

Dec. 25, 1962

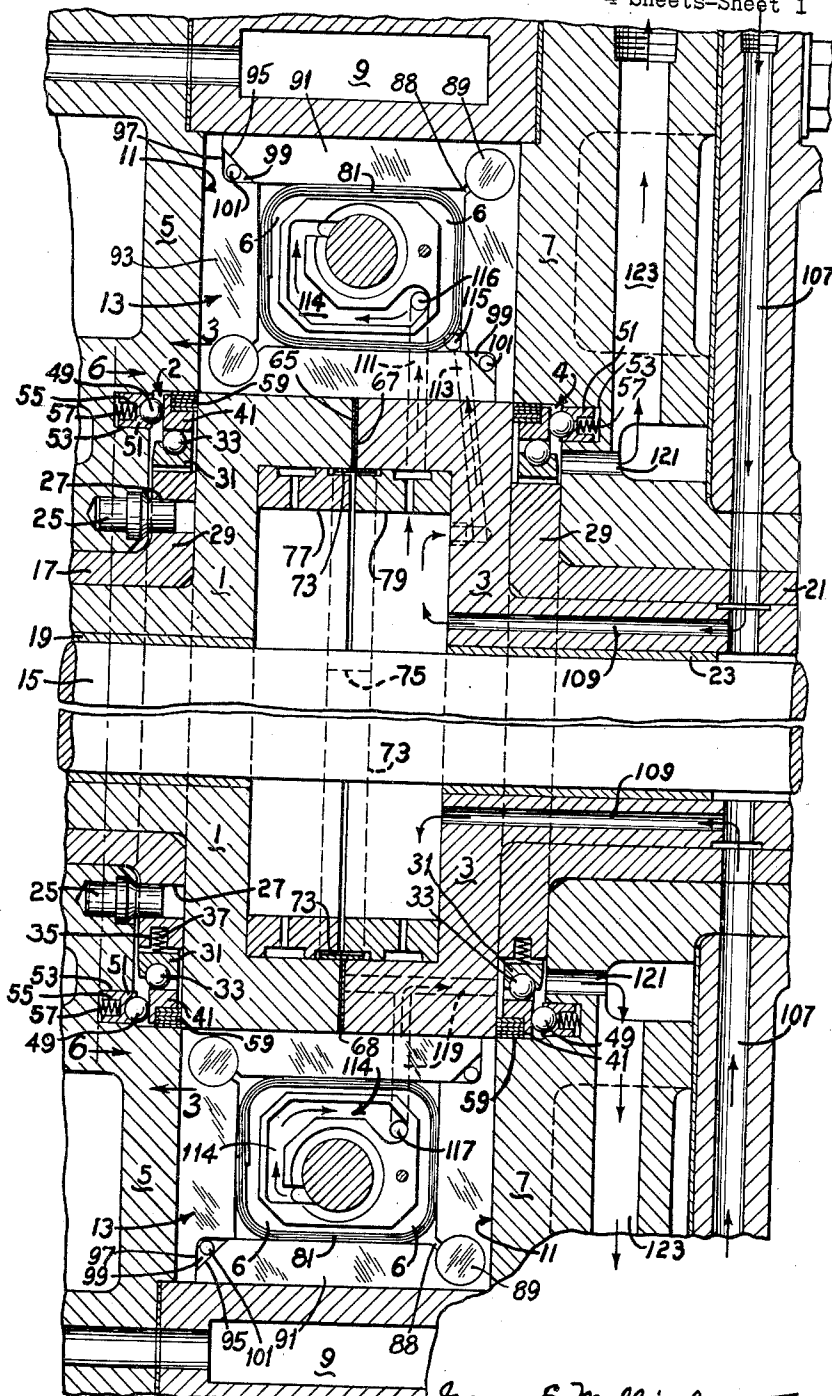
G. E. MALLINCKRODT
ROTARY ENGINE SEALING MEANS

3,070,074

Filed June 8, 1960

4 Sheets-Sheet 1

FIG. 1.



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FIG. 3.

FIG. 3 is a cross-sectional view of a multi-layered circular structure, likely a pipe or container, showing various layers and components labeled with numbers 1 through 59. The structure is shown in a cross-section with a central core (15) and multiple concentric layers. The outermost layer is labeled 59. The innermost layer is labeled 1. The structure is shown in a cross-section with a central core (15) and multiple concentric layers. The outermost layer is labeled 59. The innermost layer is labeled 1.

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FIG. 7.

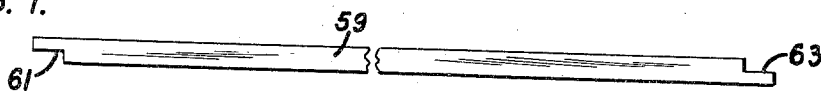


FIG. 5.

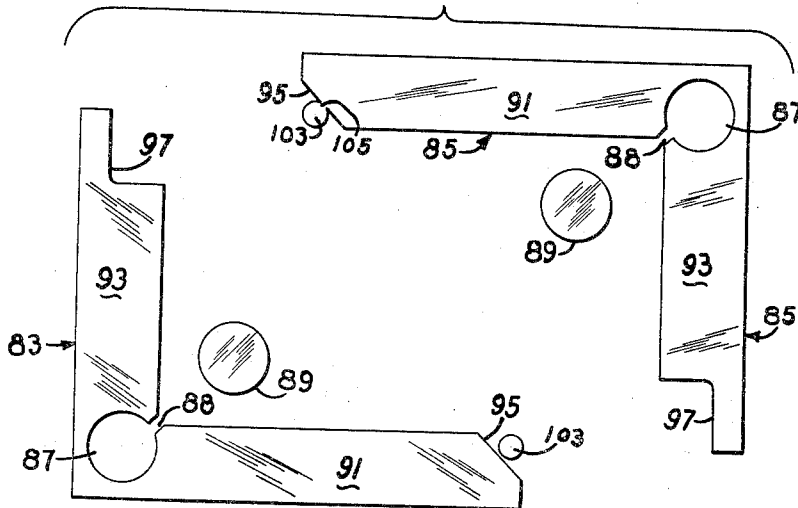
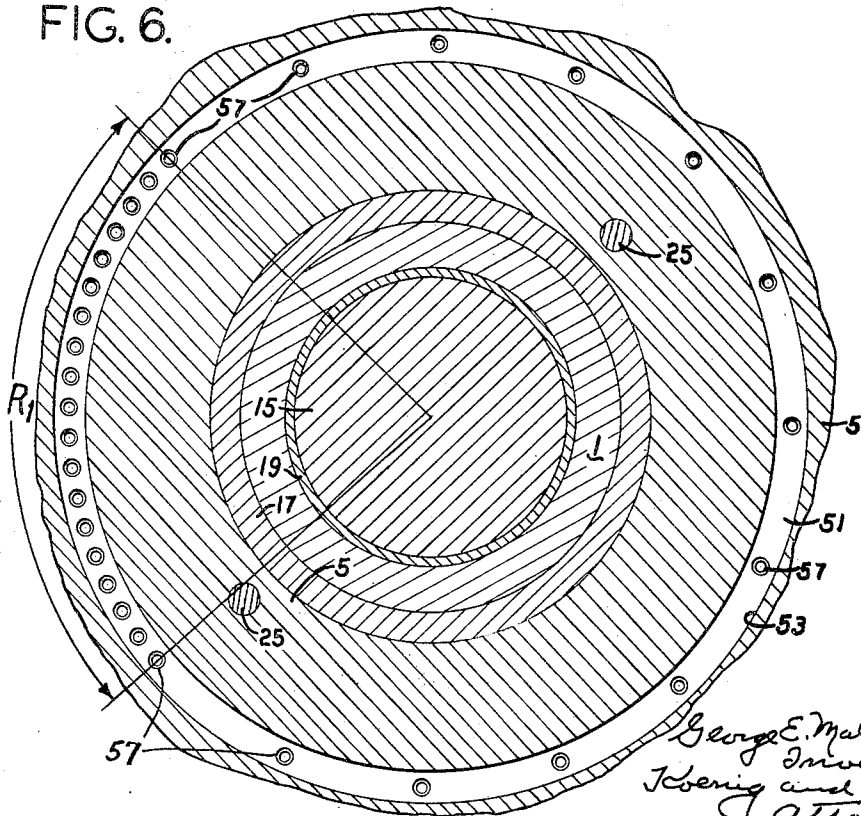


FIG. 6.



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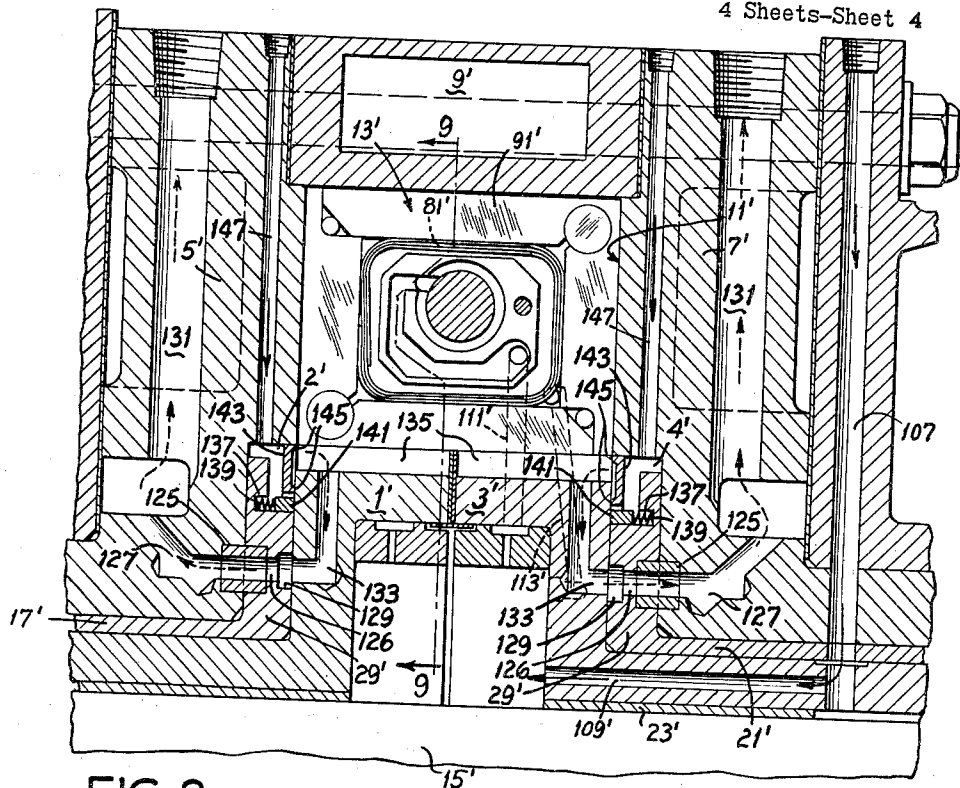


FIG. 8.

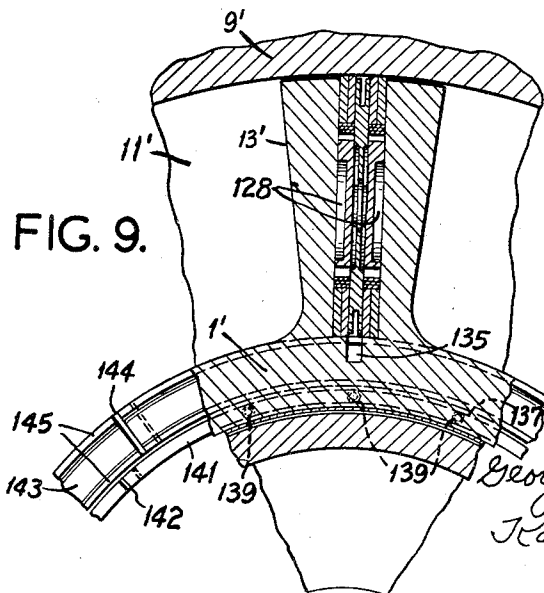


FIG. 9.

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ROTARY ENGINE SEALING MEANS

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Filed June 8, 1960, Ser. No. 34,784

24 Claims. (Cl. 121-49)

This invention relates to rotary engine sealing means, and more specifically to means for reducing gas leakage to or from rotary engine combustion chambers and for reducing oil leakage past their piston seals. The general class of rotary engines to which the invention relates and upon which it constitutes an improvement is shown for example by my United States Patents 2,796,216 and 2,834,322. These show so-called alternating-piston machines of this class.

Among the several objects of the invention may be noted the provision of a rotary internal combustion engine of the alternating-piston type, having improved means adapted to prevent leakage of the working medium to or from the combustion chambers through the running fits between their rotors and between such rotors and the stationary parts in which such rotors operate; the provision of apparatus of the class described in which the rotor seals are particularly effective against high-pressure losses during compression and expansion events; the provision of rotor sealing means of the class described which wear evenly and slowly; and the provision of piston seals for pistons of rotary engines which will seal against leakage therethrough in either direction relative to the oil supply system therefor. Other objects and features will be in part apparent and in part pointed out hereinafter.

The invention accordingly comprises the elements and combinations of elements, features of construction, and arrangements of parts which will be exemplified in the structures hereinafter described, and the scope of which will be indicated in the following claims.

In the accompanying drawings, in which one of various possible embodiments of the invention is illustrated,

FIG. 1 is a fragmentary axial section through an alternating-piston engine, illustrating the application thereto of one form of rotor sealing means made according to the invention, being taken on line 1-1 of FIG. 3;

FIG. 2 is an enlarged fragmentary axial section taken on line 2-2 of FIG. 3, showing details of a sealing means between a rotor and adjacent stationary parts;

FIG. 3 is a cross section, taken on line 3-3 of FIG. 1;

FIG. 4 is an enlarged fragmentary detail view of certain additional sealing means between rotors;

FIG. 5 is an exploded view, illustrating certain piston ring sealing elements;

FIG. 6 is a cross section, taken on line 6-6 of FIG. 1;

FIG. 7 is a developed view, illustrating one of several sealing rings;

FIG. 8 is a fragmentary view similar to FIG. 1, illustrating an alternative form of rotor sealing means; and

FIG. 9 is a fragmentary cross section taken on line 9-9 of FIG. 8.

Corresponding reference characters indicate corresponding parts throughout the several views of the drawings.

The general type of rotary internal combustion engines of the alternating-piston type to which this invention is directed appears in said Patents 2,796,216 and 2,834,322. Thus for a complete description of such an engine, said patents may be examined.

Referring now more particularly to FIG. 1, the adjacent ends of coaxial rotors of an alternating-piston engine are shown at numerals 1 and 3. These rotate in sta-

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tionary, cheek-forming casing pieces 5 and 7, respectively, which together with an outer ring 9 form an annular cylinder 11 of rectangular cross section. Adjacent the cylinder 11 are annular pockets 2 and 4. Each rotor 1 and 3 carries a number of identical pistons (four, for example), each group of four pistons being connected to one rotor and extending over the other so as to interdigitate with the pistons on the other rotor in the cylinder 11. Two typical pistons of this type on rotor 3 are indexed 13 in FIG. 1, being located at 180° intervals on this rotor. These pistons are of the oil-cooled and lubricated type set forth in said patents.

It will be understood that there are two additional opposite pistons on rotor 3, located in a plane at right angles to the paper; also that there are four similar pistons on rotor 1 but that these do not show in FIG. 1 because this rotor is shown in an angular relation with respect to rotor 3 and out of the plane of the paper. All of the pistons on both rotors are of the same rectangular form, such as shown in FIG. 1, each rectangular form substantially matching that of the cross section of the annular cylinder 11 formed by the cheeks of members 5 and 7, the inside of ring 9 and the cylindrical forms between pistons 13 of rotors 1 and 3. There is a running clearance between each piston and the cylinder walls 5, 7, 9 and the outsides of the mating rotor. This clearance requires sealing against gas and oil leakage as the pistons move relatively in the cylinder 11 in performing their operations in the cycle of power events. The rotors transmit the movements imparted to them by the pistons to a power shaft shown at 15 by suitable drive means (not shown, being known in the art).

Rotor 1 is carried in a sleeve bearing 17 located in the frame piece 5 and in turn carries an internal sleeve bearing 19 for the shaft 15. Rotor 3 is carried in a sleeve bearing 21 located in the frame piece 7 and also carries an internal sleeve bearing 23 for shaft 15. Each bearing 17 and 21 is radially flanged, as shown at 29, to provide an inner face which is approximately flush with a radial cheek of the cylinder 11.

Bearings 17 and 21 are prevented from rotating by dowels such as shown at 25, seated in the stationary member 5 and keyed into openings 27 in the bearings. The dowels 25 for holding the bearing 17 are shown, but those for holding bearing 21 do not appear because they are outside the plane of the paper in FIG. 1. The flanges 29 are located in pockets 2 and 4, respectively. The outside radial walls of the rotors 1 and 3 turn against the inner faces of flanges 29, at which areas leakage may occur from the suction, compression, combustion, and exhaust events occurring in the spaces between the pistons. It is to prevent this that annular side sealing assemblies shown in FIGS. 1, 2, 3 and 6 are employed for sealing the rotors relative to the side pieces 5 and 7. Since each annular seal is a duplicate of the other, a description of one will suffice for both, like numerals being employed for each.

Referring to the left-hand annular seal (FIGS. 1, 2, 3 and 6), numeral 31 indicates a radial ball race for roller or ball bearings 33. This race 31 surrounds the adjacent bearing flange 29. The latter, as best shown in FIG. 3, is marginally provided with a range R of pockets 35 in which biasing compression springs 37 are seated. The range R of pockets 35 and contained springs 37 is adjacent to the range of movements of the pistons 13 within which gaseous compression and explosive expansion events occur between the pistons 13, since any leakage during such events is the most critical to efficient operation.

The ball bearings 33 are surrounded by a radially tensioned, expansively springy annular race 41 which is split at one point, such as shown at 43 in FIG. 3, to per-

mit it to exert expansive force. The race 41 is L-shaped in cross section, having an inside smooth cylindrical surface 45 which radially supports the balls 33, and a flat radial surface 47 which supports ball bearings 49. The balls 49 are carried in an axial race 51, slidably located in an axially disposed groove 53 located in the stationary member 5 (or 7, as the case may be). The race 51 contains axially directed pockets 55 for the reception of axially locating biasing compression springs 57. The springs 57 react upon the stationary bottom of groove 53 to force the race 51 to press the balls 49 axially against the race 41.

Located in the angular seat of the L-shaped race 41 are a plurality (five in the present case) of circular ring-forming laminated spring sealing rings 59. A developed view of a typical one of the spring rings 59 is shown in FIG. 7. Each is oppositely notched at its opposite ends, as shown at 61 and 63, to form an interdigitated overlap when its opposite ends are brought together by looping it into circular form. It will be understood that since each ring 59 in the plurality is to be nested with others in the angular seat of the race 41 and each ring must be of different diameter, the developed length of each will be slightly different. Thus the developed length of each ring 59 that lies outside of another is slightly longer than the latter. It is also preferable that each alternate ring have opposite notchings 61 and 63 at its opposite ends reversed in aspect for maximum sealing effect when in sealing position, although this is not absolutely necessary.

As shown in FIG. 6, the locations of the springs 57 are distributed around the periphery of the race 51, with a group of them being bunched closely in the range R_1 . The range R of the springs 37 shown in FIG. 3, and the range R_1 of the springs 57 shown in FIG. 6, appear on opposite sides of these figures but they are actually adjacent one another, the cross sections of FIGS. 3 and 6 being viewed in opposite directions in FIG. 1. Thus the angular range R (FIG. 3) of the springs 37, and the angular range R_1 (FIG. 6) of the closely spaced groups of springs 57, lie contiguously. The result is that the ribbon-like spring rings 59 have their flat sides more forcefully pushed radially against the stationary member 5 (or 7, as the case may be) in the range R than elsewhere. Also, the spring rings 59 have their marginal edges more forcefully pushed against their contiguous rotor 1 (or 3, as the case may be) in the range R_1 than elsewhere. An important feature also is that the axial loading of the assembly of rings 59 by the axial springs 57 is so related to the radial loading of assembly 59 by the radial springs 37, that the predominant frictional drag effect is between the assembly 41 and its adjacent variable-velocity rotor 1 (or 3, as the case may be). The advantage of this arrangement is as follows:

As the rotors 1 and 3 alternate their rotations under successive events between their pistons 13 in the range R or R_1 , there will be a substantial radial sealing effect of the rings 59 on the member 5 (or 7, as the case may be) in the range R , and there will be a substantial axial sealing effect of these rings on the rotor 1 (or 3, as the case may be) in the contiguous range R_1 . Not only does this have the advantage of providing a very effective seal against leakage from the critical compression and expansion events between the pistons 13, but it has the added advantage that the edges of the groups of rings 59 are urged into predominantly strong frictional engagements with their respective rotors so that these rings tend to be dragged with their respectively adjacent rotors. Thus as each rotor rotates under conditions of low acceleration, the adjacent ring 59 tends to rotate with it, along with the adjacent race 41. At this time the outermost ring 59 will rub on 5 or 7, as the case may be. On the other hand, when a rotor attains substantial acceleration or deceleration, as at the end of a compression event and the start of an expansion event, the inertia of the assembly of adjacent race 41 and rings 59 will result in slip of such rings

with respect to the rotor. Some relative slippage will also occur intermittently between the rings 59 themselves. This in general has the effect of constantly bringing new contact surfaces into substantial pressure engagement between the rings, rotors and frame pieces, with the result that even wear and long effective sealing life accrue. In short, the assemblies of rings 59 intermittently creep around the pockets 2 and 4 as they become frictionally connected to and disconnected from the rotors as the latter vary their angular velocities in performing their cyclic power functions. The inner surface of each race 41 is cylindrical, so that it may move freely in an axial direction with respect to the balls 33. Also, the radial face of each race 41 is flat, so that it may move eccentrically under eccentric pressure of the range R of springs 37.

Sealing means are also required at the axial abutments between the rotors 1 and 3. Such sealing means are shown enlarged in FIG. 4. One component of these means is constituted by two adjacent flat rings 65 and 67 which are located between the rotors 1 and 3 and which are located between the rotors 1 and 3 and which abut one another. These are welded as shown at 69 along their inside interfacial regions, being left unwelded at their outside interfacial regions, as shown at 71. Their margins in region 71 are adjacently beveled as shown at 68. Inside the rings 65 and 67 is located an outwardly springy split sleeve 73. The split is shown at 75 in FIG. 1. This sleeve is held in place by rings 77 and 79, which have press fits in hollow ends of the rotors 1 and 3. The rings 77 and 79 not only hold the sleeve 73 in position, but form annular oil passages. Any leakage that tends to occur from the spaces outside of the rotors 1 and 3 from between the pistons 13 thereon is blocked by the rings 65, 67 and sleeve 73. The fact that the welded rings 65 and 67 are unwelded between them at their outer peripheries enhances their sealing effects, since under leakage pressure they tend to spread apart axially against the rotors, particularly if they are adjacently beveled as at 68.

Each piston 13 is of the general oil-cooled and lubricated type shown in the above-mentioned patents, having similar angular piston sealing parts in ring slots 6 which form sealing rings pressed outward by means of springs such as 81. The ring-sealing parts have been improved herein, as illustrated in FIG. 5. They are flat and adapted to be paired in piston slots such as indicated at 6. Each coordinated pair of ring-sealing parts is constituted by two flat angle members 83 and 85 which are identical, as shown. The angle members have openings 87 at their corners for the loose reception and retention of discs 89 which have thicknesses the same as those of the members 83 and 85. The openings 87 are connected to the interior corners of the ring-forming members by slots 88. The holes 87, in view of the slots 88 and somewhat loose fits of the discs 89 therein, provide for springiness between the right-angled legs 91 and 93. The discs 89 prevent leakage through the holes 87 when the discs are located in the holes, as shown in FIG. 1.

Each leg 91 is beveled as shown at 95, and each leg 93 is provided with a cooperating right-angular notch, as shown at 97. The notches 97 and bevels 95 are adapted to be brought in cooperative relation as shown in FIG. 1, leaving oppositely converging or triangular openings 99 therebetween. For sealing purposes, small freely movable or floating discs 101 are located in said triangular openings 99. Location is accomplished, as shown in FIG. 5, i.e., by manufacturing each ring member, which is hardened, with a small disc 103 attached by a very thin breakable neck 105 to the respective bevel 95 and of equal thickness thereto. This arrangement is illustrated in connection with the angle member 85, as shown in FIG. 5. Each disc 103 thus becomes automatically placed in a triangular opening 99 when the angle members are cooperatively assembled in the ring slot 6 of a piston 11. Then when the engine is started, vibration breaks

away each disc 103, as suggested in connection with the

member 83 shown in FIG. 5. The result is as shown in FIG. 1, wherein each triangular opening 99 contains a free-floating disc 101 which will seal the abutment between the legs 91 and 93 of the members 83 and 85, regardless of the direction in which leakage tends to occur through the openings 99. Under either inwardly or outwardly directed gaseous pressure, the discs 103 will take up wedging positions in the corners of the triangular spaces 99, which will seal against either inward or outward leakage.

Piston oil-cooling and lubricating means are described in said patents. As applied in the present construction, lubrication occurs as follows:

Oil inlet ports are located at 107. These have branches 109 leading into the space between rotors. As shown in the case of rotor 3 in FIG. 1, oil may escape through ring 79, passage 111, via port 116 and into the center circulating passages 114 of the upper piston 13. This is for cooling purposes as set forth in said patents. A return passage is shown in connection with the other lower piston 13 shown in FIG. 1, wherein numeral 117 is the outlet from the center cooling passage 114. This communicates through passages 119 with the spaces around the adjacent sealing assembly shown at the right and thence through ports 121 and outlet passage 123. The passages 123 and 107 are connected by a suitable pressurized circulating system, not shown.

Some of the oil from between rotors 1 and 3 escapes through passage 113 and enters each piston through a port 115 for lubricating the angle sealing members 83 and 85. Very little of the oil delivered through the port 115 for lubricating the angle members is lost through the sealing rings 59, but whatever amount of it may be lost will return through ports 121 to the outlet passages 123.

It will be noted, as regards the description of the oil circuits, that for clarity, inlets such as 111, 113 for the pistons are shown only in connection with the upper piston 13 in FIG. 1, and that the outlets such as 119 are shown only in connection with the lower piston 13 there shown. This is to avoid confusion in illustration, but it will be understood that each piston on each rotor contains a full complement of passages such as 111, 113 and 119. It will also be understood that the passages are formed through the material composing the connections between the respective pistons and the central portions of the rotors on which they are mounted. In FIG. 1 no passages are shown for pistons on the rotor 1 because its pistons are out of the plane of the section. Further details regarding the oil circulation will not be required, in view of the disclosure of similar oil-circulating means described in said patents.

FIGS. 8 and 9 illustrate another form of the rotor sealing means which is alternative to that shown in FIGS. 1-3. In order to avoid repetitious description, parts in FIGS. 8 and 9 that correspond in function to those shown in FIGS. 1-3 have the same numbers except that the numbers are primed, since the detailed shapes of some of such corresponding parts are slightly different in FIGS. 8 and 9. Description of the prime numbered parts will not be repeated. Points of difference will be found in the following description, wherein additional numerals are used for parts not shown or described in connection with FIGS. 1-3.

Referring to FIGS. 8 and 9, the flanges 29' of the bearings 17' and 21' are modified so as to be keyed in position by means of hollow thimbles 125. These thimbles connect with passages 127 in casing parts 5' and 7'. They also connect with peripheral grooves 129 in the flanges with outlets 131 (see openings 126). The rotors 1' and 3' contain passages 133, which connect with the grooves 129 and also with transverse slots 135 in the respective rotors. The slots 125 are at the bases of the spaces 128 in the pistons 13' which contain the piston sealing parts. The flanges 29' are marginally modified to form recesses 137 for axial spring 139, adapted to press rings 141 against the rotors 1' and 3', respectively. The rings 141

are of the split lapped-end type, indicated at 142 in FIG. 9.

In the pockets 2', 4' are located ribbon-like sealing rings 143 which have axial lands 145 seating axially on the margins of the rotors. These rings are of the split end lapped type, tending to spring outward into springing engagement with the stationary members 5' and 7', respectively. These tend to form radial seals with the stationary parts 5' and 7'. The split lapped-end arrangement is illustrated at 144 in FIG. 9. Passages 147 convey oil under pressure to the pockets 2', 4', thus tending to press the lands 145, sealing rings 143 against the respective rotors 1' and 3'. Thus radial and axial seals are obtained corresponding to those provided by the ribbons 59 in FIG. 1. Any leakage of oil past the sealing rings 143 moves through pressure relief passages 135, 133, 129, thimbles 125 and passages 127 to outlet passages 131. Any leakage that passes the rings 141 also finds its way between the parts 29' and the respective rotors 1' and 3' to the passages 131 via passages 129, 126, thimbles 125 and passages 127.

In view of the lands 145, unbalanced pressure is applied to the rings 143 by the oil pressure arriving through passages 147. This pressure is adjusted so as to force the rings toward the respective rotors 1' and 3', causing them to tend to move with the rotors during periods of comparatively low rotor acceleration or deceleration but allowing these rings 143 to slip with respect to the rotors under conditions of comparatively higher accelerations and decelerations. Thus it will be seen that as the engine operates the rings 143 will alternately cling to and release from the respective rotors. Thus the rings 143 intermittently creep from one position to another relative to the rotors and parts 5' and 7', with resulting minimization and equalization of wear. The optimum condition of intermittent creep of the rings 143 may be obtained by adjusting the pressure in the inlets 147.

Advantage of various features of the invention are as follows:

(1) The rotor sealing rings 59 are strongly pressed into sealing engagements with the respective rotors 1 and 3 over the range R_1 shown in FIG. 6, which has a two-fold result:

(a) Adequate sealing is obtained in the range of compression and expansion or firing events between pistons, under which conditions most leakage ordinarily occurs; and,

(b) The comparatively high contact pressure, with pressures against the respective rotors, causes the above-mentioned tendency of the ring seals 59 to rotate with the rotors throughout temporary intervals of low acceleration or deceleration, but to slide with respect to them throughout intervals of higher acceleration or deceleration, thereby causing evenly distributed wear and consequent long life. Thus each ring seal 59 intermittently rotates or creeps with respect to the stationary parts 5 and 7.

(2) The localized radial spring pressure on race 41 over range R, as shown in FIG. 3, enhances the radial seal against stationary portions of the engine in the above-mentioned range R of compression and expansion events.

(3) The inwardly partially welded adjacent rings 65, 67, and the inner spring sleeve 73, effectively prevent leakage from the space between pistons into the space between rotors. This is because of the spreading action which may occur between the free outer parts 71 of the rings 65 and 67.

(4) The improved abutments between legs 91 and 93 of the angular sealing ring members 83 and 85 with the associated floating discs 103, reduce oil and gas leakage at said abutments, whether or not the leakage tends to occur inwardly, as in the case of suction events between

pistons, or outwardly, under compression, expansion or exhaust events.

(5) The self-releasing discs 103 provide for their convenient insertion when the sealing components 83 and 85 are placed in the pistons 13.

In view of the above, it will be seen that the several objects of the invention are achieved and other advantageous results attained.

As various changes could be made in the above constructions without departing from the scope of the invention, it is intended that all matter contained in the above description or shown in the accompanying drawings shall be interpreted as illustrative and not in a limiting sense.

I claim:

1. Sealing means for use between a rotor and casing of a rotary alternating-piston machine, comprising circular and radially springing ring-sealing means radially springingly engaging the casing and axially engaging the rotor, said ring-sealing means comprising an assembly of axially laminated circular metal ribbons biased edge-wise and axially against the side of the rotor and biased radially flatwise against the wall of the cylindrical pocket in the casing, both engagements being with circular sliding contacts, and means adapted to bias the ring-sealing means into rotor engagement with a pressure sufficient that the ring will move with the rotor under conditions of comparatively low rotor acceleration or deceleration but will slip with respect to the rotor under conditions of comparatively high rotor acceleration or deceleration.

2. Sealing means according to claim 1, wherein said ring-sealing means includes axially and radially disposed roller-engaging surfaces, roller means engaging said surfaces, and axially and radially operative roller forcing means.

3. Sealing means according to claim 2, wherein said axially and radially operative roller forcing means are axially and radially movable roller races backed axially and radially by arrays of expansive springs.

4. Sealing means according to claim 3, wherein each array of springs is concentrated throughout a limited circular range of the same.

5. Sealing means for use between the rotors and casing of a rotary alternating-piston machine wherein the rotors accelerate and decelerate angularly, comprising ring seals adapted respectively rotatably to engage a rotor and the casing with circular sliding contacts, each ring seal comprising an assembly of axially laminated circular metal ribbons biased edgewise and axially against the side of the rotor, and biased flatwise against the wall of the cylindrical pocket in the casing, and means adapted to bias each ring seal into rotor engagement with a greater frictional circular drag therebetween than its frictional circular drag in its engagement with the casing, said greater frictional circular drag being sufficient to carry each ring seal with a rotor at rotor speeds during periods of relatively low rotor acceleration and inertially to release the ring seal from a rotor during periods of relatively high acceleration and deceleration of the latter.

6. Sealing means according to claim 5, including an annular first race supporting each ring seal, said race having inside cylindrical and endwise flat forms, second inner and third endwise movable ball races carrying balls engaging said forms respectively and inner radial and endwise axial biasing means positioned to bias said second inner and third endwise races toward the first race.

7. Sealing means according to claim 6, wherein said biasing means are constituted by localized arrays of springs operative on said second and third races.

8. Sealing means for use between a rotor and casing of a rotary alternating-piston machine, comprising a ring seal composed of a laminated assembly of circular metal

ribbons located in a cylindrical pocket of the casing and arranged for sliding contact therewith and with a side of the rotor, a first race supporting said assembly and having a flat side and a cylindrical interior, a second ball race carrying balls biased thereby into engagement with said cylindrical interior of said first race, a third axially movable ball race carrying balls engageable with said flat side of the first race, and an array of radially disposed springs engaging said third ball race adapted to force the balls thereof into axial engagement with said first race.

9. Sealing means according to claim 8, wherein said first race is split and outwardly sprung within said assembly.

10. Sealing means for use between a rotor and casing of a rotary alternating-piston machine, comprising a ring seal composed of a laminated assembly of circular metal ribbons each having abutted ends and located in a cylindrical pocket of the casing and arranged for facewise sliding contact therewith and located adjacent a side of the rotor for edgewise sliding contact therewith, a first split and outwardly sprung race of L-shaped cross section supporting said assembly and having a flat side and a cylindrical interior, an eccentrically movable second ball race carrying balls engaging within said cylindrical interior, an array of radially disposed springs within said second ball race adapted to force said second ball race radially against said first L-shaped race with an eccentric application of force, a third axial movable ball race carrying balls engageable with said flat side of the first race, and an array of radially disposed springs engaging said third ball race adapted to force said third ball race radially with said first race with an eccentric application of force.

11. Sealing means according to claim 10, wherein the ranges of said eccentric applications of forces are substantially coextensive.

12. Sealing means for use between abutting rotors of alternating-piston engines comprising adjacent rings between said rotors and respectively engaging them flatwise, the inside margins of said rings being joined and their outside margins being free.

13. Sealing means according to claim 12, including a cylindrical spring ring located within the joined margins of said adjacent rings.

14. Sealing means for use between abutting rotors of alternating-piston engines, comprising interfacially engaged flat rings between said rotors respectively engaging them flatwise, the interfacial engagements between said rings being circularly connected throughout an interior area and unconnected throughout an exterior area.

15. Sealing means according to claim 14, wherein adjacent margins of said unconnected area are beveled.

16. Sealing means for use between a rotor and a recessed casing of a rotary alternating-piston machine, comprising a ring seal composed of a split spring ring radially sprung into engagement with the recess in the casing, said ring having lands axially engaging the rotor, means for introducing fluid under pressure into said recess to force the ring against the rotor, and fluid relief passages in the rotor and the casing connecting with the space between said lands.

17. Sealing means according to claim 16, wherein said relief passages include a circular passage in the casing with which the portion of the relief passage in the rotor continuously connects.

18. Sealing means according to claim 16, including an additional ring seal between the casing and the rotor adapted to resist leakage under pressure that might occur between the casing and the rotor to said relief passages.

19. Ring-sealing means for rectangular pistons comprising flat cooperatively arranged L-shaped members, each member having two joined side legs having free

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ends abutting the free ends of the legs of the other L-shaped member respectively substantially to form a rectangle of the cooperatively arranged L-shaped members, the abutting ends of the legs being shaped to form wedge-shaped enclosing pockets, and a movable flat sealing member trapped in each pocket for movement into wedging engagement therein with the adjacent leg ends.

20. Ring-sealing means according to claim 19, wherein each pocket is in the general form of a triangle and said sealing member is of circular disc form.

21. Ring-sealing means according to claim 20, wherein prior to assembly in a piston each disc has a temporary connection with one free end portion of one leg of one L-shaped member so as to be positioned in a pocket upon assembly and to break free upon operation of the piston for floating movements in the respective pocket.

22. Sealing means for use between a rotor and casing of a rotary alternating-piston machine, comprising circular and radially springing ring-sealing means radially springingly engaging the casing and axially engaging the

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rotor, both engagements being with circular sliding contacts, said ring-sealing means having axially and radially disposed roller-engaging surfaces, roller means engaging said surfaces, and axially and radially operative roller forcing means.

23. Sealing means according to claim 22, wherein said axially and radially operative roller forcing means are axially and radially movable roller races backed axially and radially by arrays of expansive springs.

24. Sealing means according to claim 23, wherein a number of springs in each array is concentrated throughout a limited circular range which subtends less than 360 degrees of arc.

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UNITED STATES PATENT OFFICE
CERTIFICATE OF CORRECTION

Patent No. 3,070,074

December 25, 1962

George E. Mallinckrodt

It is hereby certified that error appears in the above numbered patent requiring correction and that the said Letters Patent should read as corrected below.

In the grant, lines 2 and 3, for "Elliot Enterprises, Incorporated, of St. Louis, Missouri" read -- Elliot Enterprises, Incorporated, of St. Louis County, Missouri --; in the heading to the printed specification, lines 3 and 4, for "Elliot Enterprises, Incorporated, St. Louis, Mo." read -- Elliot Enterprises, Incorporated, St. Louis County, Mo. --; column 3, line 70, for "ttime" read -- time --; column 4, line 21, strike out "which are located between the rotors 1 and 3 and --; column 6, line 12, strike out the comma.

Signed and sealed this 9th day of July 1963.

(SEAL)

Attest:

ERNEST W. SWIDER
Attesting Officer

DAVID L. LADD
Commissioner of Patents