Sept. 3, 1963 M. BENTELE ETAL 3,102,492

COMPENSATED ROTARY MECHANISM CONSTRUCTION

Filed May 10, 1961 6 Sheets-Sheet 1

FIG. 1.

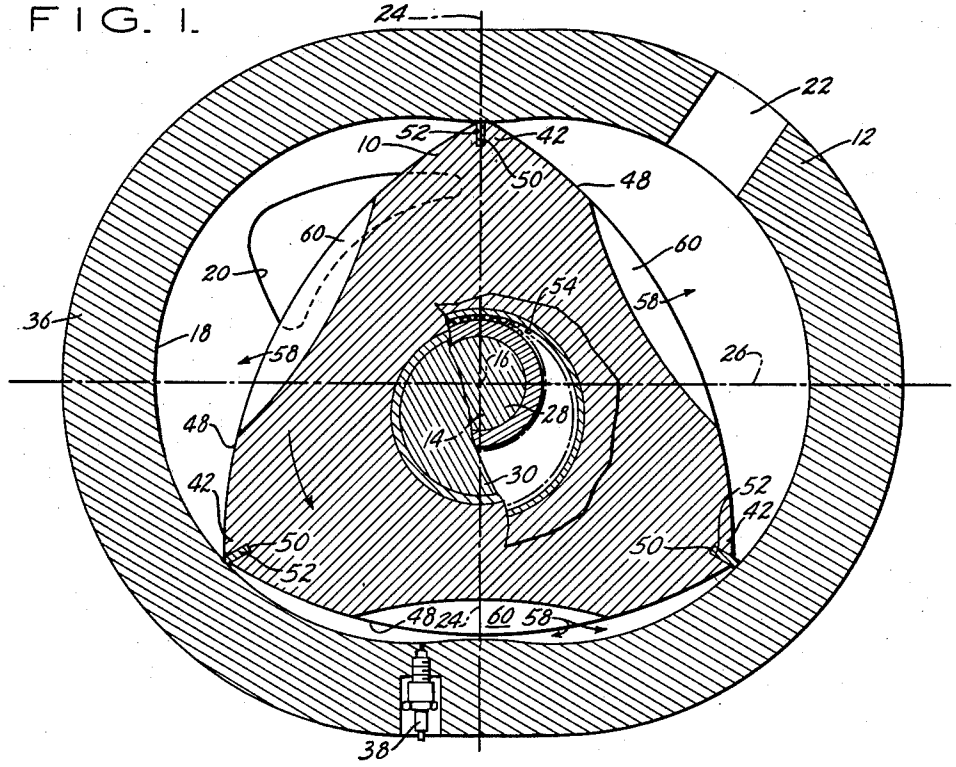
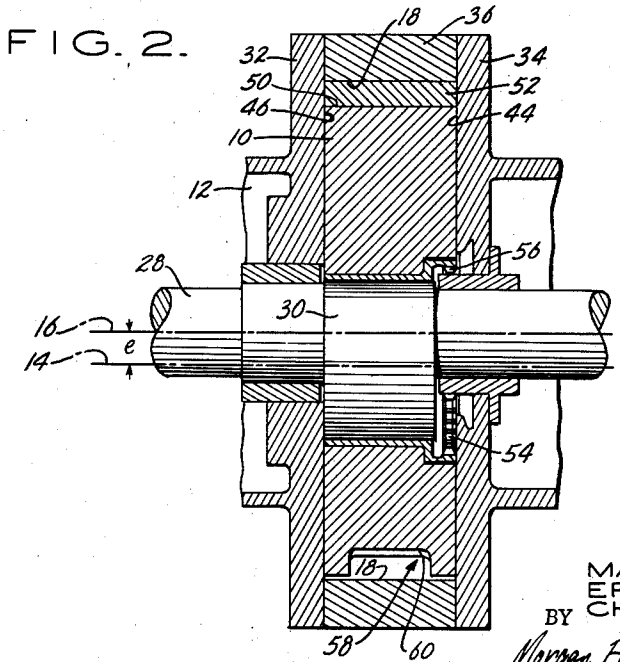


FIG. 2.



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COMPENSATED ROTARY MECHANISM CONSTRUCTION

Filed May 10, 1961 6 Sheets-Sheet 2

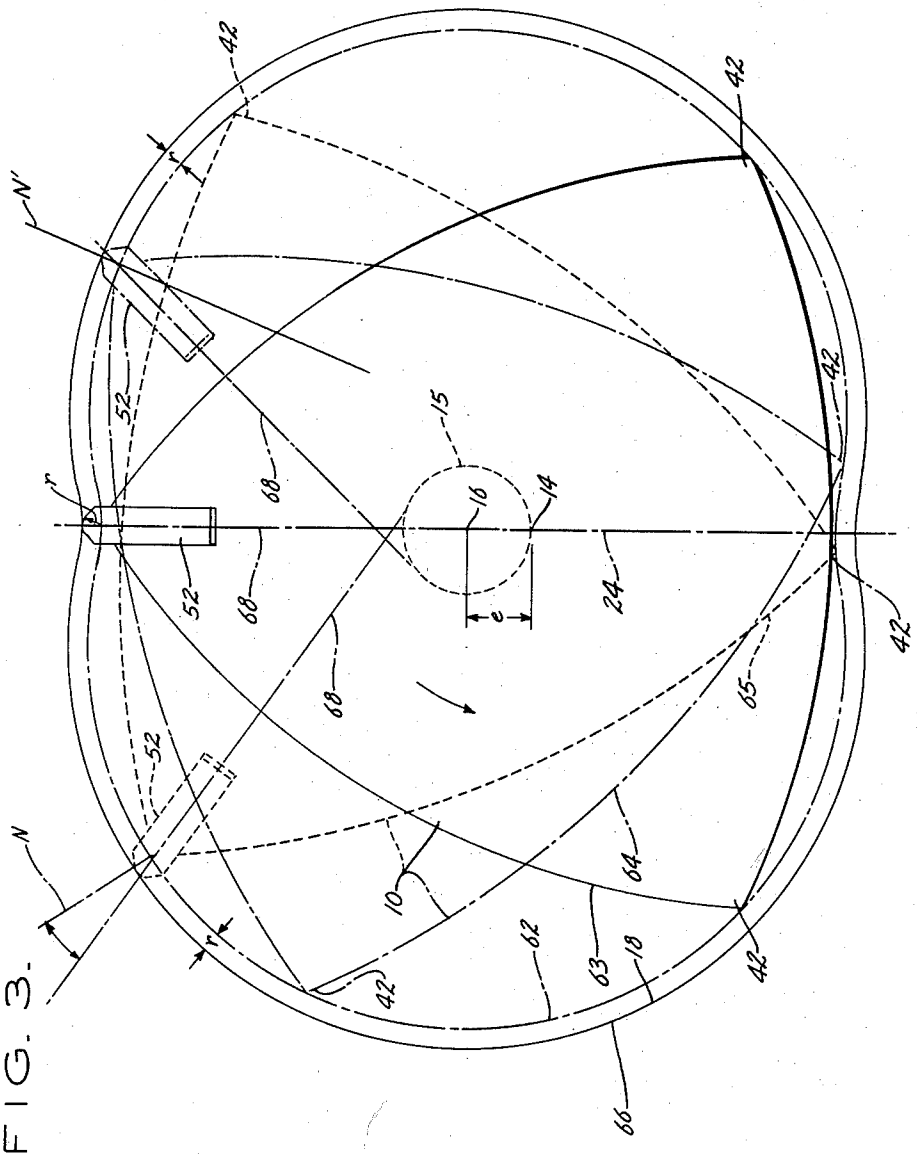


FIG. 3.

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COMPENSATED ROTARY MECHANISM CONSTRUCTION

Filed May 10, 1961 6 Sheets-Sheet 3

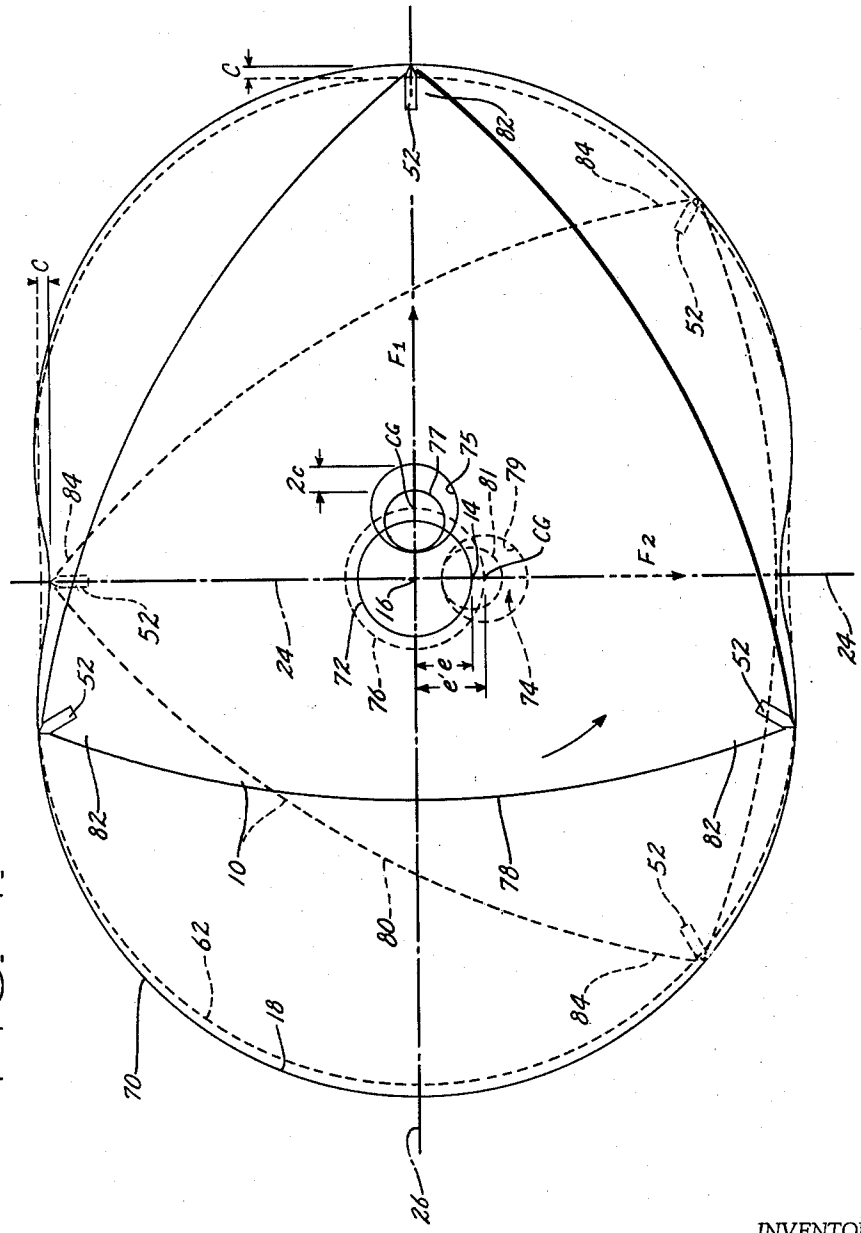


FIG. 4.

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COMPENSATED ROTARY MECHANISM CONSTRUCTION

Filed May 10, 1961 6 Sheets-Sheet 4

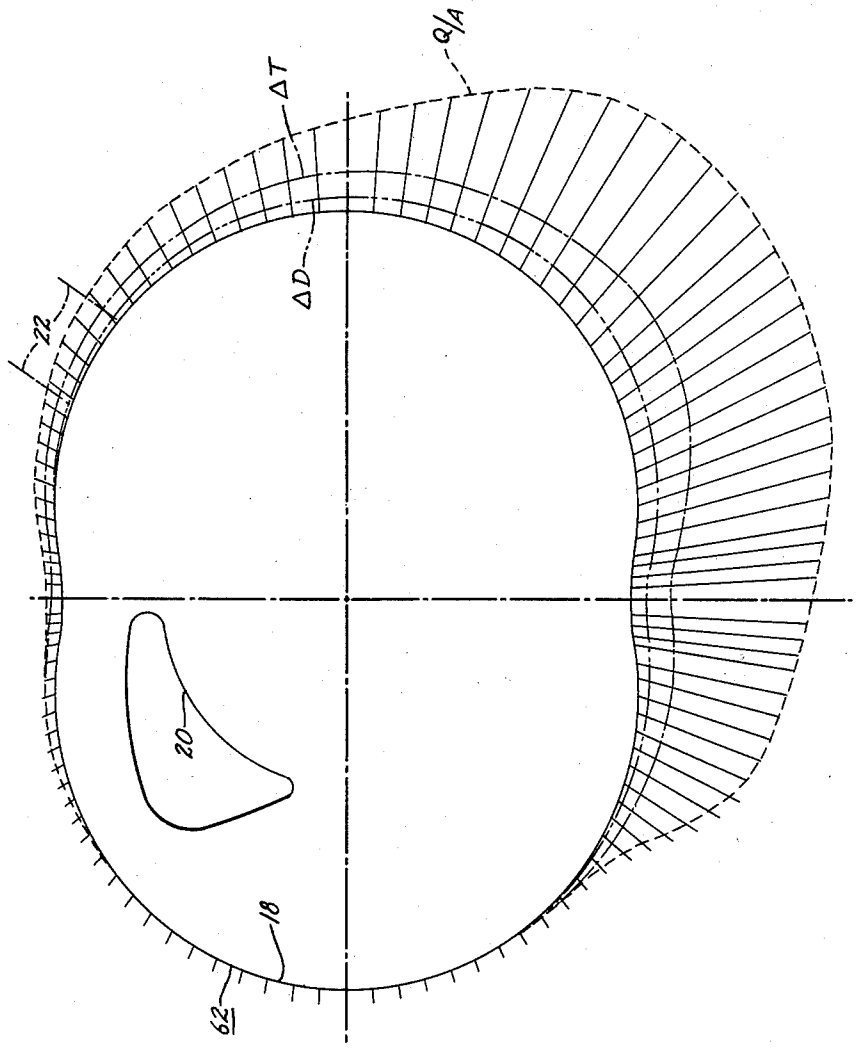


FIG. 5.

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COMPENSATED ROTARY MECHANISM CONSTRUCTION

Filed May 10, 1961 6 Sheets-Sheet 5

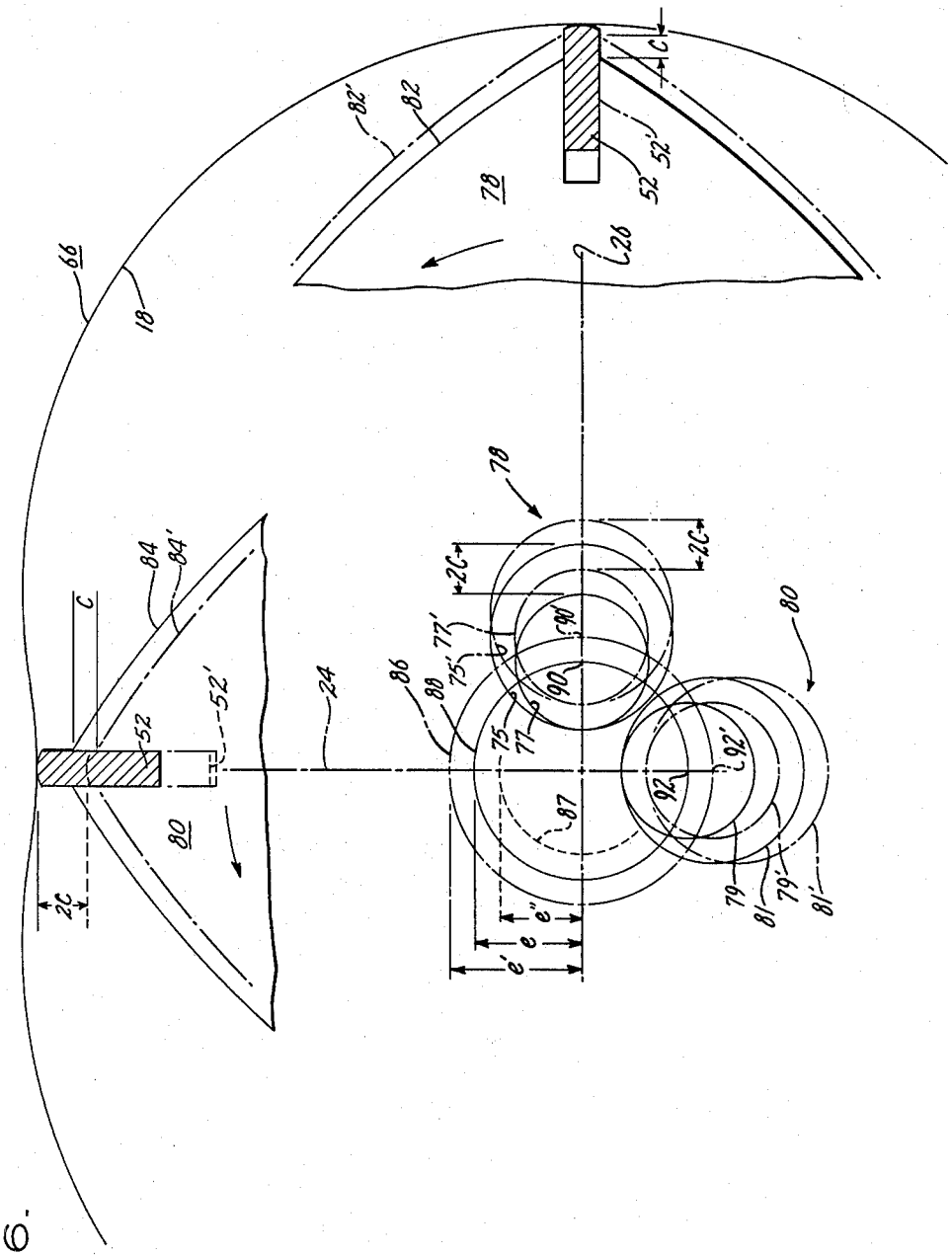


FIG. 6.

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COMPENSATED ROTARY MECHANISM CONSTRUCTION

Filed May 10, 1961 6 Sheets-Sheet 6

FIG. 8.

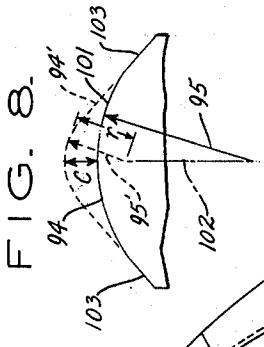


FIG. 7.

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United States Patent Office 3,102,492 Patented Sept. 3, 1963

3,102,492

COMPENSATED ROTARY MECHANISM CONSTRUCTION

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Filed May 10, 1961, Ser. No. 110,308
12 Claims. (Cl. 103—130)

This invention relates to improvements in sealing between working chambers of rotary mechanisms and, more particularly, to improvements in the sealing contact of rotor apex seals to minimize radial movement of the apex seals relative to the rotor, especially when a large pressure difference exists across the seal, and to compensate for deviations in seal positions relative to the rotor which are caused by deflections and displacements of the rotor, main shaft, and the like from their geometric centers during rotation and also to compensate for deviations in apex seal positions due to distortion of the outer body of the mechanism because of non-uniform temperature distribution as the rotor rotates relative to the outer body during operations of the mechanism.

Although this invention is applicable and useful in almost any type of rotary mechanism, such as combustion engines, fluid motors, fluid pumps, compressors, and the like, it is particularly useful in rotating combustion engines. To simplify and clarify the explanation of the invention, the description which follows will, for the most part, be restricted to the use of the invention in a rotating combustion engine. It will be apparent from the description, however, that with slight modifications, which would be obvious to a person skilled in the art, the invention is equally applicable to other types of rotary mechanisms.

The present invention is particularly useful in rotating combustion engines of the type which comprise an outer body having an axis, axially-spaced end walls, and a peripheral wall interconnecting the end walls, and an inner body or rotor which is mounted within the cavity formed between the inner surfaces of the peripheral wall and the end walls of the outer body. The inner surface of the peripheral wall is preferably parallel to the axis of the cavity and, as viewed in a plane transverse to this axis, the inner surface preferably has a multi-lobed profile which has basically the form of an epitrochoid. The axis of the rotor is eccentric from and parallel to the axis of the cavity of the outer body, and the rotor has axially-spaced end faces disposed adjacent to the end walls of the outer body and a plurality of circumferentially-spaced apex portions. The rotor is rotatable relative to the outer body and its apex portions continuously engage the inner surface of the peripheral wall in gas-sealing contact to form between the outer surface of the rotor and the inner surface of the outer body a plurality of working chambers which vary in volume during engine operation, as a result of relative rotation between the rotor and the outer body.

Such engines also include an intake passage for administering a fuel-air mixture to the chambers, an exhaust passage for removing burned gases from the chambers, and suitable ignition means so that during engine operation the working chambers of the engine undergo a cycle of operation which includes the four phases of intake, compression, expansion, and exhaust. This cycle of operation is achieved as a result of the relative rotation between the inner rotor and outer body, and for this purpose both the inner rotor and outer body may rotate at different speeds, but preferably the inner rotor rotates while the outer body is stationary.

For efficient operation of the engine, its working chambers should be sealed, and therefore an effective seal is provided between each rotor apex portion and the inner surface of the peripheral wall of the outer body, as well as between the end faces of the inner rotor and the inner surface of the end walls of the outer body.

Between the apex portions of its outer surface the rotor has a contour which permits its rotation relative to the outer body free of mechanical interference with the multi-lobed inner surface of the outer body. The maximum profile which the outer surface of the rotor can have between its apex portions and still be free to rotate without interference is known as the "inner envelope" of the multi-lobed inner surface and the profile of the rotor, which is illustrated in the accompanying drawings, approximates this "inner envelope."

For purposes of illustration, the following description will be related to the present preferred embodiment of the engine in which the inner surface of the outer body is basically a two-lobed epitrochoid, and in which the rotor or inner body has three apex portions and is generally triangular in cross-section but has curved or arcuate sides.

It is not intended that the invention be limited, however, to the form in which the inner surface of the outer body is basically a two-lobed epitrochoid and the inner body or rotor has only three apex portions. In other embodiments of the invention the inner surface of the outer body may have a different plural number of lobes with a rotor having one more apex portion than the inner surface of the outer body has lobes.

In generation of the housing or outer body for the rotating combustion engine, it has been the practice to establish the contour of the inner surface of the outer body by that shape resulting from the initial generation of a true epitrochoid having as a generating point the center of curvature of the tip of the apex seal and adding an amount r equal to the radius of curvature of the tip of the apex seal normal to the true epitrochoid over its entire length. This arrangement makes it possible to use a seal having an arcuate contact surface instead of a point contact which would obviously be impractical as an effective sealing means between adjacent working chambers of the engine. A further description of this construction may be found in copending application Serial No. 638,127, filed February 4, 1957, now Patent No. 2,988,008.

The resultant contour of the inner surface of the outer body provides a surface for contact with the apex seal tip throughout the entire revolution of the rotor with no radial motion of the seal relative to the rotor notwithstanding the fact that the seal tip has an arcuate surface rather than a point contact. Of course, this lack of radial motion of the seal relative to the rotor can take place only if the rotor and main shafts run at their geometric centers at all times with no deflection and provided there are no distortions of the inner surface of the housing or outer body due to other causes such as temperature gradients in the outer body.

In operation of the engine, centrifugal forces act on the rotor in a manner which results in movement of the rotor from its geometric center by an amount substantially equal to the bearing clearances and shaft deflections in a radial direction away from the axis of the outer body substantially through the rotor axis. If in spite of this shift or displacement, and as is necessary for proper sealing, an apex seal is to maintain contact with the inner surface of the outer body, the seal must move radially relative to the rotor. At the major axis the seal must move into the rotor and at the minor axis the seal must move out of the rotor (there is forced movement of the apex seal relative to the rotor, since at the major axis of the epitrochoidal inner surface the seal is forced to move

into the rotor and at the minor axis the seal is forced to move out of the rotor).

In a sense, the shape of an epitrochoid is governed by its K factor where

$$K=\frac{R}{e}$$

and where R is the radius of the rotor or distance from the rotor axis to the point of contact of a rotor apex portion with the epitrochoidal inner surface, and where e is the eccentricity of the rotor axis from the axis of the epitrochoidal inner surface of the outer body. From the equation

$$K=\frac{R}{e}$$

it is apparent that if R remains constant and e is increased, the K factor of the epitrochoid must become smaller.

It is a primary object of the present invention to provide compensating means to partially or fully correct for the effects of centrifugal forces which cause displacements of the rotor from its geometric center and for distortions caused by temperature gradients. These displacements and distortions, if not compensated for, result in relative radial movement of the apex seal to the rotor.

This compensating means can be achieved in three ways:

1. OUTER BODY COMPENSATION

Movement of the rotor from its geometric center due to influence of centrifugal forces is equivalent to an increase in the eccentricity (e) of the rotor axis from the outer body axis. In generating the epitrochoidal inner surface contour of the outer body, the theoretical eccentricity (e) can be replaced by a new value ($e+C$ or e') where C is a constant equivalent to the bearing clearances and any necessary additional deflection allowances.

If the radius of the rotor has a design value of R, in theory the K factor of the epitrochoidal inner surface to accommodate the rotor having a radius of R will be

$$K=\frac{R}{e}$$

but since the bearing clearances and deflection allowances must be taken into account, the epitrochoidal inner surface is modified to provide a new epitrochoid which has a K factor K' such that

$$K'=\frac{R}{e+C}=\frac{R}{e'}$$

The factor K' is sufficiently smaller than the value of K called for by the design radius R and design eccentricity e of the epitrochoidal inner surface to restore balance to the equation in spite of the growth of the actual eccentricity to a new value e' or $e+C$. With the new epitrochoid having a K factor K' no radial movement of the apex seals is required to compensate for shifting of the rotor due to centrifugal forces, bearing clearances and deflection allowances, because the shape of the epitrochoid itself with a K factor of K' compensates for the shifting.

2. ROTOR ECCENTRICITY CORRECTION

Since movement of the rotor from its geometric center due to the influence of centrifugal forces is equivalent to an increase in the eccentricity (e) of the rotor axis from the outer body axis, to a large value ($e+C$ or e'), a second means for overcoming the effects of the centrifugal forces may be achieved by reducing the enlarged eccentricity (e') by replacing the design eccentricity e with a new value e'', where $e''=e-C$ and where C is equivalent to the rotor and shaft bearing clearances and any necessary additional deflection allowances and has a magnitude such that $e'-C=e$, where e is the original or design eccentricity and e' is the enlarged or uncorrected operating eccentricity. Although the R of the rotor remains the same, the effective eccentricity e' is reduced under this method to the desired value e by subtracting the amount C from the design eccentricity e during manufacture. With this resultant reduced eccentricity e'' of the rotor, no radial movement of the apex seals is required to compensate for shifting of the rotor due to centrifugal forces because of bearing clearances and other centrifugal deflection allowances.

In a sense, reducing the design eccentricity e of the rotor by an amount C provides a compensation for the shift or deflection of the rotor from its geometric center which is the inverse of the compensation for the same problem which is achieved by reducing the K factor or value of K of the epitrochoidal inner surface of the outer body to a new value K' which restores balance to the equation

$$K=\frac{R}{e}$$

when e has been increased by rotor shift to e' such that

$$K'=\frac{R}{e'}$$

as described under 1 above.

3. COMPENSATION OF RADIUS OF CURVATURE OF CONTACTING SURFACE OF TIPS OF SEALING MEMBERS OR APEX SEAL TIP COMPENSATION

By increasing the radius of the curvature of the contacting surface of the tips of the sealing members from their design value or the value of r which is used to create an outer curve parallel to a true epitrochoid and which forms the inner surface, the forced movement of the apex sealing members can be materially reduced in the region of travel on the inner surface in the regions where large pressure differences exist across the sealing members or apex seals. These large differential pressures occur substantially between the point where the angle formed between the center line of the seal and a perpendicular to the tangent to the curve of the inner surface at its point of contact with the sealing member reaches a maximum and the point where the major axis of the epitrochoid intersects the inner surface.

This method of compensating for shifts of the rotor from its geometric center under the influence of centrifugal forces not only decreases apex seal movement relative to the rotor in the general region of high differential gas pressure across the seals, but also provides an increased radius of contact between the seal and the inner surface which beneficially decreases contact stresses acting against the peripheral wall of the outer body.

Optionally, an additional correction or modification of the resultant epitrochoid can be made after any of the above-described three ways of achieving the compensating means have been applied. This additional correction or modification compensates for any distortion of the inner surface of the outer body due to thermal gradients when the engine is in operation, and the final inner surface contour which results provides a mechanical shape that minimizes movement of the apex seals relative to the rotor.

It is a fundamental object of the present invention to provide means for minimizing relative movement between the apex seal and the rotor during revolution of the rotor within the outer body.

It is another object of this invention to minimize relative movement between the apex seal and the rotor by modifying the shape of the inner surface contour to compensate for movement of the rotor from its geometric center due to centrifugal forces and by further modifying its shape to compensate for distortion of the inner surface due to thermal gradients.

It is another object of this invention to minimize relative movement between the apex seal and the rotor by reducing the eccentricity of the rotor by a value equivalent

to the bearing clearances and any additional deflection allowances.

Another object of this invention is to reduce contact stresses of the apex seal against the inner surface of the outer body by increasing the radius of contact of the seal with the inner surface of the outer body.

A further object of this invention is to improve the sealing characteristics of the sealing members by simple and inexpensive change in the radius of curvature of the contacting surface of the tips of the sealing members.

A still further object of the instant invention is to greatly enhance the sealing properties of the engine through a combination of relatively small but important changes in the construction of the engine.

From the foregoing it is apparent that the instant invention is of outstanding importance in providing a useful rotating combustion engine by minimizing or practically eliminating relative movement of the apex seals in the rotor during revolution of the rotor inside the outer body when differential gas pressure across the seals is high. Obviously, the less the apex seals are required to move in maintaining sealing contact with the inner surface of the outer body, the more uniform and adequate the sealing will be.

Broadly described, the instant invention comprises means for minimizing movement of the apex seals relative to the rotor during rotation of the rotor within the outer body. This means comprises modification of the shape of the inner surface contour to compensate for movement of the rotor from its geometric center by centrifugal forces during rotation, modifications in the shape of the inner surface contour to compensate for thermal gradients, reduction of the eccentricity of the rotor to compensate for movement of the rotor from its geometric center and other deflections, and increasing the radius of curvature of the contacting surface tips of the apex seals from the value used to obtain an outer curve parallel to the basic epitrochoidal curve used in designing the shape of the outer body cavity.

Additional objects and advantages of the invention will be set forth in part in the description which follows and in part will be obvious from the description, or may be learned by practice of the invention, the objects and advantages being realized and attained by means of the instrumentalities and combinations particularly pointed out in the appended claims.

The invention consists in the novel parts, constructions, arrangements, combinations, and improvements shown and described.

The accompanying drawings which are incorporated in and constitute a part of the specification, illustrate one embodiment of the invention and, together with the description, serve to explain the principles of the invention.

In the accompanying drawings illustrating the mechanical aspects of the present inention, it is believed that the showing of the fundamental construction, functions, originality and advantages of the invention may be more easily understood when certain details of practical construction are omitted, where these details form no part of the claimable invention, are well-known to those skilled in the art, and could be incorporated in the present invention by any skilled workman. These details may consist of means for lubrication, such as, oil cups, grooves, reservoirs, seals, wipers, and O-rings; means for reduction of friction, such as, bushings, ball bearings, and roller bearings; means for sealing off various spaces or areas to confine fluid pressures to their functional locale, such as, packing, packing glands, O-rings, and gaskets; constructional details of fluid conducting means, such as, tube or pipe joints, unions, and elbows including supporting and securing means; and such other comparable means and devices that may be omitted for the sake of clarity.

Of the drawings:

FIG. 1 is a sectional view of the mechanism showing the outer body in section and the rotor positioned for rotation within the outer body;

FIG. 2 is a central vertical section of the principal portion of the engine showing the rotor within the outer body;

FIG. 3 is a diagrammatic view of the mechanism showing the rotor with one apex seal positioned within the outer body and also a showing of the rotor (by broken line and dot-dash) in two alternate positions. A true epitrochoid is shown by a dot-dash curve which has the center of curvature of the apex seal tip as its generating point; the inner surface contour, which is basically an epitrochoid, appears as a solid line curve and is generated by adding an amount equal to the apex seal radius normal to the true epitrochoid over its entire length;

FIG. 4 is a diagrammatic view showing the substantially epitrochoidal inner surface contour of FIG. 3 in a broken line curve, and the inner surface contour after modification to minimize seal movements which would otherwise result, for example, from movements of the rotor from its geometric center due to centrifugal forces, is shown in a solid line. The rotor is shown within the outer body in two positions. In one position one of its apex portions is aligned with the major axis of the outer body and in the other position this apex portion is aligned with its minor axis;

FIG. 5 is a schematic view of the substantially epitrochoidal inner surface of the outer body showing the location of the intake and exhaust ports and depicting the amount of heat being rejected per unit area through the inner surface into the outer body at various points along the inner surface by a broken line curve

$$\left(\frac{O}{A}\right)$$

the relative amount of temperature difference around the inner surface using the coolest spot as a reference point by dot-dash phantom curce (ΔT) and the amount of relative thermal growth around the inner surface by a double dot-dash phantom curve (ΔD).

FIG. 6 is a diagrammatic view of the mechanism showing how movement of the rotor from its geometric center under centrifugal forces during rotation may be compensated for by reducing the eccentricity of the rotor:

FIG. 7 is a diagrammatic view of the profile of the inner surface of the outer body on a plane transverse to the outer body axis. This profile is basically an epitrochoid; also shown in broken line is an epitrochoidal inner surface which is modified to minimize movement of the sealing members relative to the rotor; the intake and exhaust ports are shown schematically, and an outer curve shows the differential pressure across an apex seal at all points of location of the apex seal in its travel around the epitrochoid. An apex is also shown schematically in three different positions of its travel around the epitrochoid.

FIG. 8 is a fragmentary view of an apex seal showing the contacting surface of the tip of the seal after modification in solid line and before modification in broken line.

It is to be understood that both the foregoing general description and the following detailed description are exemplary and explanatory but are not restrictive of the invention.

Reference will now be made in detail to the present preferred embodiment of the invention, an example of which is illustrated in the accompanying drawings in which FIG. 1 shows a generally triangular rotor **10** having arcuate sides eccentrically supported for rotation within an outer body **12** on an axis **14** which is eccentric to and parallel to the axis **16** of the outer body.

As embodied, the profile of the curved inner surface **18** of the outer body **12** has a geometric shape which has basically the form of an epitrochoid with two arches lobe-defining portions, or lobes. An intake port **20** is arranged to communicate with one lobe of the epitrochoidal inner surface **18**, and an exhaust port **22** is arranged to com-

municate with the other lobe. The center of the epitrochoidal inner surface 18 has an axis 16 forming the axis of the outer body 12. There are two points of least radius from the center 16 of the epitrochoidal inner surface 18. A line connecting these two points of least radius and passing through the center of the epitrochoid is designated the minor axis 24 of the epitrochoid. Similarly, the epitrochoid has two points of greatest radius, and a line connecting these two points and passing through the center of the epitrochoid is designated the major axis 26 of the epitrochoid.

The epitrochoidal inner surface over a substantial distance adjacent to its major axis describes a concave portion, and describes a convex portion over a lesser distance adjacent its minor axis, as is shown in FIG. 1.

It is apparent that the minor axis 24 divides the epitrochoid into two halves. For convenience, the half or lobe which communicates with the exhaust port may be designated the exhaust lobe and the half or lobe which communicates with the intake port may be designated the intake lobe.

The general shape of the rotor 10 approaches as a limit the shape of the inner envelope of the epitrochoid defined by the inner surface 18 of the outer body 12. In practice, the rotor shape is made less than the theoretical maximum of the inner envelope to allow for accumulation of tolerances, bearing clearances, and deflections.

As embodied, the shaft 28 of the engine is coaxial with the axis 16 of the epitrochoidal inner surface of the outer body. An eccentric 30 is mounted on the shaft 28 and is coaxial with the rotor axis 14. The rotor 10 is rotatably mounted on the eccentric 30, and suitable counterweights (not shown) are keyed to portions of the shaft 28 and serve to counterbalance the eccentric 30 and the rotor 10 when the engine is in operation.

The outer body 12 proper comprises two end walls 32 and 34 and a peripheral wall 36 interconnecting the end walls. A spark plug 38 is mounted in the peripheral wall 36 and is so disposed that its electrodes communicate with the working chambers 58 formed in the cavity of the outer body 12 between the inner surface 18 and the outer peripheral surface of the rotor 10.

The generally triangular rotor 10 has three apex portions 42, two end faces 44 and 46 which are disposed adjacent to the end walls 32 and 34 of the outer body 12 (as may be seen in FIG. 2), and three outer peripheral surfaces or working faces 48 which extend between the end faces 44 and 46 of the rotor 10. Each apex portion 42 of the rotor is provided with a slot which extends in an axial direction from one end face 44 to the other 46. An apex sealing member 52 is mounted in each slot 50 and is spring loaded radially outwardly to ensure its continuous gas-sealing engagement with the inner surface 18 of the outer body 12 while the engine is in operation.

An internally-toothed gear 54 is rigidly secured to the rotor 10 at an opening in one of its end faces 44, and an externally-toothed gear or pinion 56 is in turn rigidly secured to one end wall 32 of the outer body 12. The teeth of the externally-toothed gear 56 are in mesh with the teeth of the internally-toothed gear 54. As embodied, the ratio of the intermeshing teeth between the rotor gear 54 and the outer body gear 56 is 3:2, so that for every revolution of the rotor about its own axis 14 the shaft 28 completes three revolutions in the same direction about its axis 16.

In operation, thus, the rotor 10 performs a planetary rotary movement in a counterclockwise direction with respect to the outer body 12, as may be seen in FIG. 1. As the rotor 10 follows its eccentric path of revolution within the outer body 12, the three working chambers 58 of the engine vary in volume, and during each complete revolution of the rotor each working chamber 58 will undergo the four phases of the engine cycle: intake, compression, expansion, and exhaust. The working chambers 58 are isolated from each other by the apex sealing members 52 which are maintained in continuous gas-sealing engagement with the inner surface 18 of the outer body 12 as the rotor revolves.

The working faces 48 of the rotor 10 have channels 60 which are cut into a substantial area of the working faces 48. These channels 60 permit the hot and burning combustion gases to pass freely from one lobe-defining portion of the outer body 12 to the other, or from the intake lobe to the exhaust lobe and vice versa, when a rotor working face 48 is in or near the top dead-center compression position, as shown in FIG. 1. The size and depth of the channel 60 may effectively determine the compression ratio for the engine.

In FIG. 3 a rotor 10 is shown mounted within an epitrochoid 62. The rotor is shown in three different positions 63, 64 and 65. The epitrochoid 62 is a true epitrochoid, and it may be seen in FIG. 3 that the apex portions of the rotor 10 in all three positions illustrated, 63, 64, and 65, are in point contact, or coincident at their points or vertexes, with the epitrochoid 62. Theoretically, an actual construction using this geometric arrangement would be feasible and would produce an operating engine; however, as a practical matter, it would be difficult or impossible to effectively seal the working chambers of such an engine from each other at the sharply pointed vertexes of the apex portions 42 of the rotor 10. A point contact between the rotor apex portions 42 and the true epitrochoid 62 will thus not yield an effective seal. Accordingly, an approximate epitrochoid 66, or curve having basically the form of an epitrochoid, is generated and forms the actual inner surface 18 of the outer body 12. This approximate epitrochoid 66 is generated by the addition of a small equal increment of length normal to the true epitrochoid 62 over its entire length to form, in effect, an outer curve 66 parallel to the true epitrochoid 62 over its entire length. The use of an outer parallel curve 66 with a true epitrochoid 62 is described in copending application Serial No. 638,127, filed February 4, 1957, now Patent No. 2,988,008.

One sealing member 52 is shown for each rotor position 63, 64, and 65 of FIG. 3. Each sealing member 52 is curved at its operative end or tip. This curve is generated by radius of curvature r for the contacting surface of the apex seal tip. The true epitrochoid 62 is generated by the center of this apex seal radius r as a generating point as the rotor revolves. Also, as shown in FIG. 3, the equal increment that has been added to the true epitrochoid 62 to generate the approximate epitrochoidal inner surface 66 is equal to the length of the apex seal radius r or 95′ (see FIGS. 3 and 8). Accordingly, as the rotor assumes different positions within the epitrochoid 66, it is apparent from FIG. 3 that because of the circular curved contact surface of the apex seal 52 and the manner in which the curve 66 is generated no radial movement of the apex seal 52 is required to maintain it in contact with the approximate epitrochoid 66 regardless of the position of the rotor, and that a line connecting the center of the apex seal radius with the point of contact will be normal to the curves 66 and 62 and will intersect them at the point of contact and at the center of curvature of the apex seal tip radius respectively, as shown by the lines N and N′ in FIG. 3.

From the foregoing, it is apparent that if the inner surface 18 of the outer body 12 is designed to conform to the approximate epitrochoid 66 of FIG. 3, and if an apex seal having a radius r is used, the difficulties presented by a point contact of the apex portions 42 with the true epitrochoid 62 of FIG. 3, can be obviated. Even if the curve of the inner surface 18 is theretically perfect, however, the apex sealing member 52 must nevertheless move to a slight extent with respect to the rotor because of centrifugal fields, bearing clearances, shaft deflections, and thermal distortions.

The eccentricity e of the rotor axis 14 from the outer

body axis **16** is shown in FIG. 3, and the circular path **15** described by the rotor center **14** as it revolves within the outer body **12** is shown in broken line.

Outer Body Compensation

FIG. 4 illustrates one means of achieving the invention and is briefly described as method 1 above. In general, FIG. 4 shows how the epitrochoid **62** (FIG. 3) may have its contour modified to allow for movement of the rotor from its geometric center during operation of the engine due to bearing clearances and deflections. The epitrochoid **62** of FIG. 3 is shown in FIG. 4 by a broken line curve, also designated **62**, the modified contour of the epitrochoidal inner surface **18** is shown in FIG. 4 as a solid line curve **70**. The theoretical or geometric eccentricity *e* of the rotor center **14** from the outer body center **16** would describe a solid line circle **72** as shown in FIG. 4 if the rotor did not move from its geometric center during operation of the engine. Because of bearing clearances between the bearing bore **74** of the rotor and the eccentric **30** upon which the rotor is mounted, other bearing clearances in the engine, and deflections, the effect of centrifugal force tends to cause the actual eccentricity *e'* to exceed the theoretical eccentricity *e* so that the geometric center and center of gravity CG of the rotor describes the broken line circle **76** rather than the solid line circle **72** as rotor **10** revolves within the outer body **12**.

In FIG. 4, the curve **70**, like the curve **62**, is a true epitrochoid, and the seals **52** must have pointed apexes to maintain contact without movement relative to the rotor, as explained above in the description of FIG. 3. In an actual rotary mechanism, however, the apex seals would be provided with a rounded apex having a radius of curvature of its contacting surface equal to the radius *r* in FIG. 3. Therefore, as explained in the description of FIG. 3, in a working rotary mechanism with rounded apex seals, the profile of the inner surface of the outer body would be a curve similar to the curve **66** of FIG. 3, or a curve parallel to the curve **70** and spaced outward from it by the distance *r*. No movement of the rounded apex seals relative to the rotor is required to maintain contact with such an outer parallel curve. For simplicity in describing the concept of outer body compensation as depicted in FIG. 4, only the true epitrochoids **62** and **70** are shown. It should be understood, however, that in an actual rotary mechanism the apex seals **52** would be rounded, and the profile of the inner surface would be a curve spaced outward from but parallel to the curve **70**.

In FIG. 4 the rotor **10** is shown in two positions: a solid line position **78** in which one of its apex seals **52** is aligned with the major axis **26** of the modified epitrochoid **70** and a broken line position **80** in which one of its apex seals **52** is in alignment with the minor axis **24** of the modified or new epitrochoid **70**. The effective bearing bore **75** of the rotor in the solid line position **78** is shown in solid line as is also the eccentric **77** for position **78**. The effective bearing bore **79** for rotor position **80** is shown in broken line as is also the eccentric **81** for position **80**. The centrifugal force acting on the rotor in position **78** and its direction is represented by the arrow **F1**. Similarly, the centrifugal force acting on the rotor in position **80** is represented by the arrow **F2**.

As can be seen from FIG. 4, the centrifugal force acting on the rotor tends to cause all of the bearing clearances to appear on one side of the respective eccentrics **77** and **81** and bearing surfaces **75** and **79**. The total diametral bearing clearance (2C) between the eccentric and the rotor accumulates in the direction along which the centrifugal forces **F1** and **F2** act, as shown by FIG. 4. In FIG. 4, F1 acts along the major axis **26**, and F2 acts along the minor axis **24**, because the centers of gravity CG of the rotors **78** and **80** are coincident with the rotor axes **14**, and the axes of the rotors **78** and **80** (in their positions as shown in FIG. 4) intersect the major axis **26** and minor axis respectively of the epitrochoids **62** and **70**. Accordingly, when the apex seals **52** are aligned with the major axis **26** and the minor axis **24** respectively, the centrifugal force F1 and F2 are also acting in a direction along these same axes.

From a study of FIG. 4, it is apparent that the accumulation of total diametral bearing clearance (2C) in a radially outward direction from the geometric center **16** of the epitrochoid **62** when the apex portion **82** of the rotor **78** is aligned with the major axis **26** would tend to force the apex seal **52** outside of the epitrochoid **62**, if the seal were not free to move relative to the rotor. Further, when the apex portion **84** of the rotor **80** is aligned with the minor axis **24**, it is apparent from FIG. 4, that the apex seal **52** would tend to be pulled in, away from the epitrochoid **62**, if the apex seal were not forced by appropriate means to move radially outward relative to the rotor **80**.

In view of the foregoing, it is apparent that movement of the apex seals relative to the rotor can be minimized by modifying the contour of the epitrochoid **62** and causing it to assume the contour of a modified or new epitrochoid **70**.

It will be seen from FIG. 4, that the maximum correction is needed along the major axis **26** and minor axis **24**. At each end of the major axis **26** it will be necessary to add to the major axis of the epitrochoid **62** an increment (*c*) equal to one-half the total bearing clearance, and at each end of the minor axis **24** it will be necessary to subtract from the length of the minor axis of the epitrochoid **62** the increment (*c*) equal to one-half the total bearing clearance. Appropriate intermediate amounts must be added to or subtracted from the epitrochoid **62** over its entire length to yield the modified epitrochoid **70** which will compensate for the shift of the geometric center of the rotor under centrifugal force due to bearing clearances. It is obvious that a similar or further change can be made to allow for any shifts of the rotor in the same direction from its geometric center due to clearances in the bearings for the rotor shaft **28** and due to deflections other than bearing clearances such as the slight deflection shift caused by the inherent elasticity of the materials from which the engine is constructed.

The new epitrochoid **70** can be created from the original or design epitrochoid **62** by changing the K factor of the rotating combustion engine. The K factor has previously been described as the value K in the equation

$$K=\frac{R}{e}$$

where R is the radius or the distance from the rotor axis **14** to the point of contact of the rotor apex portion with the epitrochoidal inner surface (see FIGS. 1 and 2), and where *e* is the eccentricity of the rotor axis **14** from the central axis **16** of the epitrochoidal inner surface of the outer body. From the equation

$$K=\frac{R}{e}$$

it is apparent that if R remains constant and *e* is increased, the value of K must become smaller if the equation is to remain in balance.

The shift of the rotor from its geometric center by an amount C equal to one-half the total bearing clearance is the equivalent of increasing the design eccentricity *e* of the engine by the amount C. The theoretical eccentricity *e* can thus be replaced by a new value $e+C$ or e', where C is equal to the radial bearing clearance and any necessary additional deflection allowances. The new epitrochoid **70** is arrived at by designing an epitrochoid which will have a K factor K' such that

$$K'=\frac{R}{e+C}=\frac{R}{e'}$$

The K factor of the new epitrochoid **70** will thus be

smaller than the K factor of the design epitrochoid 62, or the K factor which is called for by the design radius R of the rotor and the design eccentricity *e* of the engine.

Radial movement of the apex seals which would normally be required to compensate for shifting of the rotor due to centrifugal forces, bearing clearances and deflection allowances may be virtually eliminated by the use of the new epitrochoid 70 with a K factor K'. An important feature of this invention is the discovery that a new epitrochoid can be created which will compensate for all centrifugal forces, shifts in rotor position by the relatively simple expedient of decreasing the design K factor, or K, to a new K factor K' which can be mathematically predetermined from the equation

$$K'=\frac{R}{e+C}=\frac{R}{e'}$$

The curve

$$\frac{Q}{A}$$

in FIG. 5 shows the amount of heat being rejected per unit area through the inner surface 18 of the peripheral wall of the outer body at each point along the inner surface 18 of the epitrochoid 62 (or the epitrochoid 70).

The relative amount of temperature difference around the inner surface 18 is shown in FIG. 5 by a dot dash phantom curve (ΔT) which uses the coolest spot on the inner surface as a reference point and a double dot dash phantom curve (ΔD) shows the relative amount of thermal growth around the inner surface 18. The curve ΔD, however, is greatly exaggerated for clarity, and the location of the intake port 20 and exhaust port 22 are shown schematically in FIG. 5.

If it were possible to keep the epitrochoidal inner surface 18 at a fairly uniform temperature, there would be no need to modify the epitrochoid 70 of FIG. 4 to compensate for distortion of the epitrochoidal inner surface caused by thermal gradients. In practice, however, since a perfect cooling system is unattainable, there will always be some distortion of the epitrochoid 70 due to thermal gradients in the outer body 12 and this distortion is represented by the curve ΔD in FIG. 5. It is possible, however, to modify the epitrochoid 70 during manufacture so that, although the inner surface will not be a true epitrochoid when the mechanism is cold, thermal distortions will cause the inner surface to assume the shape of a true epitrochoid at running temperatures. This additional modification of the epitrochoidal inner surface can thus virtually eliminate the effect of thermal distortions at operating temperatures.

After the epitrochoid 70 has been compensated as described above it should be still further modified by adding an equal increment outside of and normal to it over its entire length to form, in effect, an outer curve parallel to the inner curve over its entire length in the same manner as previously described in the discussion of FIG. 3.

Rotor Eccentricity Correction

A further means by which some of the advantages of the present invention can be achieved is shown in FIG. 6 and was briefly described as method 2 above. In the foregoing description relating to FIG. 4 and the means of achieving the results of the invention by creation of a new epitrochoid 70, it was explained that centrifugal forces acting on the rotor during its rotation within the outer body 12 cause its center or axis to be shifted radially outward (with respect to the axis 16 of the outer body 12) from its geometric center. The amount of the shift is approximately equivalent to one-half of the diametral, or the radial bearing clearance between the rotor bearing surface and the eccentric. There is also, however, a slight deflection shift which must be added to the bearing clearance shift, and this deflection shift is caused by the inherent elasticity which is present in the materials from the engine is constructed. The total shift is then a combination of a bearing clearance shift plus the deflection shift and is designated C.

In FIG. 6, for purposes of clarifying the explanation of this aspect of the invention, two positions of the rotor are indicated by a showing of their apex portions only. Also, to permit enlargement of the scale of FIG. 6, instead of showing a complete epitrochoidal inner surface, a fragmentary portion of the inner surface comprising slightly more than one quadrant is shown, but this showing is sufficient for an understanding of the explanation accompanying FIG. 6.

The curve representing the epitrochoidal inner surface in FIG. 6 corresponds to the outer parallel curve or approximate epitrochoid 66 of FIG. 2. Although strictly speaking the approximate epitrochoid 66 is not a true epitrochoid, such as the epitrochoid 62 of FIG. 3, it differs only slightly from a true epitrochoid and, for practical purposes, in the ensuing explanation may be considered to be a true epitrochoid. It will be remembered, that in the description accompanying FIG. 3 it was explained that the outer parallel curve for approximate epitrochoid 66 was created from the true epitrochoid 62 to permit the use of apex seals with the rotor which could have curved tips at their points of contact with the epitrochoidal inner surface 18, instead of pointed tips having contact with the inner surface 18.

When the inner surface of the outer body is a parallel curve, the distance from the center of the rotor to the parallel curve is only slightly greater than the distance to the true epitrochoid on which the parallel curve is based. The difference between these two distances is quite small and has been highly exaggerated in FIG. 3. Accordingly, since the inner surface 18 of the outer body 12 is a parallel curve to a true epitrochoid, as shown in FIG. 3, the K factor of the epitrochoid on which the parallel curve inner surface is based is substantially equal to

$$\frac{R}{e}$$

where R is the distance from the center of the rotor to the inner surface and *e* is the actual eccentricity of the rotor relative to the axis of the outer body.

As used in this application:

(1) In a true epitrochoid

$$K=\frac{R}{e}$$

(2) When the inner surface of the outer body is a parallel curve that is only slightly outwardly spaced from a true epitrochoid, the K factor of the epitrochoid on which the parallel curve is based may be said to be equal to

$$\frac{R}{e}$$

for all practical purposes, where R is the distance from the center of the rotor to the inner surface of the outer body.

Also, in this application no distinction will be made between the K factor of a true epitrochoid that is determined by the parameters R and *e* and the K factor (or quasi-K factor) of a parallel curve (66 in FIG. 3) that is based on a true epitrochoid. In strict usage, the parallel curve 66 would not have a K factor per se, since it is not a true epitrochoid. The parallel curve 66, however, differs so slightly from a true epitrochoid that for all practical purposes it may be considered to have a K factor, and this K factor may be stated to be equal to

$$\frac{R}{e}$$

where R is taken as the distance from the center of the

rotor to a point of sealing engagement at one of its apex portions with the inner surface of the outer body.

Again, when a parallel curve is used the radius (R) of the rotor to the point of contact with the inner surface will be slightly greater than the radius (R) of a rotor used with the true epitrochoid upon which the parallel curve is based, because, by definition, the parallel curve is spaced outwardly from the true epitrochoid by the increment r at all points.

Accordingly, the radius of the rotor, when a parallel curve is used, would be strictly defined as the R of a rotor for a true epitrochoid plus the increment r. Thus, if R is taken as the radius of the rotor for a true epitrochoid, and R′ is taken as the radius of the rotor to be used with the parallel curve based on the true epitrochoid, the following relationship exists: $R'=R+r$. But the K factor of the parallel curve may still be represented by

$$K=\frac{R}{e}$$

because $R'=R+r$ and r is quite small as compared with R. Thus

$$K=\frac{R}{e}$$

and for all practical purposes

$$\frac{R}{e}=\frac{R'}{e}$$

Accordingly, in the claims, no distinction has been made between either R or R′ or the K factor of the true epitrochoid

$$\left(K=\frac{R}{e}\right)$$

and the quasi-K factor of the parallel curve

$$\left(K=\frac{R'}{e}\right)$$

As in the description of FIG. 4, so also in the description of FIG. 6, the rotor in position **78** has an apex portion in position **82** along the major axis **26** of the epitrochoid **66**, and when the rotor is in position **80**, it has its apex portion in position **84** along the minor axis **24** of the epitrochoid **66**.

The radially outward shift of the rotor from its geometric center due to the influence of centrifugal forces is equivalent to an increase in the eccentricity e of the rotor axis from the outer body axis. The shift of the rotor causes the design eccentricity e to increase to a larger value which gives the rotary mechanism an actual or effective eccentricity with a value of $(e+C)$ or e', where C is equivalent to the rotor and shaft bearing clearances and any necessary additional deflection allowances.

With this second principal means of achieving the invention, the enlarged effective eccentricity e' may be reduced or compensated for by a reduction in the original or design eccentricity e of the rotor, although the R of the rotor remains the same. The desired result is achieved by subtracting an amount equivalent to C from the design eccentricity e during manufacture of the rotary mechanism. With this resultant reduced design eccentricity $(e-C)$ of the rotor, no radial movement of the apex seals will be necessary to compensate for shifting of the rotor due to centrifugal forces because of bearing clearances and other centrifugal deflection allowances.

In a sense, reducing the design eccentricity e of the rotor by the amount C provides a compensation for the shift or deflection of the rotor from its geometric center which is the inverse of the compensation for the same problem which is achieved by reducing the value of K or the K factor of the epitrochoidal inner surface of the outer body to a new value K′ to restore balance to the equation

$$K=\frac{R}{e}$$

when e has been increased by rotor shift to e' such that

$$K'=\frac{R}{e'}$$

Compensation for rotor shift through reduction of the K factor was described previously in the explanation of FIG. 4 above.

In both "Outer Body Compensation" (Means 1, above) and "Rotor Eccentricity Correction" (Means 2, above), the actual or running eccentricity of the engine (which is the eccentricity that determines the shape of the epitrochoidal inner surface **18** of the outer body **12**) is greater than the geometric eccentricity of the rotor relative to the outer body axis.

In FIG. 6 the uncorrected or basic eccentricity e' is designated by the dot-dash phantom line **86**, and the corrected eccentricity $(e'-C)$ or e is shown by the solid line **88**, and the design eccentricity $(e-C)$ or e' is shown by the broken line **87**. The effective bearing bore for the rotor in position **78** is shown by the dot-dash phantom line **75′** in its uncorrected position, and the position of the eccentric mounted within it is designated by the dot-dash phantom line **77′**. The position of the bearing bore for the rotor in position **78** after correction is shown by the solid line **75** and the position of the eccentric is shown by the solid line **77**.

Similarly, the effective bearing bore for the rotor in position **80**, before the eccentricity correction has been made, is designated by the dot-dash phantom line **81′**, and, the position of the eccentric within this bearing bore is shown by the dot-dash phantom line **79′**. The position of the bearing bore for the rotor in position **80** after correction is shown by the solid line **81**, and the position of the eccentric after correction is shown by the solid line **81**, and the position of the eccentric after correction is shown by the solid line **79**.

In FIG. 6, the position of the axis and center of gravity of the rotor in position **78**, before the eccentricity correction has been made, is designated **90′**, and its position after the eccentricity correction is designated **90**. Similarly, the position of the axis and center of gravity of the rotor in position **80** before correction is designated **92′**, and its position after correction is designated **92**.

From a perusal of FIG. 6 and a study of the way in which the path of travel of the rotor axis and center of gravity is shifted closer to the center of the epitrochoid **66** by the eccentricity correction, and from a study of the manner in which the bearing clearances are shifted responsive to the shifts produced by the eccentricity correction, it will be apparent that the apex portion of the rotor in position **78** along the major axis **26** will be pulled in and away from the epitrochoid **66**, and the apex portion of the rotor in position **80** along the minor axis **24** will be pushed out and toward the epitrochoid **66**.

When the epitrochoidal inner surface is provided with a design eccentricity of $(e-C)$ or e'', the actual or effective eccentricity of the engine will reach the desired value e when the engine is running through the addition of the amount C to the design eccentricity $(e-C)$ or e'' and will thus provide exactly the desired effective eccentricity e $(e-C+C=e)$ or $(e''+C=e)$. With this relatively simple change of the design eccentricity during manufacture, it is unnecessary to compensate for rotor shift by modifying the contour of the epitrochoid **62** upon which the approximate epitrochoid **66** is based, through a change in its K factor. Of course, it may still be necessary to modify the contour of the epitrochoid **66** when the method of FIG. 6, is used, to allow for distortions due to thermal gradients when the engine is at operating temper-

ature. Such a modification was discussed in the explanation of FIG. 5 above.

In FIG. 6, the apex portion in position **82** is shown in solid line in the position it would occupy after the eccentricity correction has been made. The position of the same apex portion in postion **82'** is shown in a dot-dash phantom line in the position it would occupy without the eccentricity correction. It should be observed that the apex portion in position **82'** is closer to the epitrochoid **66** by the amount C.

The apex portion in position **84** is shown in solid line along the minor axis **24** in the position it would occupy after the eccentricity correction has been made. This same apex portion is shown in position **84'** in dot-dash phantom line in the position it would occupy without the eccentricity correction. It can be seen from FIG. 6, that the apex portion in position **84'** is further away by the amount C from the epitrochoid **66** than the apex portion in position **84**. It should be noted also that the apex portion in position **84** is the same distance from the epitrochoid **66** as the apex portion in position **82**.

In comparing the relative positions of **82'** to **82** and **84'** to **84**, it should be apparent that the apex portion in moving from position **82'** to position **84'** will pull away from the epitrochoid **66** by an amount equal to 2C, if the eccentricity correction is not made. When the eccentricity correction is made, however, the rotor portion maintains the same distance away from the epitrochoid **66** in position **82** as in position **84**.

An apex seal **52** is shown in solid line, and in section for clarity, within the rotor apex portion in the solid line positions **82** and **84**. Similarly, an apex seal **52'** is shown in broken line in the rotor apex portion in the phantom line positions **82'** and **84'**. The broken line apex seal **52'** in the rotor posiiton **82'** coincides with the solid line apex seal **52**, and therefore, is not shown separate from the apex seal **52** in the drawing of FIG. 6. The two seals **52** and **52'** may thus be considered to be superposed one on the other in FIG. 6. Apex seal **52'** is shown relative to the rotor positions **82'** and **84'** as it would appear if it did not move relative to the rotor as the rotor moves from **82'** to **84'**.

From a study of FIG. 6, it will be apparent that, if effective sealing is to be maintained, without an eccentricity correction, the apex seal **52'** will have to move through a distance equal to 2C as the rotor moves from **82'** to **84'** to maintain contact with the epitrochoid **66**. On the other hand, it is unnecessary for the apex seal **52** (solid line) to move at all, as the rotor moves from positions **82** to **84**. Accordingly, reduction of the design eccentricity during manufacture by an amount equivalent to the rotor shift, offers an effective means of achieving the primary desideratum of this invention: to minimize movement of the apex seal relative to the rotor during operation of the rotary mechanism.

When a roller type bearing is used, the above analysis is precise because the minimum clearance in such a bearing is always located at the point of application of the load on the bearing. FIGS. 1 and 2, however, show a sleeve type bearing. When a sleeve-type bearing (such as illustrated) is used, it is known that the minimum bearing clearance is displaced angularly a small amount (in the direction of relative rotation of the rotor with respect to the outer body) from the point of application of the load (centrifugal force) on the rotor to the eccentric. This small angular displacement of the minimum bearing clearance, when a sleeve type bearing is used, may also be taken into account in determining the profile of the epitrochoidal inner surface of the outer body.

Apex Seal Tip Compensation

FIG. 7 shows the approximate epitrochoid **66** (corresponding to the curve **66** of FIG. 3) in solid line, and the intake port **20** and exhaust port **22** are diagrammatically or schematically located with respect to this epitrochoid. The variation of differential gas pressure across an apex seal, or the difference in gas pressure existing between two adjacent working chambers of the engine, at all points of travel of the apex seal along the entire length of the epitrochoid **62** which represents the inner surface **18** of the outer body **12** is shown by a curve of differential pressure which is designated **93**.

FIG. 7 discloses still a third means of achieving the invention. Briefly described, this third means of achieving the invention is through compensation of the curvature, or radius of curvature, of the contacting surface at the tip of the apex seal. For clarity of explanation, only one apex seal is shown in FIG. 7 but it is shown in three of the different positions that it assumes as it travels along the inner surface **18** within one quadrant of the epitrochoid **66**.

Also shown in broken line in FIG. 7 is the approximate epitrochoid **70'**. The epitrochoid **70'** corresponds to an outer parallel curve formed from the modified epitrochoid **70** of FIG. 4 in the same manner that the approximate epitrochoid **66** is formed from the epitrochoid **62** in FIG. 3. The apex seal in FIG. 7 is shown greatly exaggerated in size (particularly its width) compared to the epitrochoids **66** and **70'**. This size exaggeration is shown for clarity of explanation only. It will be recalled that the modified epitrochoid **70** of FIG. 4 has a K factor, such as K', which has a value smaller than the value K of the theoretically correct or design K factor of the epitrochoid **62**. The modified epitrochoid **70** with a K factor of K' compensates for shifts in position of the center of the rotor **10**, and this method of achieving the invention has been previously described in the explanation accompany-FIGURE 4. Because the epitrochoid **70'** in FIG. 8 is formed from the modified epitrochoid **70** with its K factor K', it compensates for shift of the rotor from its geometric center, and the tips of the apex seals of the rotor **10** in their paths of travel will exacly follow the inner surface **18** without the necessity for movement relative to the rotor when the epitrochoid has the shape of the modified epitrochoid **70'**.

FIG. 8 is a detail view of the tip of an apex seal. The usual or design configuration of the tip is shown in broken line and is designated **94'**. A modified tip or a tip with its radius of curvature compensated to achieve the results of the invention is shown in solid line and is designated **94**. It will be observed that the compensated tip **94** has a greater radius of curvature than the design tip **94'**. The radius of curvature of the compensated tip is designated **95** and the radius of curvature of the original or design tip is designated **95'**. This latter radius of curvature is equivalent to the value *r* shown in FIG. 3, which is the distance by which the outer parallel curve **66** is spaced from the true epitrochoid **62**.

As stated above, the apex seal in FIG. 7 is shown in section for clarity and is shown in three of the different positions that it assumes as it travels along the inner surface **18** within one quadrant of the epitroichoid **66**. As the apex seal proceeds in a counterclockwise direction from the major axis **26** to a position along the minor axis **24**, the three positions in turn are designated as **96**, **97** and **98**. The apex seal which is shown in solid line section in the three positions just described represents an apex seal having a radius of curvature of its tip compensated or modified as shown by **94** in FIG. 8.

Also shown in FIG. 7 in broken line is an apex seal with a design or unmodified radius of curvature, i.e., an apex seal having a radius of curvature of its tip which corresponds to the radius of curvature **95'** of the tip designated **94'** in FIG. 8. This apex seal is also shown in three different positions which it assumes in its path of travel along the inner surface **18** from the major axis **26** to the minor axis **24**. The tip of this latter seal, however, is shown with its path of travel along the modified broken line epitrochoid **70'** rather than along the unmodified solid line epitrochoid **66**. The three different

positions assumed by this latter apex seal are shown in broken line and are designated **96'**, **97'**, and **98'** to correspond to the three similar positions of the compensated apex seal in positions **96**, **97**, and **98**.

In FIG. 8, it may be seen that the tip of the uncompensated apex seal **94'** extends above the tip of the compensated seal **94** by a distance at the center of the seal which is equal to C. The amount C has previously been defined as an amount equivalent to the increase in eccentricity *e* caused by shift of the rotor center during operation from its theoretical or design position. The tip **94** of the apex seal with the compensated radius of curvature in position **96** in FIG. 7 will thus be in contact with the epitrochoid **66**, while the tip **94'** of the apex seal with the design radius of curvature in position **96'** will be in contact with the modified epitrochoid **70'**, because along the major axis **26** the modified epitrochoid **70'** extends outward beyond the epitrochoid **66** by an amount equivalent to C (see FIG. 4 and accompanying explanation).

It may also be observed that when the apex seal is in the position **96** or **96'** its center line coincides with a perpendicular to the tangent to the epitrochoidal curve at the point of contact, and the center line of the seal also coincides with the major axis **26** of the epitrochoids **66** and **70'**. As the seal travels along the epitrochoidal inner surface **18** toward the minor axis **24**, however, the angle which its center line forms with a perpendicular to a tangent to the curve at the point of contact increases from zero to a significant magnitude until it becomes a maximum and then decreases back to zero again when the seal reaches the minor axis. The point on the epitrochoidal curve at which the angle becomes a maximum or the point of maximum lean from the perpendicular approximately coincides with the point where the epitrochoid **70'** intersects the epitrochoid **66** and is represented by the positions **97** and **97'** of the apex seals, as may be seen in FIG. 7.

In FIG. 7 the point of contact of the tip of the apex seal with the epitrochoidal inner surface **18** when the seal is at its point of maximum lean from the perpendicular is desgnated **100** on the inner surface **18**. In FIG. 8 the corresponding point of contact on the tip of the apex seal when the seal is at its maximum angle of lean from the perpendicular is designated **101**. It may be seen in FIG. 8 that both the modified and unmodified or compensated and uncompensated tips of the apex seals coincide at point **101**.

It will be observed in FIG. 7 that with the seals in positions **97** and **97'**, they are in contact with the inner surfaces **18** of the epitrochoids **66** and **70'** at a point on their tips corresponding to point **101** of FIG. 8. Since the two epitrochoids **66** and **70'** also coincide at this point and since, by hypothesis, the broken line seal or unmodified seal in moving from position **96'** to position **97'** does not move relative to the rotor **10**, the modified or compensated seal in moving from position **96** to position **97** likewise does not move relative to the rotor, for the relative positions of the two seal tips as shown at **96** and **96'** of FIG. 7 (and also as shown in FIG. 8) has not changed in positions **97** and **97'**, i.e., the relative positions of the two seal tips has been preserved in movement of the seal tips from **96** and **96'** to **97** and **97'**.

By hypothesis, we know that the unmodified seal tip has not moved relative to the rotor in passing from **96'** to **97'**, therefore, since the relative positions of the two seal tips has not changed in moving from **96**, **96'** to **97**, **97'** the modified seal tip also has not moved relative to the rotor.

As shown in FIG. 8 the seal tips **94** and **94'** maintain a curvature for only a portion of the seal width on each side of the center line **102** of the seal. The remainder of the seal width is faced off by a straight angle cut **103**.

In accordance with the foregoing description and analysis, when the compensated or modified tip of the apex seal (**94** in FIG. 8) is used with the epitrochoid **66**, forced movement of the apex seal relative to the rotor will be minimized in the region of the epitrochoid quadrant between the position **96** in which the seal is aligned with the major axis **26** and the position of maximum seal angularity with a perpendicular to a tangent to the curve of the epitrochoid **62** at its point of contact with the seal tip. As can be seen from FIG. 7, this particular quadrant is on the side of the major axis **26** opposite from the side in which the intake port **20** and the exhaust port **22** are located, and this quadrant also coincides with the region in which differential pressures as represented by curve **93**, across an apex seal, or between adjacent working chambers of the engine, reach their highest values.

When the differential pressure across an apex seal is high, the gas pressure acting on one side of the seal tends to force the other side of the seal into close frictional engagement with the adjacent side of the slot **50** in which it is carried by the rotor (see FIG. 1), and because of this gas pressure force, it becomes difficult to move the seal relative to the slot **50** when the seal is within this high differential pressure region. Accordingly, this method of achieving the invention reduces seal movement relative to the rotor in the critical region of apex seal travel on the epitrochoid **62** to a minimum.

From a study of FIG. 7, it is apparent that if the uncompensated or unmodified apex seal tip is used, which has a radius of curvature approximately equivalent to *r*, or the distance by which the outer parallel curve **66** is separated from the epitrochoid **62** in FIG. 3, it will be necessary for the apex seal to move outward relative to the rotor **10** in travelling from position **96** to position **97**, and it will have to move relative to the rotor by an amount approximately equal to C. This movement of the apex seal relative to the rotor is required if the unmodified or uncompensated tip of the seal is used with the epitrochoid **66**.

Movement of the apex seal relative to the rotor in the critical region of travel where differential pressures are high can be almost completely eliminated by a relatively simple and inexpensive alteration of the shape or configuration of the tip of the apex seal from a configuration having a radius of curvature equal to *r* to one having a substantially greater radius of curvature.

In addition to minimizing movement of the apex seal in the critical region, apex seal tip compensation through the use of a substantially greater radius of curvature for the tip of the apex seal than the design radius of curvature provides a further benefit. There is an increase radius of seal contact with the inner surface of the outer body which results in decreased contact stresses, since the forces pushing the seal into contact with the inner surface are distributed over a larger area of the inner surface.

Both a compensated and an uncompensated apex seal tip are also shown in FIG. 7 in positions **98** and **98'**, respectively, along the minor axis **24** of the epitrochoid **66**. From a study of positions **98** and **98'** in FIG. 7 it is apparent that the uncompensated tip in position **98'** is closer to the inner surface of epitrochoid **62** along the minor axis **24** than is the compensated tip in position **98**. The compensated tip **98** will thus have to move further relative to the rotor than the uncompensated tip **98'** to effectively seal the engine in the region adjacent the minor axis **24**. Accordingly, it is important to provide a force sufficient to move the compensated tip radially outward from its position **98** along the minor axis **24** to a position which will place the tip in contact with the epitrochoid **66**.

To accomplish this movement an additional force F must be applied to the apex seal **98**. This force F is normally provided by means, such as a leaf spring, which will hold the seal radially outward against the inner surface of the epitrochoid **66**. In accordance with the invention, in the region of seal travel adjacent the minor axis **24**

between the points of maximum seal angularity with the perpendicular to the tangent to the curve of the epitrochoid **66** at the point of contact (see point **100** in FIG. 7) the differential gas pressure acting across the apex seal is relatively small, and it is insufficient in magnitude to seriously interfere with the radially outward movement of the seal by the supplementary force F.

Apex seal tip compensation, or the use of an apex seal tip with a radius of curvature considerably greater than the theoretical design value *r*, reduces apex seal movement relative to the rotor to minimal or insignificant proportions in the regions where differential gas pressure across the seal is high. Although apex seal tip compensation does not reduce movement of the apex seal in the region adjacent to the minor axis **24** to an insignificant amount (in fact, it actually increases the amount of movement which the apex seal must make to ensure engagement with the epitrochoid in this region), and although it does not obviate the necessity for substantial movement of the seal relative to the rotor in this region, the differential pressure across the seal is small enough, or sufficiently insignificant in magnitude, to permit the necessary movement of the seal in this region to be accomplished by the application of a small supplementary force F. Apex seal tip compensation thus eliminates the need for any substantial movement of the seal relative to the rotor in the region where this movement is difficult because of the differential gas forces that act on the seal to tremendously increase its inertia and resistance to movement.

As previously explained, the shape of the epitrochoidal inner surface of the outer body, or the shape of the epitrochoid itself is determined by the radio between the two variables R and *e*. The variables R and *e* thus determine the K factor or the value of K, that in turn is an index of the shape of the epitrochoid that is generated by a given R and *e*. These same parameters of the shape of the epitrochoid R, and *e*, also determine the magnitude of the angle ϕ (see FIG. 7) formed between the center line of the seal and the perpendicular **105** to the epitrochoid inner surface **18** at the point of contact of the seal with the inner surface where the lean of the seal way from the perpendicular **105** is greatest.

In other words, the greater the value of the angle ϕ the greater is the angle of the lean of the seal away from the perpendicular. With these conditions set forth the angle ϕ may be defined as follows:

$$\phi = \text{arc sin } \frac{3e}{R} = \text{arc sin } \frac{3}{K}$$

In the foregoing description of the invention three means for achieving a primary common object are disclosed. This object is to minimize the relative movement between the apex seal and the rotor during revolution of the rotor within the outer body and particularly to minimize the relative movement that is caused by the effects of centrifugal fields and the resultant deflections that are in turn caused by these fields. The invention provides three means by which the desired results may be achieved during manufacture of the rotary mechanism:

(*a*) Compensation or modification of the shape of the epitrochoidal inner surface of the outer body;

(*b*) Reduction of the design eccentricity of the mechanism; and

(*c*) Apex seal tip compensation.

The invention in its broader aspects is not limited to the specific mechansims shown and described but also includes within the scope of the accompanying claims any departures made from such mechanisms which do not depart from the principles of the invention and which do not sacrifice its chief advantages.

What is claimed is:

1. A rotary mechanism comprising a hollow outer body having an axis, axially-spaced end walls, and a peripheral wall interconnecting the end walls, the inner surface of said peripheral wall having basically the profile of a multi-lobed epitrochoid; a rotor mounted within the outer body for rotation relative to the outer body on an axis eccentric from and parallel to the axis of the outer body and for planetary motion about the outer body axis; the rotor having end faces disposed adjacent to the end walls and a plurality of circumferentially-spaced apex portions, one more in number than the number of said lobes with each apex portion substantially tracing said epitrochoidal inner surface during rotor rotation and being disposed in sealing engagement with the inner surface of the peripheral wall to form a plurality of working chambers between the rotor and the peripheral wall which vary in volume upon relative rotation of the rotor within the outer body; sealing members at each apex portion of the rotor for maintaining sealing engagement with the inner surface of the peripheral wall, the sealing members being mounted for radial movement in the rotor, and said epitrochoidal inner surface having a K factor which is smaller than that called for by the relation

$$K = \frac{R}{e}$$

where R is the distance from the center of the rotor to a point on the rotor tracing said epitrochoidal inner surface and *e* is the geometric eccentricity of the rotor axis for the axis of the outer body.

2. The invention as defined in claim 1, in which said value of K is equal to K′, where K′ is defined as:

$$K' = \frac{R}{e+C}$$

in which R and *e* are defined as in claim 1, and in which C is defined as the amount by which the actual eccentricity of the rotor axis from the outer body axis is increased above its geometric value during operation of the mechanism by displacement of the geometric center of the rotor radially outward from the outer body axis due to the action of centrifugal forces.

3. The invention as defined in claim 2, in which the profile of the inner surface is defined by a curve having basically the form of an epitrochoid but parallel to and outwardly spaced by an amount equal to *r* from the true epitrochoid having a value of K equal to K′, the value of *r* being substantially equal to the radius of curvature of the contacting surface of the tips of the sealing members.

4. The invention as defined in claim 1, in which the shape of the inner surface of the peripheral wall is modified to counteract radial movement of the sealing members which results from changes in the shape of the inner surface due to thermal gradients.

5. The invention as defined in claim 1, in which the profile of the inner surface is defined by a curve having basically the form of an epitrochoid but parallel to and outwardly spaced from the true epitrochoid having a smaller value of K than that obtained from the equation:

$$K = \frac{R}{e}$$

in which R and *e* are defined as in claim 1.

6. The invention as defined in claim 1, in which the epitrochoidal inner surface of the outer body has two lobes and the inner body has three apex portions.

7. A rotary mechanism comprising a hollow outer body having an axis, axially-spaced end walls, and a peripheral wall interconnecting the end walls, the inner surface of said peripheral wall having basically the profile of a multi-lobed epitrochoid; a rotor mounted within the outer body for rotation relative to the outer body on an axis eccentric from and parallel to the axis of the outer body and for planetary motion about the outer body axis; the rotor having end faces disposed adjacent to the end walls and a plurality of circumferentially-spaced apex portions, one more in number than the number of said lobes with each apex portion substantially tracing said epitrochoidal

inner surface during rotor rotation and being disposed in sealing engagement with the inner surface of the peripheral wall to form a plurality of working chambers between the rotor and the peripheral wall which vary in volume upon relative rotation of the rotor within the outer body; sealing members at each apex portion of the rotor for maintaniing sealing engagement with the inner surface of the peripheral wall, the sealing members being mounted for radial movement in the rotor, said rotor having a geometric eccentricity of its axis from the outer body axis which is less than the eccentricity determining the shape of the epitrochoidal inner surface.

8. The invention as defined in claim 7, in which the shape of the inner surface of the peripheral wall is modified to counteract radial movement of the sealing members which results from changes in the shape of the inner surface due to thermal gradients.

9. The invention as defined in claim 7, in which said geometric eccentricity of the rotor is less than the eccentricity determining the shape of the epitrochoidal inner surface by the amount that bearing clearances, deflections, and the like displace the center of the rotor radially outward from its geometric position relative to the outer body avis because of centrifugal forces on the rotor created during operation of the mechanism.

10. The invention as defined in claim 7, in which the profile of the inner surface is defined by a curve having basically the form of an epitrochoid but parallel to and outwardly spaced by an amount equal to r from a true epitrochoid, the value of r being substantially equal to the radius of curvature of the contacting surface of the tips of the sealing members.

11. A rotary mechanism comprising a hollow outer body having an axis, axially-spaced end walls, and a periphery wall interconnecting the end walls, the inner surface of said peripheral wall having basically the profile of a multi-lobed epitrochoid; a rotor mounted within

the outer body for rotation relative to the outer body on an axis ecentric from and parallel to the axis of the outer body and for planetary motion about the outer body axis; the rotor having end faces disposed adjacent to the end walls and a plurality of circumferentially-spaced apex portions, one more in number than the number of said lobes with each apex portion susbtantially tracing said epitrochoidal inner surface during rotor rotation and being disposed in sealing engagement with the inner surface of the peripheral wall to form a plurality of working chambers between the rotor and the peripheral wall which vary in volume upon relative rotation of the rotor within the outer body; sealing members at each apex portion of the rotor for maintaining sealing engagement with the inner surface of the peripheral wall, the sealing members being mounted for radial movement in the rotor, the profile of the epitrochoidal inner surface being defined by a curve having basically the form of an epitrochoid but parallel to and outwardly spaced from a true epitrochoid by a predetermined distance r and in which the tip of each seal member engaging the epitrochoidal surface has a radius of curvature which is substantially greater in magnitude than the predetermined amount r.

12. The invention as defined in claim 11, in which the radius of curvature of the contacting surface of the tips of the sealing members is sufficiently greater than r to greatly reduce radial movement of the sealing members relative to the rotor in those regions of the inner surface of the outer body where pressure differences across the sealing members are substantially high.

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