereich der inneren Gleitringdichtung (11) eine Einlaßausnehmung (36) aufweist, die außerhalb der Pumpkanalwand (34) beginnt.

Confining pressure seal according to claim 5 or 6, characterized in that the sleeve (17) has a outlet recess (37), the innerhalbder pump channel wall (34) begins.

Sperrdruckdichtung nach Anspruch 5 oder 6, dadurch gekennzeichnet, daß die Hülse (17) eine Auslaßausnehmung (37) aufweist, die innerhalbder Pumpkanalwand (34) beginnt.

Confining pressure seal according to claim 6 or 7, characterized in that the inlet recess (36) and / or the outlet recess (37) are conical.

Sperrdruckdichtung nach Anspruch 6 oder 7, dadurch gekennzeichnet, daß die Einlaßausnehmung (36) und/oder die Auslaßausnehmung (37) konisch ausgebildet sind.

Confining pressure seal according to claim 8, characterized in that the pumping dieFlächen duct wall (34) and the inlet recess (36) and / or the outlet recess (37) are parallel.

Sperrdruckdichtung nach Anspruch 8, dadurch gekennzeichnet, daß dieFlächen der Pumpkanalwand (34) und der Einlaßausnehmung (36) und/oder der Auslaßausnehmung (37) parallel verlaufen.

Confining pressure seal according to one of claims 1-9, characterized in that the pump channel wall (34) has a diameter which is smaller alsdie inner diameter of the seal rings.

Sperrdruckdichtung nach einem der Ansprüche 1–9, dadurch gekennzeichnet, daß die Pumpkanalwand (34) einen Durchmesser hat, der kleiner ist alsdie Innendurchmesser der Gleitringe.

It follows a sheet drawings

Es folgt ein Blatt Zeichnungen

(19)



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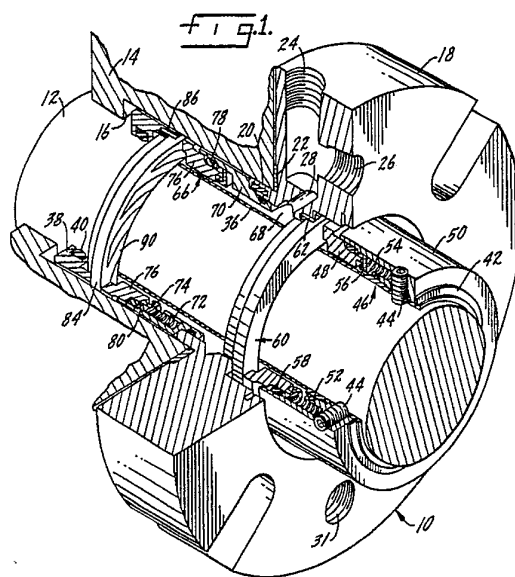
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W-8000 München 22(DE)(54) **Improvements in barrier seal systems.**

(57) This invention relates to barrier seal systems for sealing the annular space between a housing and a rotating shaft which extends through a bore in the housing. The seal system has inboard and outboard mechanical end face seals. The inboard seal faces have spiral grooves for pumping a buffer fluid from a buffer chamber between the two seals toward the high pressure side of the inboard seal. The primary ring of the inboard seal may be double balanced. For high pressure applications, the buffer fluid may be pressurized and a pumping ring may be included in the outboard seal to circulate buffer fluid. High pressure applications also require two hard seal rings and optimum dam and face widths. Buffer fluid can be supplied to the seal system by a reservoir connected by a gravity feed line. A buffer fluid containment device may be attached to the high pressure side of the seal to prevent process fluid from contaminating the seal faces or secondary seals.

**EP 0 447 898 A2**

Summary of the Invention

This invention relates to barrier seal systems which are suitable for sealing a rotating shaft extending through a bore in a housing. The housing may be part of a pump, compressor or similar equipment containing a high pressure fluid. The seal system includes inboard and outboard mechanical end face seals, each having primary and mating rings. The space between the inboard and outboard seals defines a buffer chamber. A buffer fluid is supplied to the buffer chamber. The inboard seal pumps buffer fluid across the seal faces toward the high pressure side of the seal. Seals of this general nature are disclosed in Sedy, U.S. Patent No. 4,290,611.

One of the objects of the present invention is a barrier seal system of the type described which reduces or eliminates the requirement for buffer fluid pressure, minimizes potential emissions and simplifies the buffer fluid supply equipment.

Another object of the invention is a barrier seal system in which the inboard seal is double balanced so that the inboard seal will function in the event of pressure reversals.

Another object of the invention is a barrier seal system which is operable in high pressure applications. Barrier seal systems according to the present invention which are intended for use in high pressure environments have hard primary and mating rings to resist deflection. High pressure seals also have optimum face and dam widths to maximize the pressure developed by the inboard seal.

A further object of the invention is a barrier seal system which is pressurized on the outside diameter of the inboard seal faces, thereby utilizing centrifugal force to prevent fouling of the seal faces and secondary seal.

Yet another object of the invention is a barrier seal system of the type described which includes a buffer fluid containment element forming a containment chamber on the high pressure side of the inboard seal faces. The containment chamber prevents the process fluid from contacting secondary seals or the primary and secondary ring seal faces.

Other objects may appear from time to time in the following specification, drawings and claims.

The above objects are realized by a seal system having a gland plate connected to the housing of the equipment to be sealed. A sleeve is mounted on the shaft for rotation therewith. A first static seal in the form of an O-ring is placed between the sleeve and shaft to prevent leakage along the shaft. An outboard mechanical end face seal has a mating ring connected to the gland plate and a seal head connected to the sleeve for rotation with the shaft. The seal head includes a retainer and a plurality of springs disposed between the retainer

and primary ring, urging the primary ring into contact with the mating ring. A secondary seal is located between the retainer and the primary ring.

The barrier seal further includes an inboard mechanical end face seal, at least a portion of which is mounted in the annular space between the housing and the rotating shaft. The inboard seal has a primary ring and a retainer fixed to the gland plate. The retainer at least partially encloses the primary ring to fix its radial position while allowing axial movement of the primary ring. A secondary seal is disposed between the retainer and the primary ring. A mating ring of the inboard seal surrounds the sleeve and is connected thereto for rotation with the shaft. A static seal is placed between the sleeve and the mating ring to prevent leakage between them.

The mating and primary rings of the inboard seal have opposed, relatively rotating, radial seal faces on one side thereof. The outer diameters of the seal faces are exposed to the pressure of the process fluid.

The inboard seal is positioned axially of the outboard seal to define a buffer chamber. The mating ring of the inboard seal has grooves formed therein which extend from the buffer chamber partially across the mating ring seal face to a non-grooved dam portion. The grooves pump buffer fluid across the seal faces into the high pressure side of the inboard seal.

The primary ring of the inboard seal is arranged to be double balanced so that pressure reversals can be tolerated without loss of closing force. The buffer fluid may be gravity fed to the buffer chamber from a reservoir. A buffer fluid containment element spanning the inboard primary and secondary rings and in contact therewith defines a containment chamber which prevents process fluid from contacting the seal faces or the secondary seal of the inboard seal.

Brief Description of the Drawings

Fig. 1 is a perspective view, with portions cut away, of the barrier seal system according to the present invention.

Fig. 2 is a section taken along an axial, vertical plane of the seal of Fig. 1, on an enlarged scale.

Fig. 3 is a front elevation view of a portion of a mating ring seal face, showing the spiral grooves.

Fig. 4 is a schematic view of the barrier seal system, illustrating the buffer fluid supply arrangement.

Fig. 5 is a section similar to Fig. 2 of an alternate embodiment of the invention.

Fig. 6 is a diagrammatic detail section through a primary ring and a portion of a retainer, illustrating the double balance design of the primary ring.

Cross hatching is omitted for clarity.

Fig. 7 is an enlarged detail view in section of a buffer containment element.

Fig. 8 is similar to Fig. 7, showing an alternate embodiment.

Fig. 9 is similar to Fig. 7, showing another alternate embodiment.

Detailed Description of the Invention

Figs. 1 and 2 illustrate a barrier seal system 10 according to the present invention. The seal system 10 seals the annular space between a rotating shaft 12 and a housing 14. The shaft 12 extends through a bore 16 in the housing 14. The housing 14 contains a high pressure fluid on the side of the seal labeled -Pressure- in Fig. 2. The exterior of the seal system is labeled -Atmosphere-. The specific seal shown is intended to contain a fluid in liquid form, the liquid being under pressure.

A gland plate assembly 18 is bolted to the outer face of the housing. The gland plate has a boss 20 interfitting with the bore 16 to locate the gland plate. A gasket 22 prevents leakage between the bore 16 and gland plate 18.

The gland plate has a set of upper, vertical and horizontal taps 24 and 26, respectively. An internal passage 28 is in fluid communication with the taps 24 and 26. A buffer fluid supply line from a reservoir connects to one of the taps. The threaded taps are provided in both the vertical and horizontal orientations to accommodate the most convenient physical connection of the supply line for a particular installation. Only one of the taps will be used, the other one will be plugged. Further, looking at Figs. 1 and 4, it can be seen that a similar lower set of vertical and horizontal taps 29, 31 is provided in the lower half of the gland plate. One of these taps is also used by a buffer fluid line connecting to the reservoir. Connections of the reservoir to the gland plate will be explained in further detail below.

The seal system 10 has an elongated sleeve 36 surrounding the shaft 12. The sleeve extends from the exterior of the gland plate to a point well inside the bore 16 where it terminates at an integral flange 38. A seat is built into the flange to retain an O-ring 40, which prevents leakage between the sleeve and the shaft. At the exterior end of the sleeve there is a groove into which a snap ring 42 fits. The sleeve is held in place on the shaft by a plurality of set screws 44.

The snap ring 42 retains an outboard seal shown generally at 46. The outboard seal is preferably a contacting mechanical end face seal. The outboard seal has a rotating seal head which includes a primary ring 48, held in place by a retainer 50. A static O-ring 52 prevents leakage be-

tween the retainer and sleeve. The seal head further includes a plurality of springs 54 engageable between the retainer and a drive ring 56. The seal head is completed by a secondary seal formed by O-ring 58.

The springs 54 urge the primary ring 48 into engagement with the face of a mating ring 60. The mating ring is stationary as it is fixed to the gland plate by a plurality of pins 62. The mating ring is sealed to the gland plate by an O-ring 64.

An inboard seal 66 is axially spaced from the outboard seal 46. The inboard and outboard seals 66 and 46, together with the sleeve 36 and gland plate 18, define a buffer chamber 68. The buffer chamber is in communication with the passage 28 in the gland plate so that buffer fluid is supplied to the buffer chamber from the reservoir.

Looking now at the details of the inboard seal 66, it has a stationary seal head including a retainer 70 which is pinned to the boss 20 of the gland plate. The retainer carries a plurality of springs 72 which urge a drive ring 74 into engagement with a primary ring 76. The drive ring has a plurality of tabs which engage depressions in the back face of the primary ring to rotationally lock the retainer and primary ring together. A secondary seal 78 is held between the retainer 70 and the primary ring 76. A disk 80 may also be positioned against a shoulder of the retainer to assist in holding the secondary seal in place against the process pressure.

The primary ring has a plurality of axial passages or grooves 82 (Fig. 2) formed at its inside diameter. The grooves 82 facilitate movement of the buffer fluid through the buffer chamber 68 to the seal faces.

The springs 72 urge the primary ring 76 into engagement with a mating ring 84. The mating ring is pinned to the flange 38 of the sleeve by pins 86. A static O-ring 88 prevents leakage between the flange 38 and the mating ring 84.

The mating ring 84 has a radial seal face 90 on one side thereof. The other side of the mating ring may be termed a back face, which, in this case, includes radial and beveled surfaces 92. Similarly, the primary ring 76 has a seal face 94 on one side and a back face on the other side which includes several radial surfaces. Details of the primary ring surfaces will be explained below.

Details of the mating ring's seal face are shown in Fig. 3. The mating ring face 90 has a plurality of spiral grooves 98 formed therein. The spiral grooves extend from the inside diameter of the seal face partially across the face. The ungrooved portion of the face forms a dam 100 which extends from the outside diameter of the grooved portion to the outside diameter of the mating ring. The face width is defined as half the difference between the inside and outside diameters of the face of the ring

(primary or mating) having the smaller face outside diameter. The dam width is defined as the difference between the outside diameter of the primary ring and the outside diameter of the spiral grooves. For reference purposes only, the face width of the seal in Fig. 1 is .150 inches and the dam width is .050. The spiral grooves have a depth of about 180 micro-inches.

The grooves pick up buffer fluid from the buffer chamber and accelerate it against the dam 100, thereby increasing the pressure of the buffer fluid to a level greater than that of the process fluid in the housing. This prevents leakage of process fluid from the high pressure side of the seal to the buffer chamber. In fact, the rotating mating ring pumps some buffer fluid from the relatively low pressure buffer chamber to the high pressure side of the seal. This results in a small flow of buffer fluid into the process fluid.

Fig. 4 illustrates how buffer fluid is supplied to make up for the losses caused by the pumping of the inboard seal. Fig. 4 illustrates a complete apparatus in schematic form. A motor 102 engages shaft 12 through a coupling 104. The shaft drives a pump 106, which includes housing 14 and bore 16. Barrier seal system 10 is shown in place in the bore. The upper horizontal tap 26 is shown connected to the buffer fluid reservoir 30 through a pipe or tube 108. The lower horizontal tap 31 also receives buffer fluid through a pipe 112. The supply reservoir has a float 32 connected to an auto-fill valve 34. The valve 34 is connectable to a buffer fluid supply line (not shown). One of the lines 108 or 112 may be finned to facilitate heat removal, while the other may be insulated to establish a thermal siphon. The thermal siphon is an optional feature which may be used to enhance the gravity feed of buffer fluid to the buffer chamber.

Turning now to Fig. 5, an alternate form of the barrier seal system is shown generally at 114. This embodiment is intended for use in high pressure, high temperature applications. The seal system of Fig. 5 will withstand process pressures up to 1400 psi and temperatures of 700° F. In this embodiment the gland plate 116 is bolted to a liner assembly 118; these parts are bolted to the housing 120. The liner has an elongated extension 122 which extends into the bore 124 of the housing. The liner is sealed to the housing by O-ring 126. The liner also has a buffer fluid outlet tap 128 in communication with a fluid passage 130. Gland plate 116 includes a similar tap 132 and passage 134. These may actually be in the lower half of the gland plate, but are shown in phantom for convenience.

The seal system includes a sleeve 136 surrounding the shaft 138 and fastened thereto by a collar 140 held by set screws 142. O-rings 144 seal the sleeve against the shaft.

The outboard seal includes a stationary mating ring 146 pinned to an extension portion of the gland plate 116. An O-ring 148 seals the mating ring 146 against the gland plate.

The seal head includes a retainer 150 carrying springs 152 which contact a disk 154 and urge a primary ring 156 into engagement with the mating ring 146. A secondary seal 158 is held between the sleeve and primary ring. The retainer carries a pumping ring 160 which circulates buffer fluid from an outer buffer chamber 161 to an inner buffer chamber 163. These chambers are in communication with the passages 134 and 130, respectively.

The inboard seal includes a mating ring 162 which is pinned to the flange of the sleeve 136. The mating ring has a seal face 164 on one side thereof. The seal face is similar to that shown in Fig. 3. Leakage between the mating ring and sleeve is prevented by a static seal 166.

The seal head of the inboard seal includes a retainer 168 which is pinned to the liner assembly 118. Springs 170 act upon a disk 172 and the retainer 168 to urge a primary ring 174 into sealing engagement with the mating ring 162. A secondary seal 176 is disposed between the retainer and the primary ring. The primary ring has a radial seal face 178 opposite that of the mating ring. The primary ring also has an ear 180 engaging a notch 182 in the retainer.

Buffer fluid is supplied to the inlet tap 132 through a line 184, which is connected to a reservoir 186. The reservoir is pressurized by a nitrogen bottle 188 connected to the reservoir through line 190 and pressure regulator 192. Buffer fluid returns to the reservoir from outlet tap 128 through return line 194. The nitrogen bottle pressurizes the buffer fluid and, in turn, the buffer chambers 161, 163 to about 200 psi. The buffer pressure is required to urge the buffer fluid into the spiral grooves of the inboard mating ring. Even with the pressurized buffer fluid, the pressure differential between the buffer fluid and process fluid may be on the order of 1200 psi. It has been found that to increase the buffer fluid pressure this amount requires a different seal face configuration than previously used. Specifically, it has been found that where the process fluid is crude, NGL or ethane, and the buffer fluid is condensate at 200 psi and with a shaft speed of about 3550 rpm, a face width of .312 inches and a dam width of .093 inches will produce the required pressure in order to pump buffer fluid from the buffer chamber into the process fluid.

Another feature of the seal of Fig. 5 is the use of high strength materials for both the primary ring and mating ring of the inboard seal. For example, the primary ring may be made of tungsten carbide and the mating ring may be silicon carbide. Silicon carbide has a modulus of elasticity of 56×10^6 psi

while that of tungsten carbide is 89×10^6 . Thus, these materials will resist deflection under the process pressure. For purposes of comparison, the modulus of carbon-graphite (a commonly-used seal ring material) can range from about 1.8 to 4 or 5×10^6 psi. If low modulus materials are used for the rings, they become subject to deformation, which can distort the parallel arrangement of the seal faces. The seal faces must remain parallel to operate as intended. For purposes of this application, materials having a modulus of elasticity which is an order or magnitude greater than plain carbon will be termed high modulus materials.

It will be noted that the embodiments of both Figs. 1 and 5 are arranged such that the process pressure is present at the outside diameters of the seal rings. This is referred to as an O.D. pressurized seal. While O.D. pressurization is shown and preferred, it will be understood that an I.D. pressurized arrangement could be used with the present invention. An advantage of O.D. pressurization is the centrifugal force generated by the rotating seal member on the process fluid tends to project the fluid away from the seal faces. This helps prevent contamination of the seal faces, particularly by particulates that may be carried in the process fluid.

It has been noted above that the inboard seals in the embodiments of both Figs. 1 and 5 are double balanced. The meaning and significance of this will be explained in conjunction with Fig. 6. Several details of the primary ring and retainer are shown in Fig. 6. The primary ring has a cylindrical seat defined by an axial ledge 196 and a radial shoulder 198. There is also a notch having surfaces 200 and 202. The notch receives the end of the retainer 70.

The front face of the primary ring includes the seal surface 94 and a front neutral surface 204. The back face of the primary ring includes shoulders 198 and 202 and a back neutral surface 206. The back neutral surface 206 is that portion of the back face which is opposite the front neutral surface 204, i.e., the portion below dotted line 207. These faces are termed "neutral" because they cancel one another in the balance calculation, as will be explained. Line 207 also defines a smaller surface 208 between neutral surface 206 and the ledge 196.

The retainer 70 has a notch defined by a radial shoulder 210 and an axial surface 212.

These various components of the retainer and primary ring define a series of diameters, including an outside diameter D of the primary ring face, an internal pressurization balance diameter B, an external pressurization balance diameter b and an internal diameter of the seal face d.

Seal balance relates to the axial forces on the

primary ring due to the pressure of the process fluid. These forces, together with the spring force, determine how tightly the primary ring fits against the mating ring, thereby governing the gap between them, which, in turn, is an important factor in determining overall seal performance. The process fluid pressure will act on both sides of the primary ring. Since the pressure is acting on both the front and back sides of the primary ring, the ratio of the areas of those surfaces becomes critical. The seal balance, therefore, is defined as the ratio of the area of the primary ring subject to a force which will close the primary ring against the mating ring to the area of the primary ring subject to forces which will open the gap between the primary ring and mating ring. In short, the seal balance is the area of closing divided by the area of opening.

Looking specifically at Fig. 6, a pressure gradient will act on seal face 94, tending to open the seal gap. Thus, the static area of opening is simply the seal face 94, which is $\pi (D^2 - d^2) / 4$.

The area of closing depends on whether the seal is externally pressurized or internally pressurized. Consider first the case of internal pressurization. The process pressure is then located in the area designated with the letter A. The full process pressure will operate on front neutral surface 204, back neutral surface 206 and surface 208. Internal pressure will also force the secondary seal 78 against the radial shoulder 198, thereby effectively operating on a surface to the limit of the axial surface 212. Forces operating on the neutral surfaces 204 and 206 operate in opposite directions and effectively cancel each other. Thus, the area of closing becomes the area between the internal pressurization balance diameter B and the internal seal face diameter d. Thus, the area of closing equals $\pi (B^2 - d^2) / 4$. So for internal pressurization the balance equals $B^2 - d^2 / D^2 - d^2$. This ratio is chosen to be about .8. Generally speaking, seals having a balance of less than one are said to be balanced, while seals having a balance greater than one are said to be unbalanced or overbalanced.

Considering the case of external pressurization or O.D. pressurization, the process pressure operates at the location indicated at B in Fig. 6. The pressure moves the O-ring 78 against the axial shoulder 210, thereby leaving the pressure to operate on faces 198 and 202. So the area of closing for external pressurization becomes $\pi (D^2 - b^2) / 4$. The static area of opening remains the same, namely, $\pi (D^2 - d^2) / 4$. So the balance for O.D. pressurization is $(D^2 - b^2) / (D^2 - d^2)$. This is also chosen to be about .8. Thus, the balance for both the internal and external pressurization is the same. This is said to be a double balanced primary ring.

It will be noted that the physical arrangement that brings about double balancing in this case is

that the internal and external balance diameters B, b are symmetrical about the centerline c of the primary ring seal face 94. As a result, the area of closing for internal pressurization approximately equals the area of closing for external pressurization. Consequently, of course, the balance figures for both cases are the same and the seal is said to be double balanced. With double balancing the seal has the same balance whether the primary ring sees internal pressure or external pressure. This is important because if for some reason the internal seal stopped pumping, it could still function as a contacting seal. Also, in arrangements such as Fig. 5 where there is some buffer pressure, a double balanced seal can withstand pressure reversals. That is, ordinarily the seal of Fig. 5 expects to have O.D. pressurization from the process fluid. If, for some reason, the process fluid pressure drops below that of the buffer chamber, the seal would become internally pressurized. If the seal were not double balanced, it would blow open upon the occurrence of such a pressure reversal.

Turning now to another aspect of the invention, Figs. 7-9 show three embodiments of a containment means located on the pressurized side of the seal faces and defining a containment chamber. The containment chamber retains buffer fluid pumped across the seal faces while preventing entrance of process fluid into the containment chamber. This is important because in pusher-type seals, such as in Figs. 1 and 5, the process fluid is exposed to the secondary seal. Fluid solidifying in the O-ring area of the secondary seal may eventually restrict primary ring movement, resulting in seal leakage as the faces hang open. While an internal flush may be provided to prevent this, the equipment required to do so is expensive. It also results in flush flow in the range of 1 to 2 gallons per minute, which is added to the process fluid. In many instances, this product dilution rate is unacceptable. The containment means of Figs. 7-9 solve these problems. In Figs. 7-9, parts having little or no modification from the form of Figs. 1 and 2 are given the same reference numerals.

The containment means of Fig. 7 includes an enclosure or shell shown generally at 214. The shell 214 includes a generally cylindrical body portion 216 which integrally connected to a gasket 218 at one end. The gasket 218 is trapped between the gland plate 18 and housing 14. It can replace the gasket 22 of Fig. 1. At the opposite end of the shell 214 is a lip seal 220 which is in sliding engagement with the rotating flange 38 of the sleeve. Together the shell 214, sleeve flange 38, mating ring 84, primary ring 76 and retainer 70 define a containment chamber 222. The lip seal 220 is angled so as to restrict leakage from the process fluid side of the seal into the containment chamber

222. Buffer fluid pumped across the seal faces into the containment chamber 222 is permitted to flow from the containment chamber, past the lip seal and into the high pressure side of the seal. The shell 214 is preferably made of polytetrafluoroethylene.

Fig. 8 shows an alternate embodiment wherein the containment means comprises a modified retainer 70A. The retainer has an extension portion 224 spanning the seal faces. Extension 224 has a seat which mounts a flexible seal 226. Seal 226 contacts the flange 38 of the rotating sleeve. A containment chamber is defined at 228.

A further alternate embodiment is shown in Fig. 9. It utilizes a generally cylindrical adaptor 230, which is press fit onto the retainer 70. A modified sleeve flange 38A has a threaded shoulder 232 on which a collar 234 is mounted. The collar 234 traps a lip seal 236 on the end of the flange 38A. The adaptor 230 and seal 236 define the containment chamber 238.

The containment means of any of the embodiments of Figs. 7-9 creates a containment chamber so that a clean buffer fluid surrounds the O-ring area of the secondary seal. The containment chamber has a small enough volume so that the seal pumping rate can fill the containment chamber without requiring external filling. The lip seals function as check valves which seal product pressure while allowing buffer fluid to get past. Once the pump or other device starts operating, the containment chamber will fill with pumpage from the seal faces. When the containment chamber pressure overcomes the process pressure and the lip seal resistance, the buffer fluid will bleed past the lip into the process. This keeps the product away from the O-ring secondary seal area and will continually lubricate the lip seal in dynamic operation. Statically, the lip seal will actuate with O.D. pressure to prevent product intrusion.

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

Claims

1. A seal system for sealing the annular space between a housing and a rotating shaft extending through a bore in the housing, the housing containing a high pressure fluid, the seal system comprising:
 - a gland plate connectable to the housing;
 - an outboard mechanical end face seal having a first portion connected to the gland plate and a second portion connectable for rotation with the shaft;
 - an inboard mechanical end face seal at

- least a portion of which is mounted in the annular space, the seal having a mating-ring and a primary ring, one of which is mounted for rotation with the shaft and the other of which is fixedly connected to the gland plate, the mating and primary rings having opposed, radial seal faces on one side thereof and back faces on the other side thereof, the seal and back faces of the primary ring being arranged such that the primary ring is double balanced, the inboard seal being positioned axially of the outboard seal to define a buffer chamber; and means carried by said inboard seal for pumping a fluid from said buffer chamber towards the inboard high pressure side of the inboard seal in opposition to fluid leakage from said high pressure side.
2. The seal system of claim 1 further comprising a sleeve mounted on the shaft for rotation therewith, the rotating portions of the inboard and outboard seals being fixedly connected to the sleeve.
 3. The seal system of claim 2 further comprising a retainer at least partially enclosing the primary ring to fix its radial position while allowing axial movement of the primary ring, and a secondary seal disposed between the retainer and the back face of the primary ring.
 4. The seal system of claim 3 further comprising a plurality of springs disposed between the retainer and primary ring, urging the primary ring toward the mating ring.
 5. The seal system of claim 1 further comprising a retainer at least partially enclosing the primary ring to fix its radial position while allowing axial movement of the primary ring.
 6. The seal system of claim 5 further comprising a first seat having an axial ledge and a radial shoulder formed in the back face of the primary ring, a second seat having an axial ledge and a radial shoulder formed in the retainer opposite said first seat, the diameters of said axial ledges being symmetric about the centerline of the primary ring seal face, and a secondary seal being disposed between said axial ledges.
 7. The seal system of claim 1 wherein the ring of the inboard seal which is closest to the buffer chamber has axial grooves formed therein to enhance buffer fluid flow from the back face of said ring to the seal face.
 8. The seal system of claim 1 further comprising a buffer fluid reservoir located externally of said housing at a greater elevation than the buffer chamber and in fluid communication with the buffer chamber so to supply buffer fluid thereto under gravity feed.
 9. The seal system of claim 8 further comprising a float valve attached to the reservoir and connectable to a fluid supply line for automatically refilling the reservoir when the buffer fluid therein reaches a predetermined level.
 10. The seal system of claim 8 wherein the fluid communication between the reservoir and buffer chamber includes an inlet line and an outlet line forming a thermal loop.
 11. The seal system of claim 10 wherein the inlet line is finned and the outlet line is insulated to enhance a thermal syphon effect.
 12. The seal system of claim 1 wherein the primary ring is made of tungsten carbide and the mating ring is made of silicon carbide.
 13. The seal system of claim 1 wherein one of the primary or mating ring seal faces has a circumferential dam on the high pressure side of said face, and a plurality of spiral grooves extending from the dam to the low pressure buffer chamber for pumping fluid from the buffer chamber into the high pressure side of the seal.
 14. The seal system of claim 13 wherein the primary ring face width is about .312 inches and the dam width is about .093 inches.
 15. A seal system for sealing the annular space between a housing and a rotating shaft extending through a bore in the housing, the housing containing a high pressure fluid, the seal system comprising:
 - a gland plate connectable to the housing;
 - an outboard mechanical end face seal having a first portion connected to the gland plate and a second portion connectable for rotation with the shaft;
 - an inboard mechanical end face seal at least a portion of which is mounted in the annular space, the seal having a mating ring and a primary ring, one of which is mounted for rotation with the shaft and the other of which is fixedly connected to the gland plate, the inboard seal being positioned axially of the outboard seal to define a buffer chamber;
 - means carried by said inboard seal for

- pumping a fluid from said buffer chamber towards the inboard high pressure side of the inboard seal in opposition to fluid leakage from said high pressure side; and
 a buffer fluid reservoir located externally of said housing at a greater elevation than the buffer chamber and in fluid communication with the buffer chamber so to supply buffer fluid thereto under gravity feed.
16. The seal system of claim 15 further comprising a float valve attached to the reservoir and connectable to a fluid supply line for automatically refilling the reservoir when the buffer fluid therein reaches a predetermined level.
17. The seal system of claim 15 wherein the fluid communication between the reservoir and buffer chamber includes an inlet line and an outlet line forming a thermal loop.
18. The seal system of claim 17 wherein the inlet line has cooling fins and the outlet line is insulated to enhance a thermal syphon effect.
19. A seal system for sealing the annular space between a housing and a rotating shaft extending through a bore in the housing, the housing containing a high pressure fluid, the seal system comprising:
 a gland plate connectable to the housing;
 an outboard mechanical end face seal having a first portion connected to the gland plate and a second portion connectable for rotation with the shaft;
 an inboard mechanical end face seal at least a portion of which is mounted in the annular space, the seal having a mating ring and a primary ring, one of which is mounted for rotation with the shaft and the other of which is fixedly connected to the gland plate, the inboard seal being positioned axially of the outboard seal to define a buffer chamber;
 means carried by said inboard seal for pumping a fluid from said buffer chamber towards the inboard high pressure side of the inboard seal in opposition to fluid leakage from said high pressure side;
 a buffer fluid reservoir located externally of said housing in fluid communication with the buffer chamber so to supply buffer fluid thereto;
 means for pressurizing said reservoir; and
 pumping means rotatably connected to the shaft in the buffer chamber for circulating buffer fluid between the reservoir and buffer chamber.
20. The seal system of claim 19 wherein the primary ring is made of tungsten carbide and the mating ring is made of silicon carbide.
21. The seal system of claim 20 wherein the mating and primary rings have opposed, radial seal faces on one side thereof and back faces on the other side thereof, the seal and back faces of the primary ring being arranged such that the primary ring is double balanced.
22. The seal system of claim 19 wherein the mating and primary rings have opposed, radial seal faces on one side thereof and back faces on the other side thereof, the seal and back faces of the primary ring being arranged such that the primary ring is double balanced.
23. The seal system of claim 19 wherein one of the primary or mating ring seal faces has a circumferential dam on the high pressure side of said face, and a plurality of spiral grooves extending from the dam to the low pressure buffer chamber for pumping fluid from the buffer chamber into the high pressure side of the seal.
24. The seal system of claim 23 wherein the primary ring face width is about .312 inches and the dam width is about .093 inches.
25. A seal system for sealing the annular space between a housing and a rotating shaft extending through a bore in the housing, the housing containing a fluid at a pressure of at least 300 psi, the seal system comprising:
 a gland plate connectable to the housing;
 an outboard mechanical end face seal having a first portion connected to the gland plate and a second portion connectable for rotation with the shaft;
 an inboard mechanical end face seal at least a portion of which is mounted in the annular space, the inboard seal being positioned axially of the outboard seal to define a buffer chamber, the inboard seal having a mating ring and a primary ring, one of which is mounted for rotation with the shaft and the other of which is fixedly connected to the gland plate, the primary and mating rings being made from materials having a high modulus of elasticity; and
 means carried by said inboard seal for pumping a fluid from said buffer chamber towards the inboard high pressure side of the inboard seal in opposition to fluid leakage from said high pressure side.

26. The seal system of claim 25 wherein the mating and primary rings have opposed, radial seal faces on one side thereof and back faces on the other side thereof, the seal and back faces of the primary ring being arranged such that the primary ring is double balanced. 5
27. The seal system of claim 25 wherein one of the primary and mating ring seal faces has a circumferential dam on the high pressure side of said face, and the other seal face has a plurality of spiral grooves extending from the dam to the low pressure buffer chamber for pumping fluid from the buffer chamber into the high pressure side of the seal. 10
28. The seal system of claim 27 wherein the primary ring face width is about .312 inches and the dam width is about .093 inches. 15
29. A seal system for sealing the annular space between a housing and a rotating shaft extending through a bore in the housing, the housing containing a high pressure fluid, the seal system comprising: 20
- a gland plate connectable to the housing;
 - a sleeve mounted on the shaft for rotation therewith;
 - a first static seal between the sleeve and shaft to prevent leakage between them;
 - an outboard mechanical end face seal having a first portion connected to the gland plate and a second portion connected to the sleeve for rotation with the shaft;
 - an inboard mechanical end face seal at least a portion of which is mounted in the annular space, the inboard seal having, 25
 - a primary ring;
 - a retainer fixed to the gland plate and at least partially enclosing the primary ring to fix its radial position while allowing axial movement of the primary ring;
 - a secondary seal disposed between the retainer and the primary ring;
 - a mating ring surrounding the sleeve and connected thereto for rotation with the shaft;
 - a static seal between the sleeve and the mating ring for preventing leakage therebetween;
 - the mating and primary rings having opposed, relatively rotating, radial seal faces on one side thereof, the outer diameters of said seal faces being exposed to the fluid pressure in the housing;
 - the inboard seal being positioned axially of the outboard seal to define a buffer chamber; and 30
 - means carried by said inboard seal for pumping a fluid from said buffer chamber towards the inboard high pressure side of the inboard seal in opposition to fluid leakage from said high pressure side.
30. The seal system of claim 29 further wherein the primary and mating rings have back faces on the sides opposite the seal faces, the seal and back faces of the primary ring being arranged such that the primary ring is double balanced. 35
31. The seal system of claim 30 further comprising a first seat having an axial ledge and a radial shoulder formed in the back face of the primary ring, a second seat having an axial ledge and a radial shoulder formed in the retainer opposite said first seat, the diameters of said axial ledges being symmetric about the centerline of the primary ring seal face, the secondary seal being disposed between said axial ledges. 40
32. The seal system of claim 29 wherein the primary ring has axial grooves formed therein to enhance buffer fluid flow from a back side of the primary ring to the seal face. 45
33. The seal system of claim 29 further comprising a buffer fluid reservoir located externally of said housing at a greater elevation than the buffer chamber and in fluid communication with the buffer chamber so to supply buffer fluid thereto under gravity feed. 50
34. The seal system of claim 33 further comprising a float valve attached to the reservoir and connectable to a fluid supply line for automatically refilling the reservoir when the buffer fluid therein reaches a predetermined level. 55
35. The seal system of claim 33 wherein the fluid communication between the reservoir and buffer chamber includes an inlet line and an outlet line forming a thermal loop.
36. The seal system of claim 35 wherein the inlet line has cooling fins and the outlet line is insulated to enhance a thermal syphon effect.
37. The seal system of claim 29 wherein the primary ring is made of tungsten carbide and the mating ring is made of silicon carbide.
38. The seal system of claim 29 wherein one of the primary or mating ring seal faces has a circumferential dam on the high pressure side of said face, and a plurality of spiral grooves

extending from the dam to the low pressure buffer chamber for pumping fluid from the buffer chamber into the high pressure side of the seal.

39. The seal system of claim 38 wherein the primary ring face width is about .312 inches and the dam width is about .093 inches.

40. A seal system for sealing the annular space between a housing and a rotating shaft extending through a bore in the housing, the housing containing a high pressure fluid, the seal system comprising:

a gland plate connectable to the housing;
an outboard mechanical end face seal having a first portion connected to the gland plate and a second portion connectable for rotation with the shaft;

an inboard mechanical end face seal at least a portion of which is mounted in the annular space, the seal having a mating ring and a primary ring, first mounting for connecting one of said rings for rotation with the shaft and second mounting means for connecting the other ring to the gland plate in non-rotating relation, the mating and primary rings having opposed, radial seal faces on one side thereof, the inboard seal being positioned axially of the outboard seal to define a buffer chamber;

containment means having one end slidably engaged with at least one of said one ring or the first mounting means, and the other end of the containment means being engaged with at least one of said other ring or the second mounting means, the containment means spanning the seal faces and being spaced therefrom on the high pressure side of the inboard seal to define a containment chamber; and

means carried by the inboard seal for pumping a fluid from the buffer chamber towards the containment chamber in opposition to fluid leakage from the high pressure side of the housing.

41. The seal system of claim 40 wherein the containment means includes a lip seal at the intersection of said means with said one ring or the first mounting means.

42. The seal system of claim 40 wherein the second mounting means includes a retainer at least partially enclosing the primary ring to fix its radial position while allowing axial movement of the primary ring.

43. The seal system of claim 42 wherein the con-

tainment means comprises a shell having a gasket at one end disposed between the gland plate and housing, and a lip seal at the other end engaged with said one ring or the first mounting means.

44. The seal system of claim 43 wherein the lip seal is angled so as to restrict leakage from the high pressure side of the seal into the containment chamber while permitting flow of buffer fluid from the containment chamber into the high pressure side.

45. The seal system of claim 42 wherein the containment means comprises a polytetrafluoroethylene cylinder attached to the retainer and having a lip in contact with the rotating ring.

46. The seal system of claim 40 wherein the second mounting means comprises a retainer at least partially enclosing the primary ring to fix its radial position while allowing axial movement of the primary ring, and the containment means includes an extension portion connected to the retainer, spanning the seal faces and mounting a flexible seal which is engageable with said one ring or the first mounting means.

47. The seal system of claim 42 wherein the containment means comprises a generally cylindrical adaptor press fit on the retainer, and a lip seal carried by one of said one ring or the first mounting means, the lip seal being slidably engageable with the adaptor.

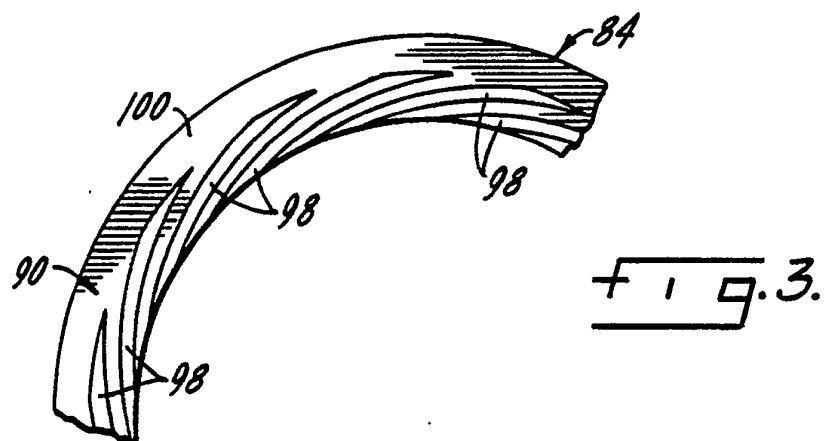
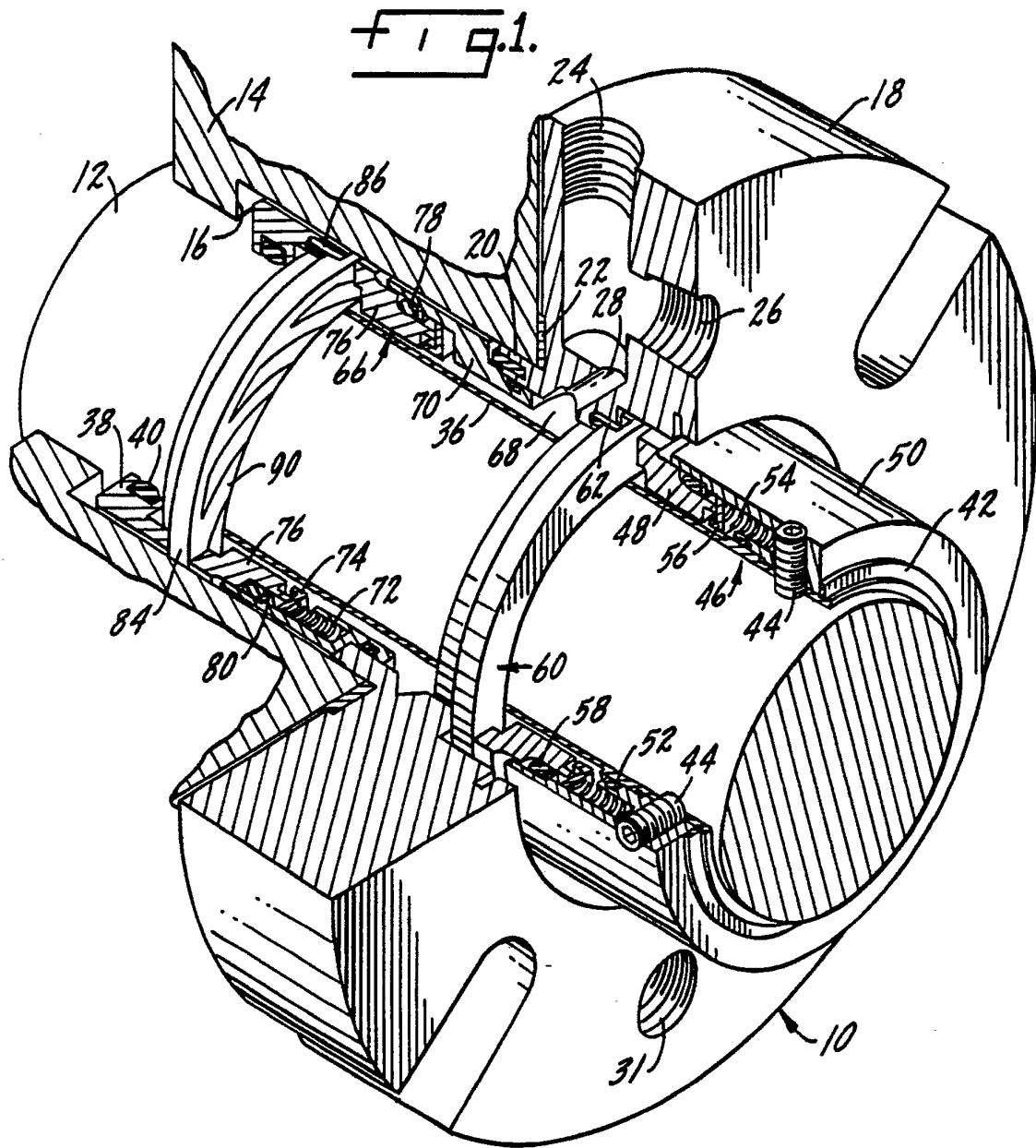
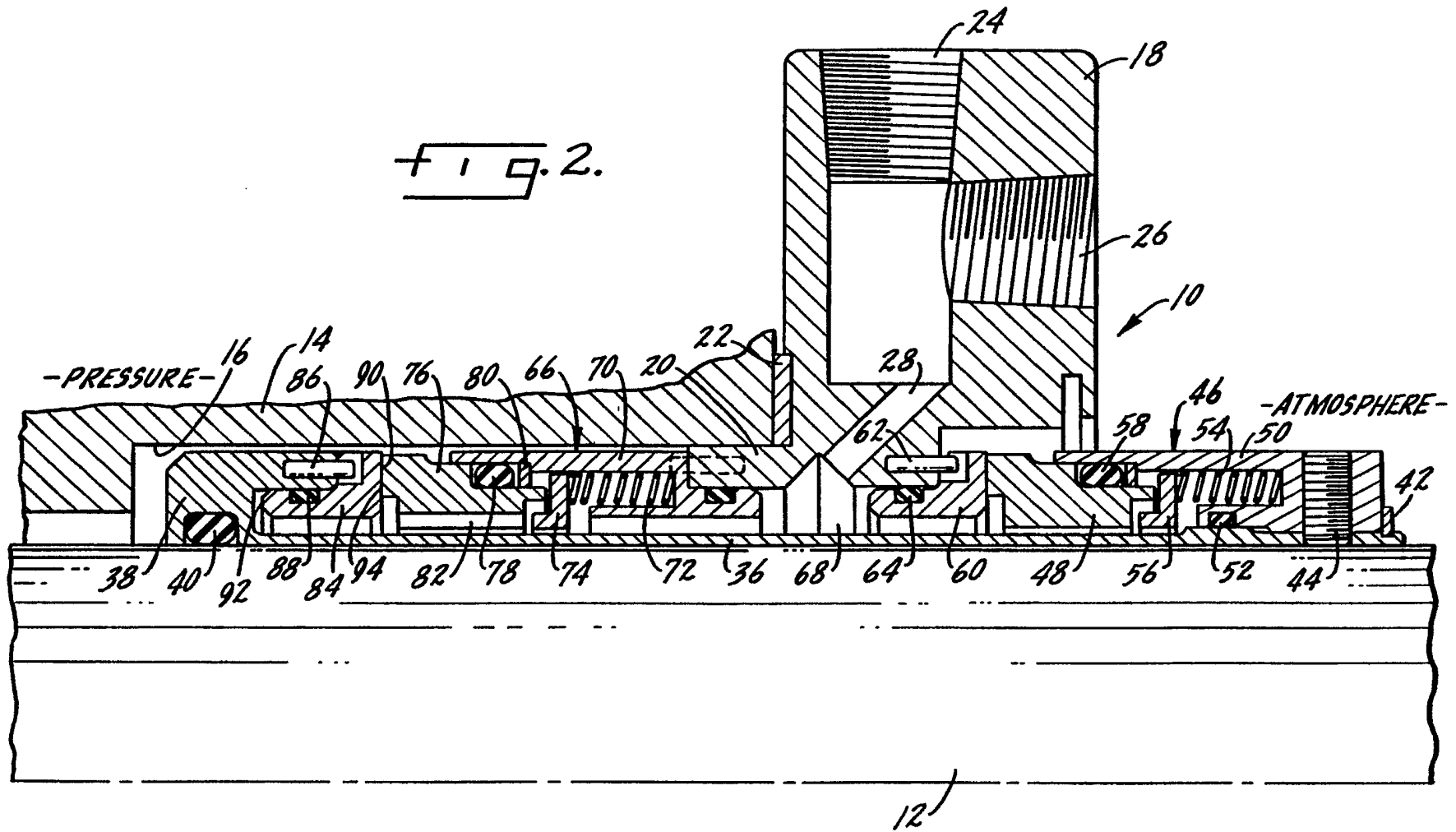
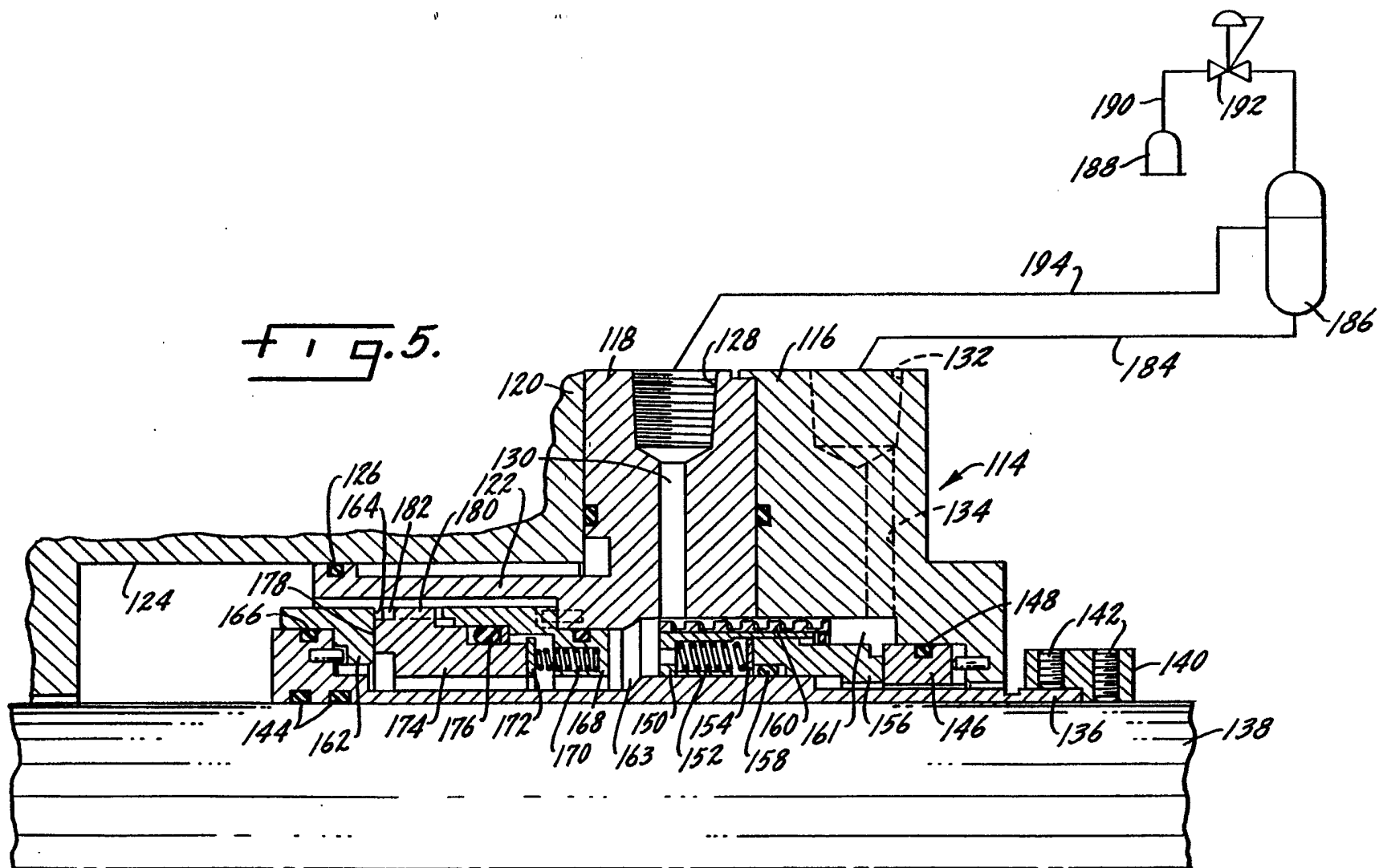
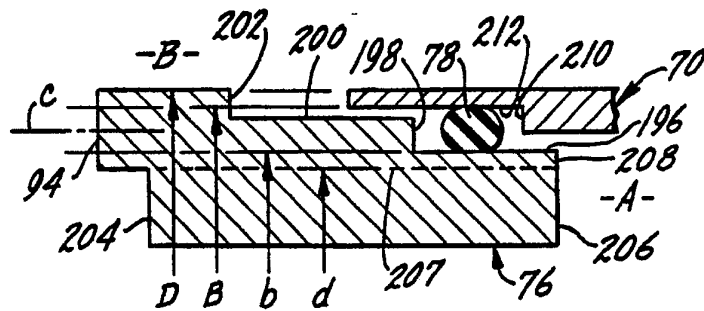
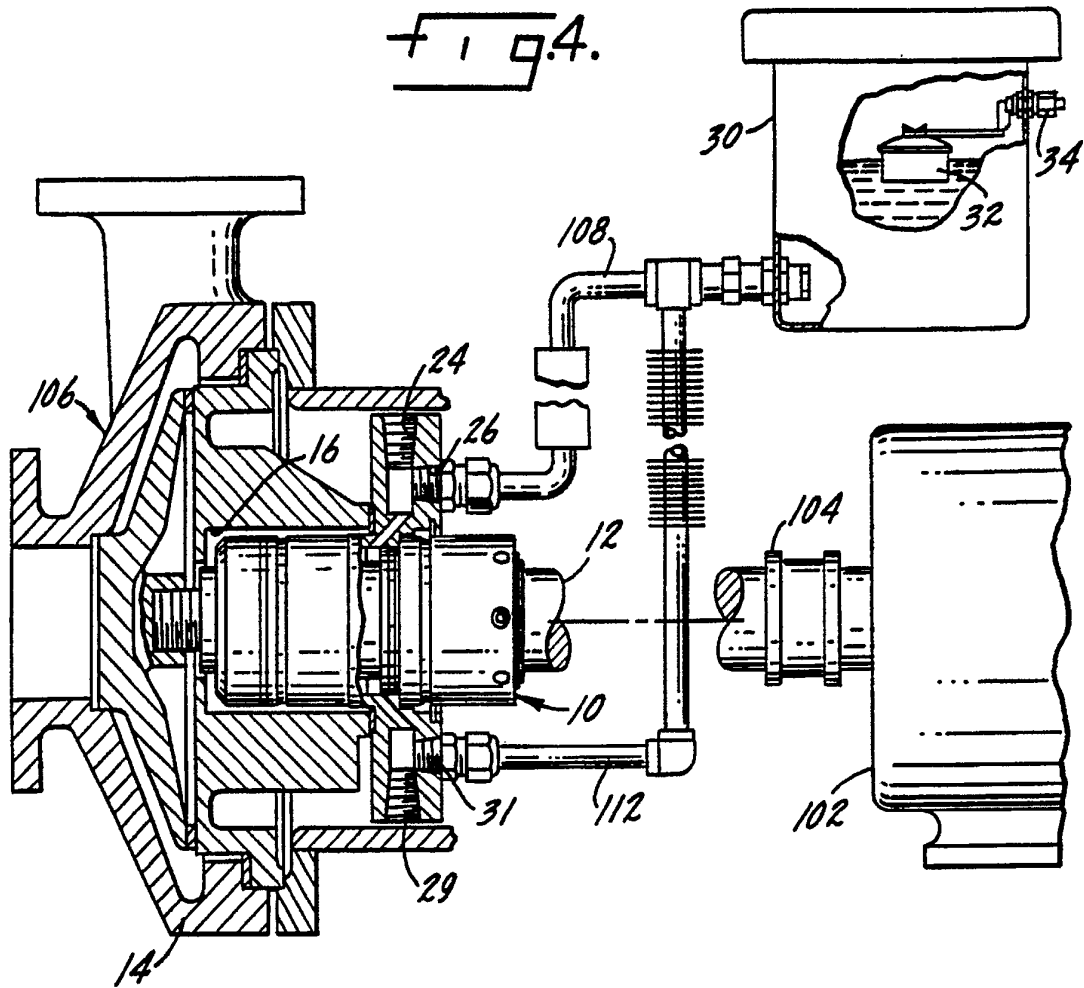


Fig. 2.







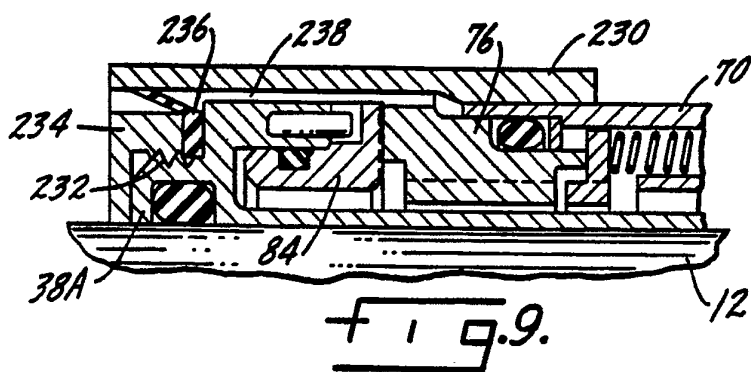
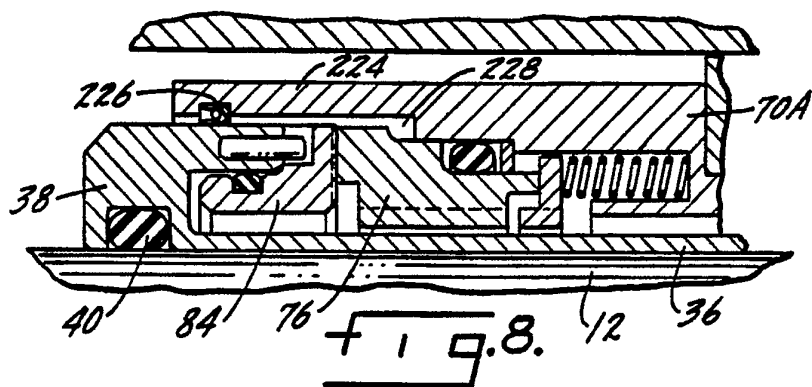
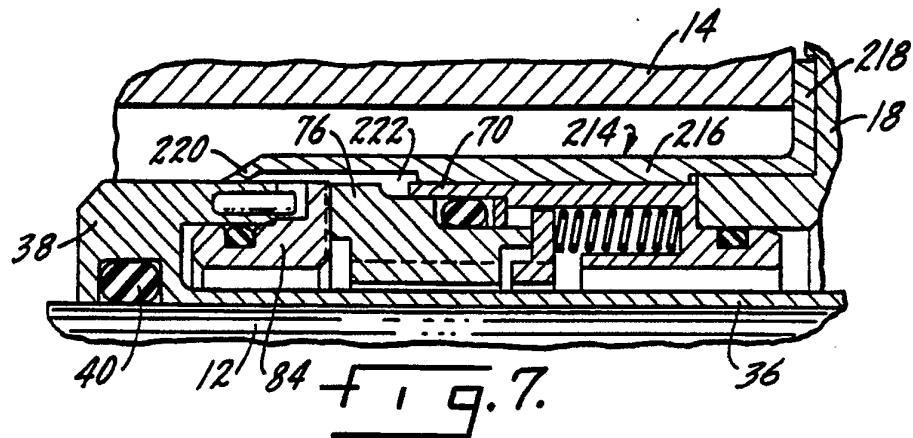
$$BALANCE = \frac{AREA_{CLOSING}}{AREA_{OPENING}}$$

$$A_o = \pi(D^2 - d^2)/4$$

$$I.D. PRESSURE A_c = \pi(B^2 - d^2)/4$$

$$O.D. PRESSURE A_c = \pi(D^2 - b^2)/4$$

Fig. 6.



[54] **MECHANICAL SEAL WITH HEAT EXCHANGER**

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[52] U.S. Cl. **277/9.5; 165/179; 277/22; 277/81 R**

[58] Field of Search **277/22, 32, 81 R, 85, 277/87, 93 SD, 9.5; 165/179**

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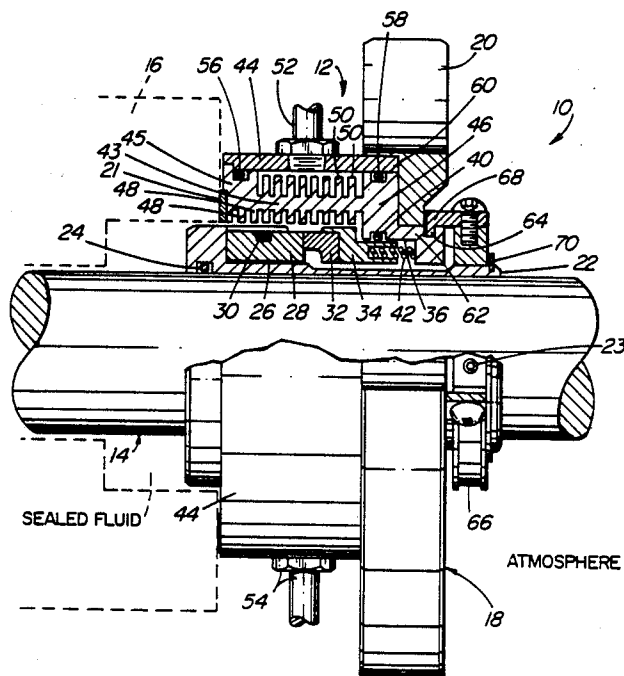
Seven page presentation entitled "Advanstech Trans-therm Cartridge Mechanical Seal" (undated).

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Attorney, Agent, or Firm—Howson & Howson

[57] **ABSTRACT**

The present invention is particularly concerned with rotary end face seals of the type including a pair of annular sealing members disposed in relatively rotatable sealing proximity about a rotation axis. Each of these annular members typically includes a radially arranged annular sealing surface with these sealing surfaces being in face-to-face sealing proximity with one another during use so as to define a radially arranged annular sealing interface which separates a region of higher pressure from a region of lower fluid pressure. The invention particularly provides an annular heat exchanger which, in use, surrounds the annular sealing members in radially spaced relation thereto to define, within one of the above-noted regions, an annular zone of fluid pressure between the heat exchanger and the sealing members. The heat exchanger acts to transmit heat energy into or out of the above-noted annular zone thereby to affect the temperature of the fluid in such zone during use and hence affect the operating temperatures of the sealing members and the interface temperature to achieve greater sealing efficiency and/or longer seal life.

13 Claims, 2 Drawing Sheets



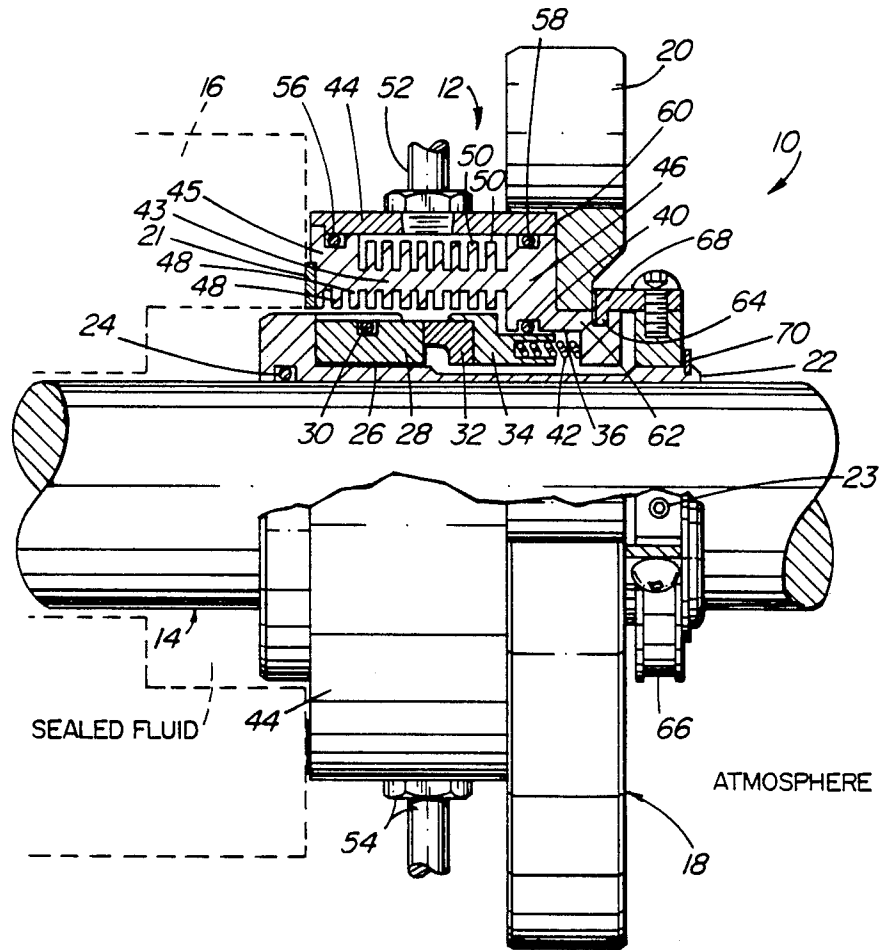


FIG. 1

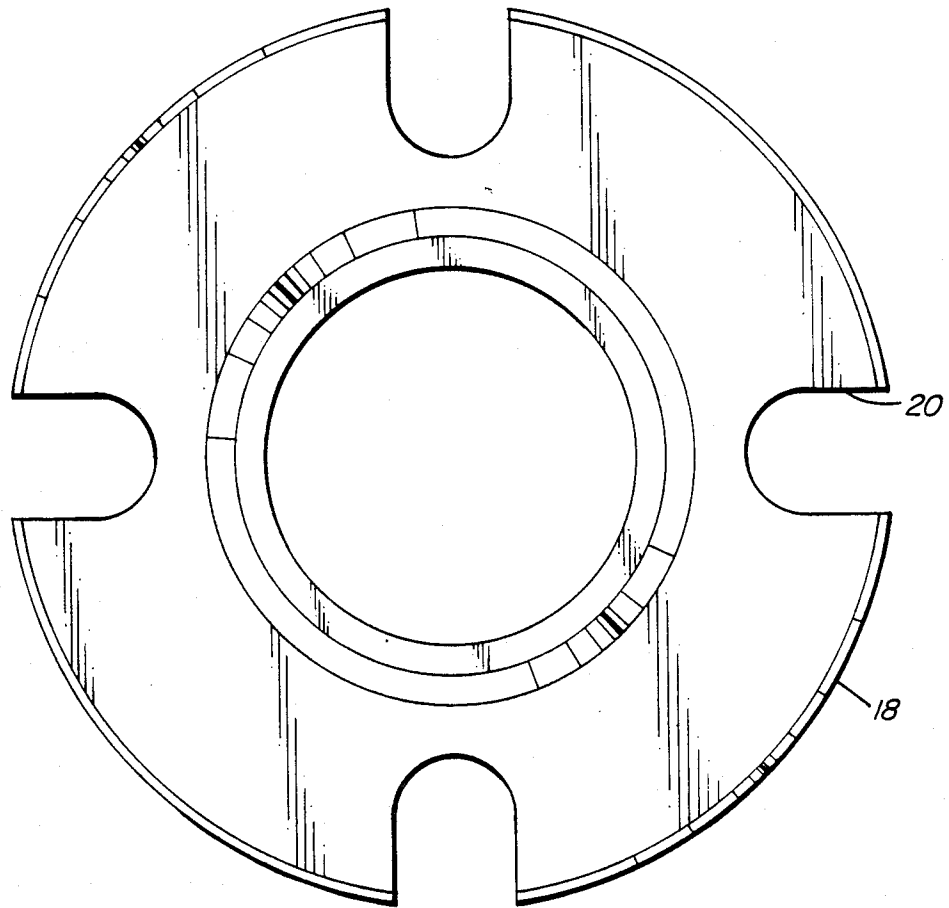


FIG. 2

MECHANICAL SEAL WITH HEAT EXCHANGER

BACKGROUND OF THE INVENTION

This invention relates to mechanical seals and in particular to heat exchangers for use in combination with such seals.

In general, it is well known to modify the temperature of certain components of a mechanical seal in order to provide a more efficient operation of the seal and/or longer seal life. Mechanical seals generate heat at the seal faces. It is also well known that the stability of the fluid film between the seal faces is critical for the satisfactory operation of the seal. The prevailing fluid film temperatures depend on the heat generated, the latter being a function of sealed pressure, shaft speed and coefficient of friction, and the heat dissipation properties of the seal including the heat transfer properties of the sealed fluid. Under certain conditions it may be necessary to introduce cooling to reduce the temperatures existing at the sealed faces. In other situations, as in the sealing of very heavy and viscous materials, it may be necessary to introduce a certain amount of heating thereby to achieve satisfactory conditions at the seal faces.

Many such thermosensitive fluids can have detrimental effects on other seal components, if temperatures are not closely controlled. Also, temperature limitations of secondary sealing components (eg. O-rings) can be exceeded resulting in premature failure.

Several traditional methods have been utilized in the past to modify the temperature of the seal chamber and/or mechanical seal components in an effort to ensure successful operation of the seal.

The first of these traditional methods involves the use of a jacket on the equipment's stuffing box to enable the circulation of a suitable heat exchange medium. These arrangements have a number of disadvantages. When considering the case of cooling, one must cool almost the whole casing, i.e. heat must be drawn away from the whole area. This requires a relatively large volume of coolant, and is consequently not very cost-effective or efficient.

Another method involves recirculation of the fluid being sealed through an external heat exchanger and back into the seal chamber. For example, fluid from the discharge side of a pump is used and this higher pressure fluid is injected into the seal chamber to prevent flashing at the seal faces. The combined effect of cooling the pumpage and increasing the sealed pressure often serves to prevent flashing in the seal interface. However, the increased pressure acts to increase the seal face closing force and this in turn increases the heat generated at the seal faces. Hence, the system has a built-in measure of inefficiency. Moreover, abrasive materials in the pumpage fluid may be forced into the sealing area. Also, an external heat exchanger is both expensive and relatively inefficient since it too often requires large volumes of heat exchange medium.

Another method involves the use of buffer fluid systems between double seals or quench and drain systems on the atmospheric side of a single seal. In the case of a buffer fluid, a heat exchange medium, separate from the fluid being pumped, is directed into the area between the two seals. The means for pumping or supplying this barrier fluid gives rise to additional expense. Even with double seal arrangements and a barrier fluid, the hot pumpage at the seal interface can give rise to flashing

and dry seal faces. If the barrier fluid pressure is made sufficiently high as to avoid this problem, then relatively costly pumping equipment is needed. Another drawback is that one cannot readily detect inboard seal leakage with this system. Also, it too is relatively inefficient, requiring large volumes of heat exchange media either through the seal to the drain (in a "once-through" system) or through a coil in the convection tank (in a recirculating or "thermosiphon" closed-loop system).

Another arrangement involves the use of a jacketed gland or a jacketed stationary seat designed to permit circulation of a heat transfer medium. Again, difficulties are encountered in that a relatively large gland plate must be heated or cooled before any cooling effect at the seal faces can be achieved. Furthermore, the design is such that there is a very lengthy heat transfer path from the jacketed gland to the seal interface. It is solely by means of conductive heat transfer through the stationary seal face and holder and/or gland plate that any cooling is effected. Since it has been shown that the greater percentage of heat dissipation (80 to 90 percent) naturally occurs through the rotating face, this points to another shortcoming of designs of this type. The result is that very large flow rates for the heat transfer media are required to effect any reasonable change in temperature at the seal faces. This technique also requires a fairly special design which cannot be applied easily to every situation.

Another variation involves simple recirculation of pumpage but without the use of an auxiliary heat exchanger. Thus, no cooling is effected; rather, at best, some percentage of the heat generated at the seal interface is dissipated. This technique again has most of the disadvantages associated with the second technique noted above.

It can be seen from the above that there is therefore a need for a very simple yet effective means for effecting heating or cooling at the seal faces and which means is applicable to a wide variety of seal designs.

SUMMARY OF THE INVENTION

The present invention is particularly concerned with rotary end face seals of the type including a pair of annular sealing members disposed in relatively rotatable sealing proximity about a rotation axis. Each of these annular members typically includes a radially arranged annular sealing surface with these sealing surfaces being in face-to-face sealing proximity with one another during use so as to define a radially arranged annular sealing interface which separates a region of higher pressure from a region of lower fluid pressure. The invention particularly provides an annular heat exchanger which, in use, surrounds the annular sealing members in radially spaced relation thereto to define, within one of the above-noted regions, an annular zone of fluid pressure between the heat exchanger and the sealing members. The heat exchanger acts to transmit heat energy into or out of the above-noted annular zone thereby to affect the temperature of the fluid in such zone during use and hence affect the operating temperatures of the sealing members and the interface temperature to achieve greater sealing efficiency and/or longer seal life.

Preferably, the heat exchanger extends in the axial direction to axial positions spaced apart on opposing sides of the axial position of the sealing interface so that at least substantial parts of the axial extent of both of the

annular sealing members are exposed to the annular zone of fluid pressure. Stated differently, the annular heat exchanger is of sufficient axial extent as to embrace at least a major part of both of the annular sealing members.

Preferably, the radial dimension of the annular zone between the heat exchanger and the sealing members is small in relation to the outside diameters of the sealing members thereby to minimize or inhibit movement or flow of fluid into and out of the annular zone.

The heat exchanger preferably comprises a hollow annular body having a fluid inlet and a fluid outlet for passage of a heat transfer medium therethrough.

The hollow body of the heat exchanger typically comprises inner and outer radially spaced apart cylindrical walls joined by opposed end walls. The inner wall has annular ribs extending therearound on both of its opposing major surfaces to provide enhanced heat transfer while at the same time assisting in inhibiting the flow of fluid into and out of the annular zone by virtue of the labyrinth effect provided by the annular ribs. Such ribs also increase the strength of this component which is of importance especially when containing fluids at relatively high pressures.

In a typical embodiment, the heat exchanger may include a suitable flange for mounting and/or centering same on a gland plate of the seal. In certain designs, the heat exchanger may be provided with an annular recess for receiving or mounting one of the annular sealing members.

BRIEF DESCRIPTION OF THE DRAWING

FIG. 1 is a side elevation view, partly in section, of a shaft mounted rotary end face seal incorporating a heat exchanger in accordance with an embodiment of the present invention.

FIG. 2 is an end view of a gland plate per se

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENT

With reference now to the drawing there is shown a rotary end face seal assembly 10 which includes a heat exchanger 12 in accordance with the teachings of the present invention. The seal 10 is illustrated as being mounted on a shaft 14, the left hand end of shaft 14, as seen in the drawing, extending into the interior of a machine housing 16. In order to secure the seal assembly including the heat exchanger 12 to the housing 16, a circular gland plate 18 is provided, the same being provided with a plurality of bolt slots, extending inwardly from the periphery of gland plate 18, as shown in FIG. 2, wherein one of the bolt slots is indicated at 20. Bolts (not shown) may be passed through these bolt slots thereby to firmly secure the entire assembly to the housing 16. An annular gasket 21 interposed between heat exchanger 12 and housing 16 prevents leakage of the contained fluids.

The seal assembly includes a pair of annular seal rings disposed in relatively rotatable sealing proximity about the axis of the shaft 14. The rotary seal ring assembly includes an elongated seal sleeve 22 which extends around the shaft 14 and which, in use, is locked for rotation therewith by means of a plurality of set screws 23 positioned adjacent the outer end of the sleeve. In order to prevent leakage of fluid between the sleeve 22 and shaft 14, an O-ring seal 24 is provided located in an annular recess adjacent the inner end of sleeve 22. This inner end of sleeve 22 is also provided with an enlarged

annular pocket or recess 26. This annular recess 26 receives a rotating seal ring 28, such seal ring 28 being preferably made from a very hard, long wearing and thermally conductive material such as silicon-carbide. A further O-ring 30 located in a groove within the seal ring prevents leakage of fluid between the seal ring 28 and the sleeve 22.

The opposing stationary seal ring assembly comprises a seal ring 32 which is made of a softer material such as carbon graphite. This seal ring is mounted in known fashion in an annular metal seal ring holder 34. Seal ring holder 34 in turn is disposed within an annular recess 36 which is defined in an end portion of the heat exchanger 12 to be hereinafter described. A further O-ring 40 within recess 36 engages a portion of the seal ring holder 34 to prevent leakage of fluids thereby. The seal ring holder is provided with a number of small axially extending holes each of which receives a coil compression spring 42, these coil compression springs serving to urge the seal ring holder 34 and attached seal ring 32 against the face of the rotary seal ring 28. A plurality of axial pins, not shown, co-operate in known manner between recess 36 and seal ring holder 34 to prevent rotation of the latter while allowing limited movement thereof in the axial direction.

The rotary seal ring 28 and the stationary seal ring 32 are in face-to-face sealing proximity during use along a radially arranged annular sealing interface. They serve to separate a region of higher fluid pressure from a region of lower fluid pressure. The region of lower fluid pressure, i.e. atmospheric pressure, is disposed radially inwardly of the annular sealing interface while the zone of higher fluid pressure is disposed outwardly thereof.

The above-noted heat exchanger 12 is of annular configuration and it completely surrounds the rotary seal ring 28 as well as the stationary seal ring 32 and its holder. The heat exchanger is in radially spaced relation thereto such that an annular zone of fluid pressure is defined between the heat exchanger 12 and the sealing rings 28, 32. The function of the heat exchanger 12 is to transmit heat energy into or out of this annular zone thereby to affect the temperature of the fluid in such zone during use and thus affect the operating temperatures at the interface of the seal rings 28 and 32 during use.

It will be noted that the heat exchanger 12 extends in the axial direction to axial positions which are spaced well apart on opposing sides of the axial position of the sealing interface defined between seal rings 28 and 32 so that both of the seal rings 28 and 32 are exposed to the above-noted annular zone which exists between their outer surfaces and the interior surface of the heat exchanger 12.

As also shown in the drawing, the radial dimension of the annular zone between the heat exchanger 12 and the sealing rings 28, 32 and their holders is relatively small in relation to the outside diameters of the sealing rings thus minimizing movement or flow of the fluid being sealed into and out of such annular zone. In a typical embodiment wherein the outside diameter of the seal rings is in the order of two inches, the radial gap between the seal ring holders and the inner surfaces of the heat exchanger is only about 25 thousandths of an inch. Although this dimension is not particularly critical, it should be kept as small as is practical thereby to reduce ingress of fluid and thus minimize the volume of fluid to be heated or cooled within this annular zone.

The heat exchanger 12, as clearly shown, is in the form of a hollow body which includes inner and outer radially spaced cylindrical walls 43 and 44 joined by end wall portions 45 and 46. The inner wall 43 is provided with a series of annular ribs 48 and 50 extending therearound on both of its opposing major surfaces to provide enhanced heat transfer. At the same time, the inner annular ribs 48 assist in inhibiting flow of fluid into and out of the above-noted annular zone by virtue of the labyrinth effect provided by such ribs. It is also believed that this labyrinth effect reduces the effect of pressure surges in the contained fluid on seal face pressures. Also, such ribs serve to strengthen this component eliminating the danger of deformation when containing relatively high pressures. The heat exchanger includes an inlet 52 for admitting the heat exchange medium into the interior of the hollow body and an outlet 54 for releasing such heat transfer medium. In the usual case where cooling of the seal rings is desired, the heat exchange medium is commonly cool water supplied at the usual line pressures in the order of 50 to 80 pounds per square inch via, for example, $\frac{1}{4}$ inch diameter inlet and outlet pipes.

The heat exchanger outer cylindrical wall 44 is sealingly engaged with the remaining portions of the heat exchanger by way of O-ring seals 56, 58. This outer wall 44 may readily be slid endwise outwardly off the remainder of the heat exchanger body in the axial direction after disassembly thereby to provide for inspection and cleaning of the interior of the heat exchanger.

The spacing and depth of fins 48 and 50 is not critical but they should be selected so as to provide the maximum heat transfer capability. To further enhance heat transfer capability, the heat exchanger should be made of a material having a relatively high heat transfer coefficient. One such material is bronze although in certain cases where corrosion may be a problem, it may be necessary to use stainless steel or other alloys.

In the embodiment shown, the heat exchanger 12 has its outer end portion disposed and centered by and within an annular recess 60 disposed in the gland plate 10. An axially extending flange portion 62 of the heat exchanger projects axially outwardly of the front end of the assembly and this is provided with an annular groove 64. A locking ring 66 extends around the outer end of the shaft sleeve 22 and a plurality of centering clips 68 are secured by screws to this ring with such clips engaged in the annular groove 64. A snap ring 70 located in the outer end of the shaft sleeve 22 prevents relative axial movement between shaft sleeve 22 and the remainder of the assembly. This assembly temporarily serves to set the axial operating distance and to keep the springs 42 in compression and the seal ring faces in close contacting properly centered relationship during mounting and assembly of the seal assembly on a piece of equipment. After the seal together with its heat exchanger have been installed, and the sleeve 22 locked to the shaft 14 by means of set screws 23, the clips 68 are removed.

By virtue of the use of the heat exchange apparatus described above, seal ring life can be greatly extended since by varying the flow volume and/or the temperature of the heat transfer medium passing through the heat exchanger, the temperatures at the sealing ring interface can be brought toward an optimal condition for the particular service conditions concerned. The principles of the invention are applicable to a wide variety of seal designs, i.e. the particular seal arrange-

ment described above is only one of many with which the heat exchanger can be utilized. By utilizing the principles of the present invention, many of the disadvantages associated with previous methods of cooling or heating seals can be greatly alleviated or eliminated altogether.

I claim:

1. A mechanical seal assembly comprising:

- (a) a pair of annular sealing members disposed in relatively rotatable sealing proximity about a rotation axis;
 - (b) each of said members defining a generally radially arranged annular sealing surface, said annular sealing surfaces being in face-to-face sealing proximity with one another during use thereby defining a generally radially arranged annular sealing interface separating a region of higher fluid pressure from a region of lower fluid pressure while allowing relative rotation between said sealing surfaces;
 - (c) and an annular heat exchanger surrounding said sealing members in radially spaced relation thereto to define within one of said regions an elongated annular zone located between said heat exchanger and said sealing members with said annular zone, during use, being filled with fluid from one of said regions of fluid pressure, said heat exchanger being capable of acting to transmit heat energy into or out of said annular zone thereby to affect the temperature of the fluid in said annular zone during use and hence the operating temperatures of said sealing members and the temperature at said sealing interface.
 - (d) said heat exchanger and said elongated annular zone defined between the heat exchanger and the sealing members extending in the axial direction to axial positions spaced apart sufficiently on opposing sides of the axial position of the sealing interface so that at least substantial parts of the axial extent of both of said annular sealing members are within the axial length of said heat exchanger and in direct heat transfer relation when in use with the fluid within said elongated annular zone, and,
 - (e) the radial dimension of the annular zone between said heat exchanger and the sealing members being small in relation to the outside diameters of the sealing members such as to restrict movement or flow of the fluid into and out of said annular zone and thus minimize the volume of fluid to be heated or cooled within said annular zone and wherein
 - (f) said heat exchanger comprises an annular hollow body having a fluid inlet and a fluid outlet for passage of a heat transfer medium therethrough, said hollow body comprising inner and outer radially spaced cylindrical walls joined by end walls, said inner wall having a pair of opposed major surfaces and said inner wall having ribs extending therearound on at least that one of its opposing major surfaces which is exposed to said annular zone to provide enhanced heat transfer with fluid in said annular zone while further assisting in inhibiting the flow of fluid into and out of said annular zone by virtue of the labyrinth effect provided by said ribs exposed within said annular zone.
2. A mechanical seal assembly comprising:
- (a) a pair of annular sealing members disposed in relatively rotatable sealing proximity about a rotation axis;

- (b) each of said members defining a generally radially arranged annular sealing surface, said annular sealing surfaces being in face-to-face sealing proximity with one another during use thereby defining a generally radially arranged annular sealing interface separating a region of higher fluid pressure from a region of lower fluid pressure while allowing relative rotation between said sealing surfaces;
- (c) and an annular heat exchanger surrounding said sealing members in radially spaced relation thereto to define within one of said regions an elongated annular zone located between said heat exchanger and said sealing members with said annular zone, during use, being filled with fluid from one of said regions of fluid pressure, said heat exchanger being capable of acting to transmit heat energy into or out of said annular zone thereby to affect the temperature of the fluid in said annular zone during use and hence the operating temperatures of said sealing members and the temperature at said sealing interface,
- (d) said heat exchanger and said elongated annular zone defined between the heat exchanger and the sealing members extending in the axial direction to axial positions spaced apart sufficiently on opposing sides of the axial position of the sealing interface so that at least substantial parts of the axial extent of both of said annular sealing members are within the axial length of said heat exchanger and in direct heat transfer relation when in use with the fluid within said elongated annular zone, and,
- (e) the radial dimension of the annular zone between said heat exchanger and the sealing members being small in relation to the outside diameters of the sealing members such as to restrict movement or flow of the fluid into and out of said annular zone and thus minimize the volume of fluid to be heated or cooled within said annular zone wherein
- (f) said heat exchanger comprises an annular hollow body adapted for passage of a heat transfer medium therethrough, said mechanical seal assembly including a gland plate for mounting the mechanical seal assembly to a machine casing, and flange means on said heat exchanger for centering and/or mounting the latter to the gland plate.
3. The seal according to claim 2 the heat exchanger further including an annular recess therein for receiving/mounting one of said annular sealing members.
4. The seal according to claim 2 further including temporary securing clips engaged with said flange means to secure the seal assembly together and to center the mechanical seal assembly relative to said rotation axis.
5. A heat exchanger for a seal assembly comprising: a hollow annular body adapted to be mounted to the seal assembly in surrounding, radially spaced relation to a pair of annular sealing members of the seal assembly to transmit heat energy toward or away from them, fluid inlet and outlet means on said body for passage of heat transfer medium therethrough; said annular body having inner and outer radially spaced cylindrical walls joined by end walls, said radially inner wall having a pair of opposing major surfaces and radially disposed ribs extending therearound on both of said opposing major surfaces to provide enhanced heat transfer while assisting in inhibiting the flow of fluid into and out of an annular zone which exists, in use, between said inner wall of the heat exchanger and the outer surfaces of the annular sealing members.

6. The heat exchanger of claim 5 including flange means for mounting the heat exchanger to a gland plate or mounting plate.
7. The heat exchanger of claim 5 including means for seating and/or sealingly engaging one of the annular seal rings.
8. A heat exchanger for a seal assembly comprising: a hollow annular body adapted to be mounted to the seal assembly in surrounding, radially spaced relation to a pair of annular sealing members of the seal assembly to transmit heat energy toward or away from them, fluid inlet and outlet means on said body for passage of heat transfer medium therethrough; said annular body having inner and outer radially spaced cylindrical walls joined by end walls, said radially inner wall having a radially innermost major surface and a radially outmost major surface with ribs extending on at least the radially innermost major surface to provide enhanced heat transfer while assisting in inhibiting the flow of fluid into and out of an annular zone which exists, in use, between said radially inner wall of the heat exchanger and the radially outer surfaces of the annular sealing members.
9. The heat exchanger of claim 8 including flange means for mounting same to a gland plate or mounting plate.
10. The heat exchanger of claim 8 including means for seating and/or sealingly engaging one of the annular seal rings.
11. A mechanical seal assembly including a pair of relatively rotatable annular sealing members disposed in sealing proximity to one another around an axis to separate regions of differing fluid pressure, one of said sealing members adapted to be fixed to a shaft for rotation therewith and the other sealing member adapted to remain stationary, and an annular heat exchanger surrounding both said annular sealing members in close radially spaced proximity thereto so as to transfer heat energy into or out of said annular sealing members to heat or cool the same when in use, the annular heat exchanger being of sufficient axial extent as to substantially encompass both of said annular sealing members, the radial spacing between the heat exchanger and the sealing members being small in relation to the outside diameters of the sealing members so as to reduce circulation of fluid from one of said regions of fluid through the radial space between the heat exchanger and said sealing members, said heat exchanger being a hollow body having inlet and outlet means for circulation of heat transfer medium therethrough, said hollow body comprising inner and outer radially spaced cylindrical walls joined by end walls, said inner wall having a pair of opposed major surfaces and said inner wall having radially extending ribs extending therearound on at least that one of its opposing major surfaces which is exposed to said radial space to provide enhanced heat transfer with fluid in said radial space during use while assisting in inhibiting the flow of fluid into and out of said radial space by virtue of the labyrinth effect provided by said ribs which are exposed within said radial space.
12. The mechanical seal assembly of claim 11 including a mounting plate for attaching the mechanical seal assembly to a machine housing, said annular heat exchanger adapted to be clamped between the mounting plate and the machine housing.
13. The mechanical seal assembly of claim 11 wherein said heat exchanger includes a recess therein for receiving the stationary seal ring.

* * * * *

UNITED STATES PATENT AND TRADEMARK OFFICE
CERTIFICATE OF CORRECTION

PATENT NO. : 4,872,689
DATED : October 10, 1989
INVENTOR(S) : Kevin R. Drumm

It is certified that error appears in the above-identified patent and that said Letters Patent is hereby corrected as shown below:

At column 1, line 51, change "sea" to --seal--.

At column 6, line 33, change "interface." to --interface,--.

Signed and Sealed this
Twenty-fifth Day of February, 1992

Attest:

Attesting Officer

HARRY F. MANBECK, JR.

Commissioner of Patents and Trademarks