April 6, 1965 M. BENTELE ETAL 3,176,915

COOLING SYSTEM FOR A ROTARY MECHANISM

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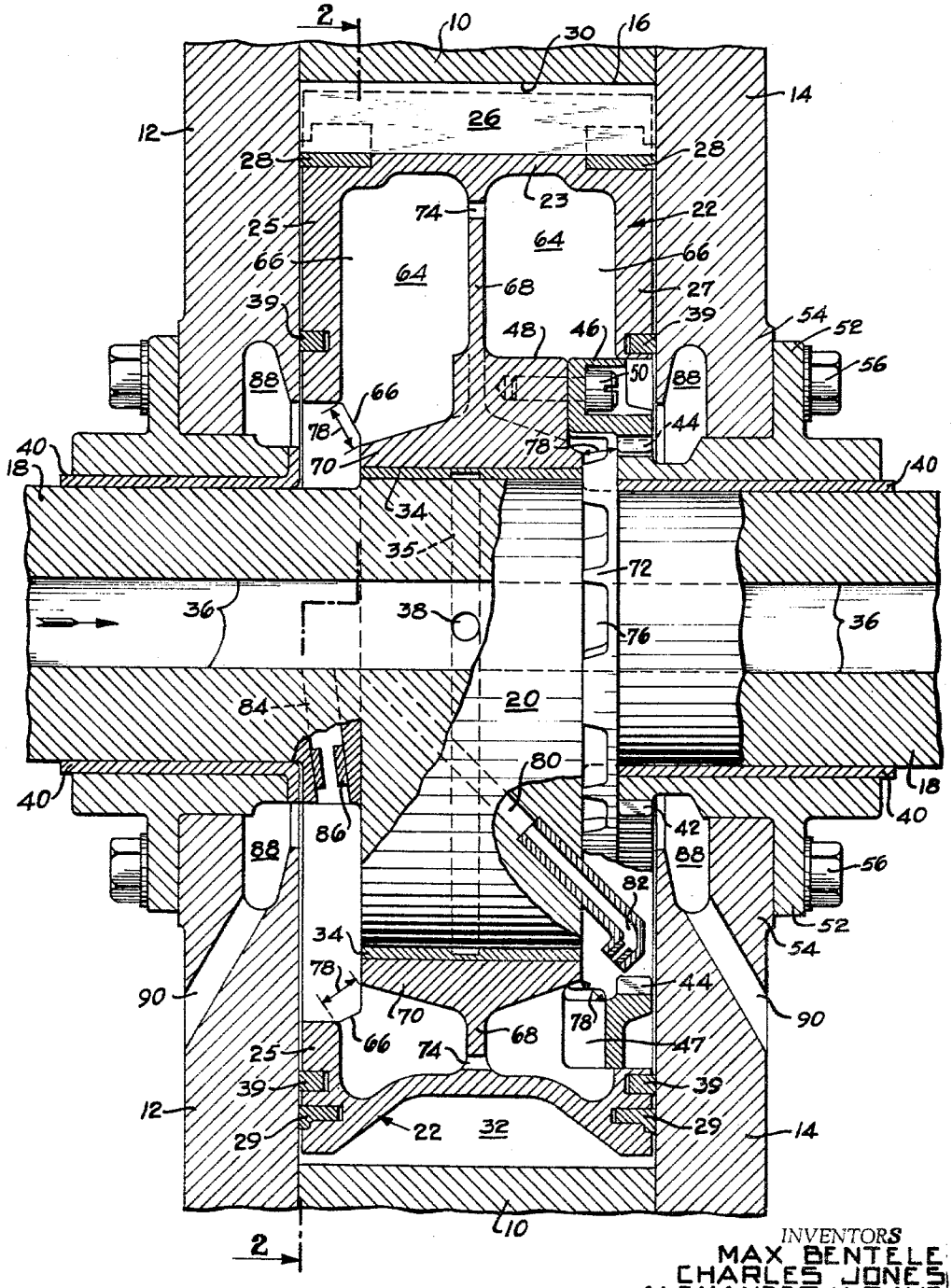


FIG-1

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COOLING SYSTEM FOR A ROTARY MECHANISM

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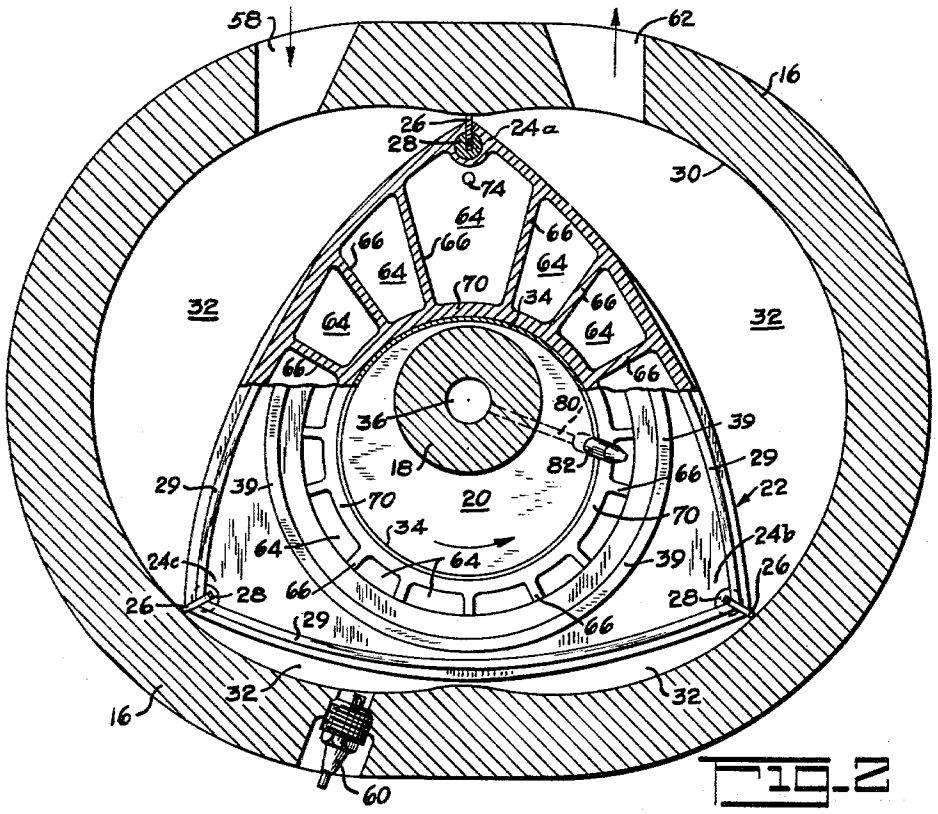


FIG. 2

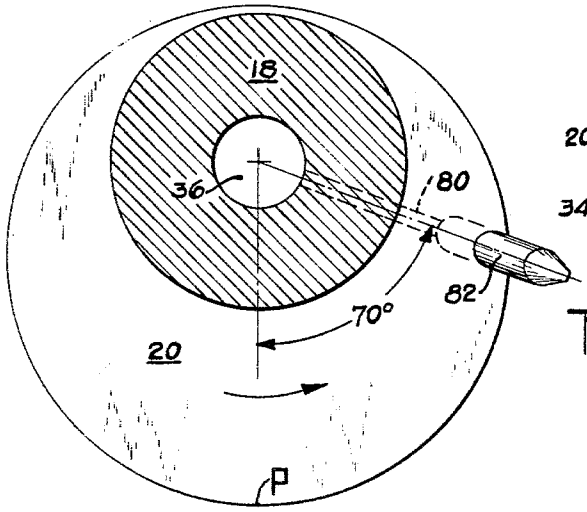


FIG. 4

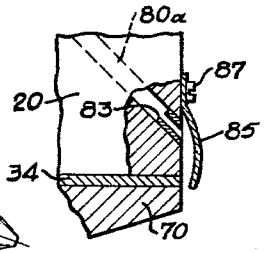


FIG. 5

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COOLING SYSTEM FOR A ROTARY MECHANISM

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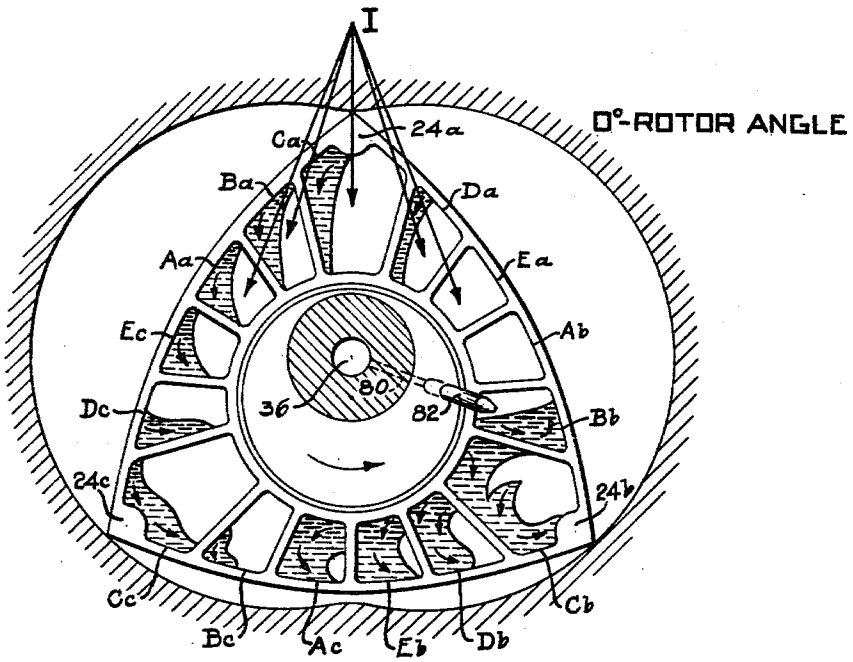


FIG. 3A

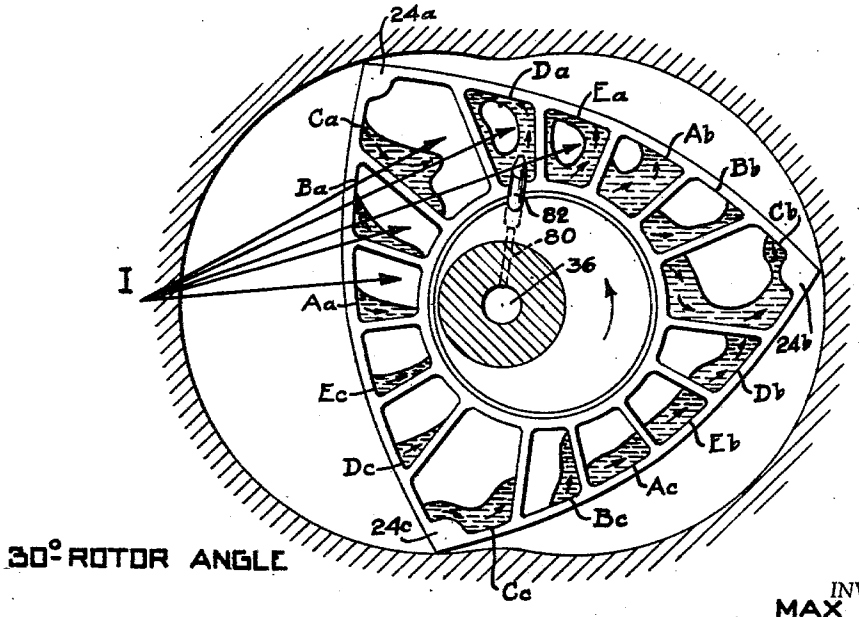


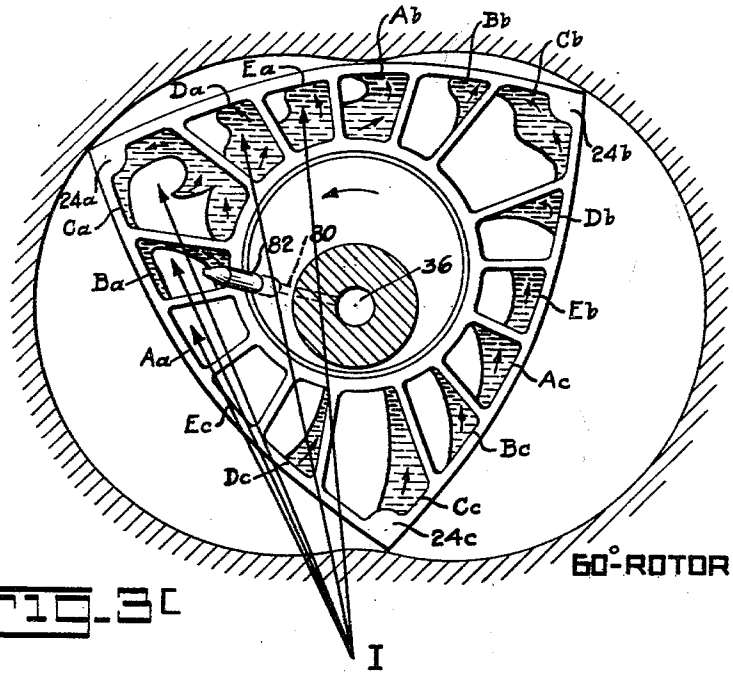
FIG. 3B

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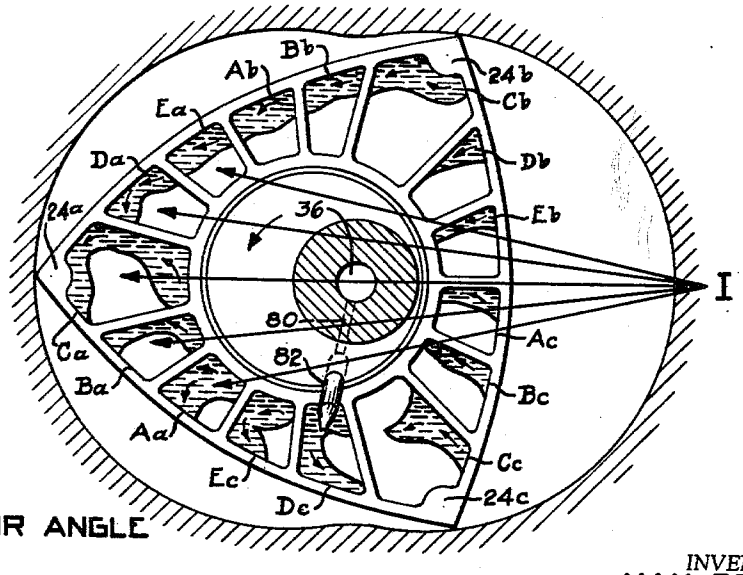
COOLING SYSTEM FOR A ROTARY MECHANISM

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60°-ROTOR ANGLE

FIG-3C



90°-ROTOR ANGLE

FIG-3D

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COOLING SYSTEM FOR A ROTARY MECHANISM

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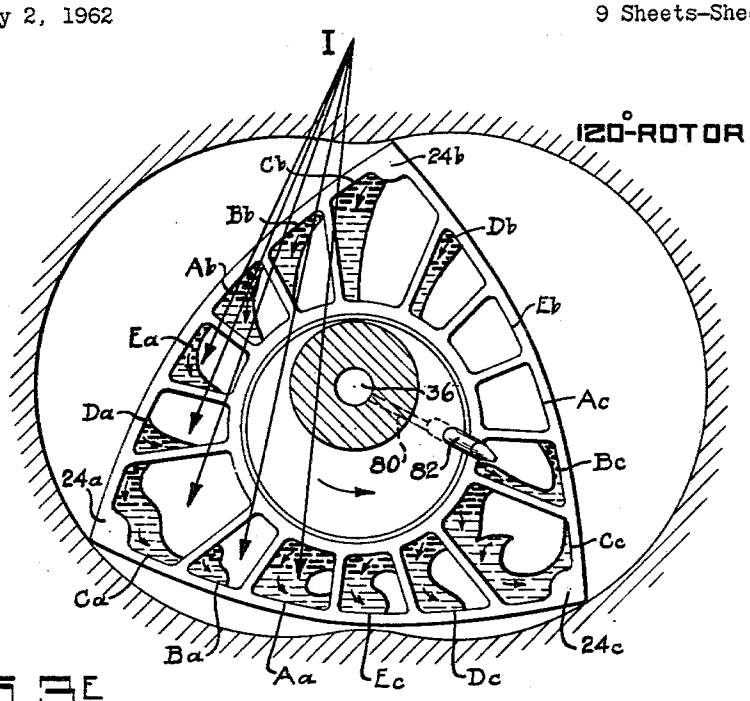
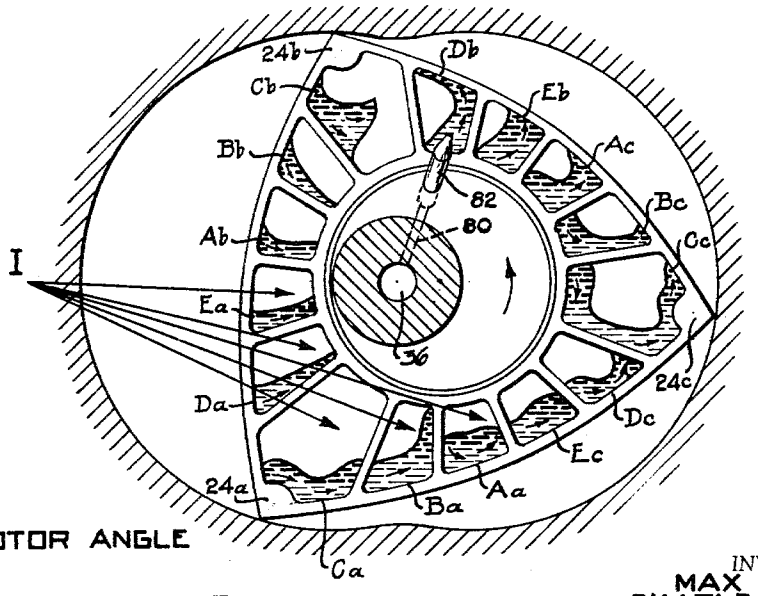


FIG. 3E



150°-ROTOR ANGLE

FIG. 3F

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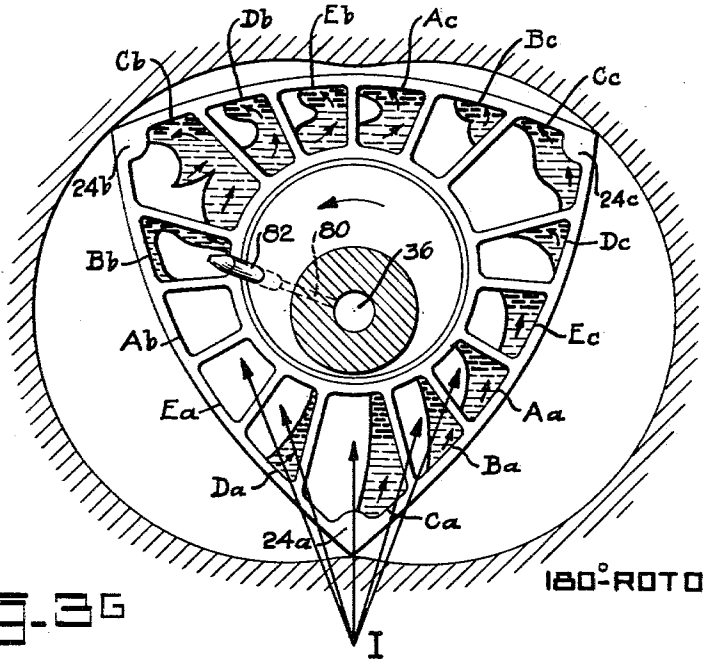
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COOLING SYSTEM FOR A ROTARY MECHANISM

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180°-ROTOR ANGLE

FIG-3G

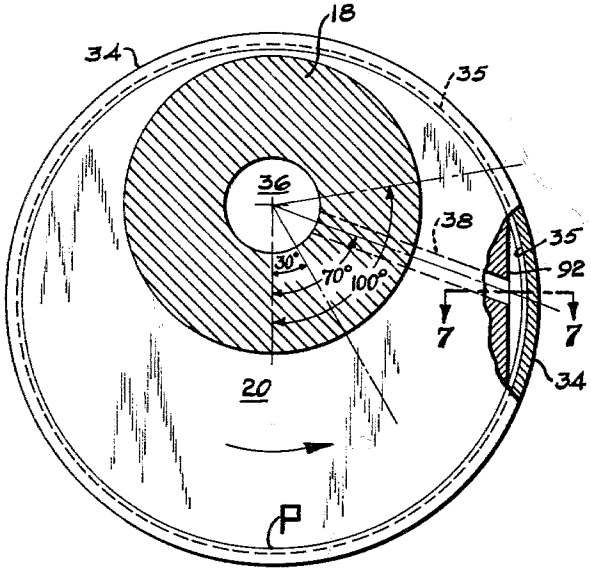


FIG-6

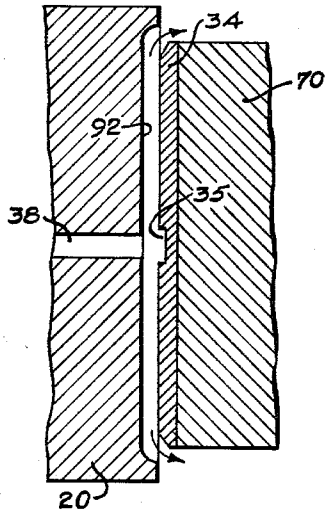


FIG-7

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COOLING SYSTEM FOR A ROTARY MECHANISM

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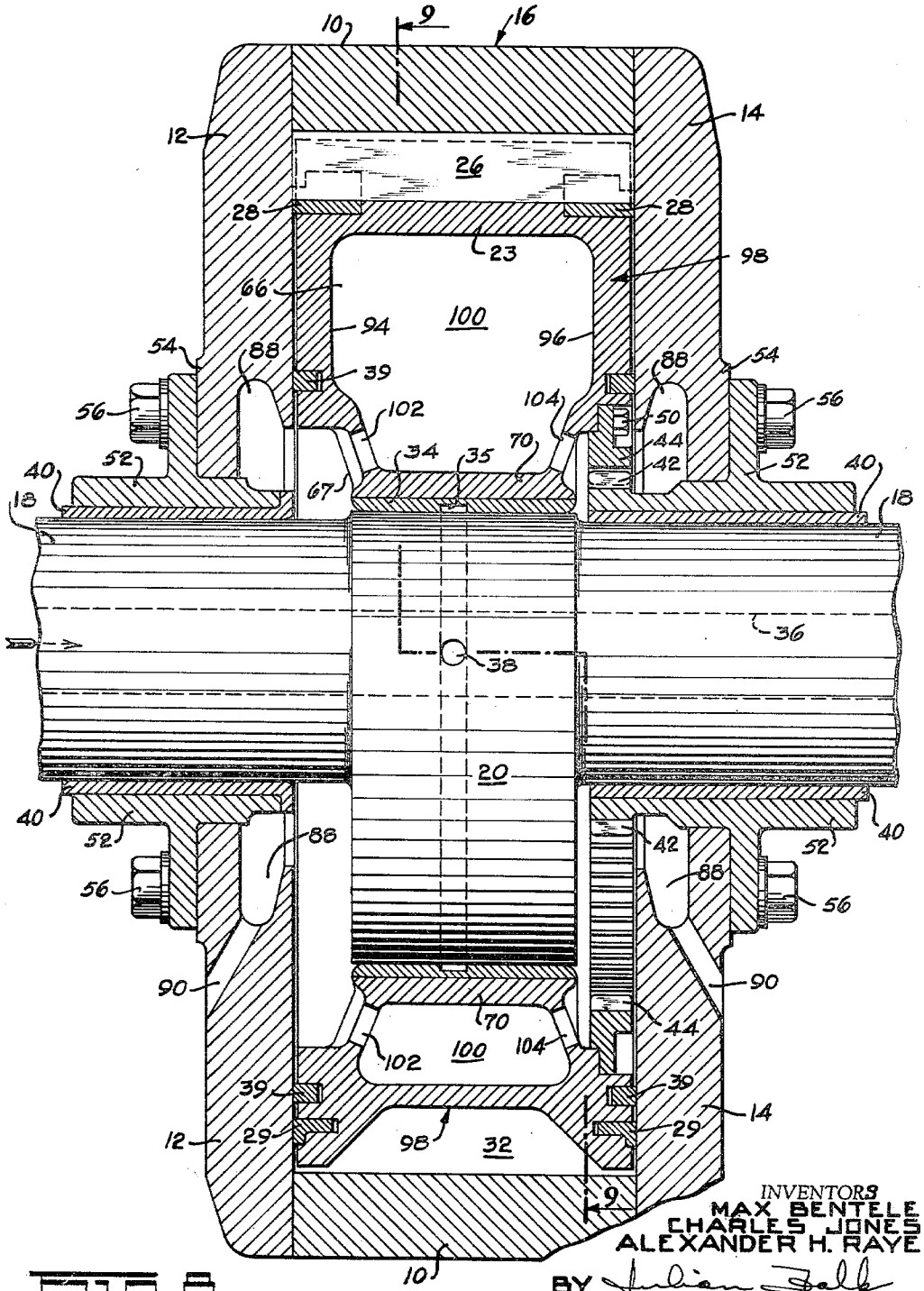


FIG. 8

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COOLING SYSTEM FOR A ROTARY MECHANISM

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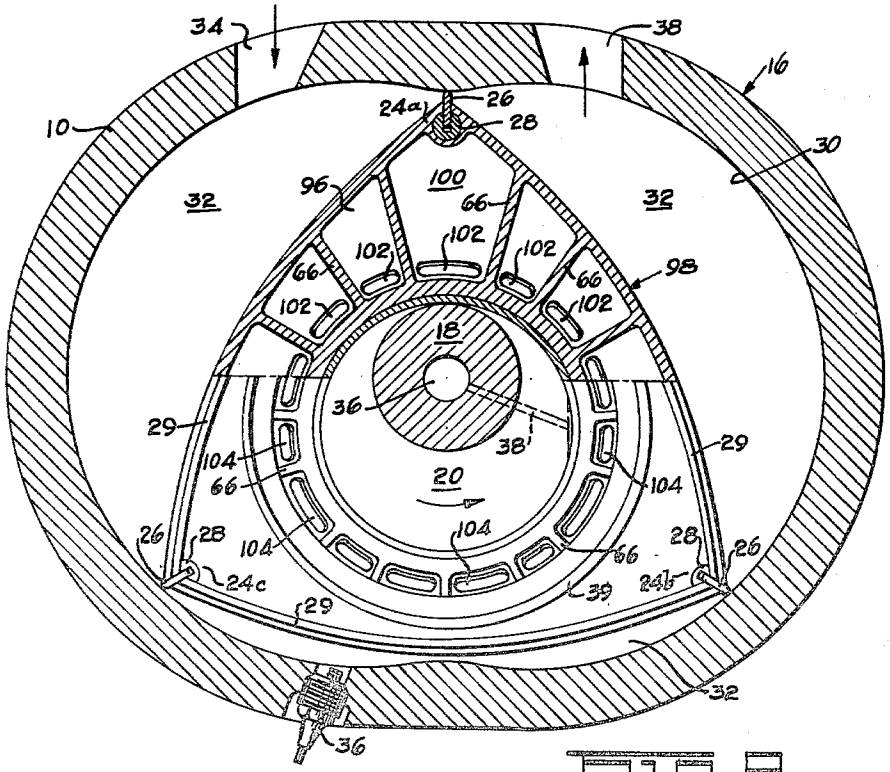


FIG. 9

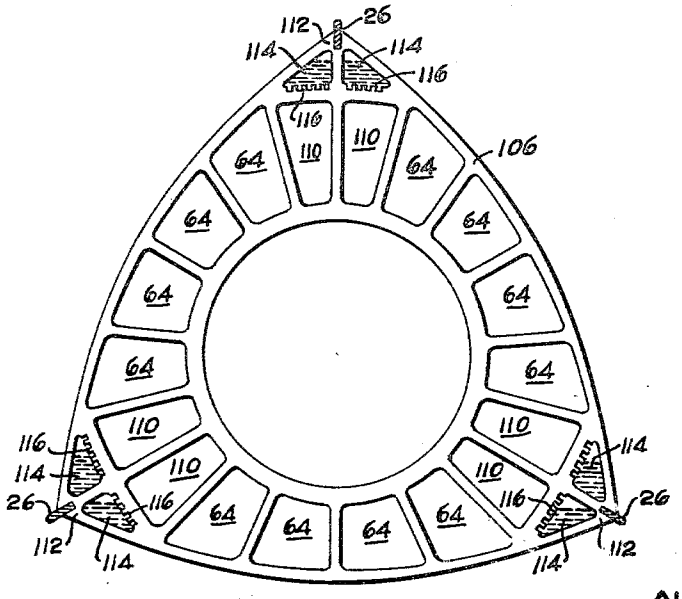


FIG. 10

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COOLING SYSTEM FOR A ROTARY MECHANISM

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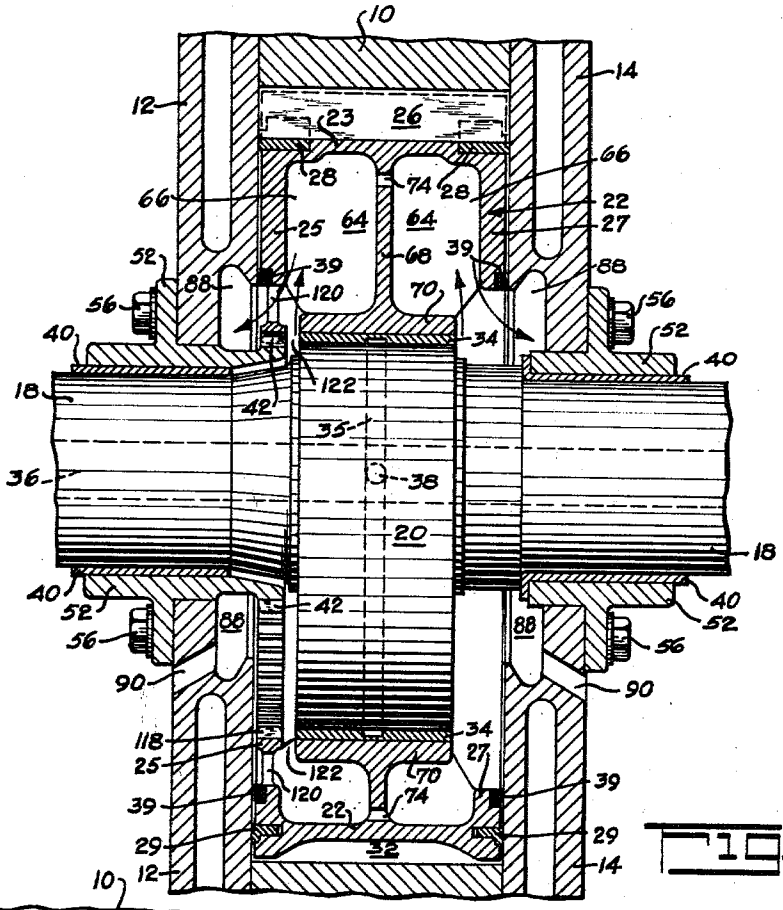


FIG. 11

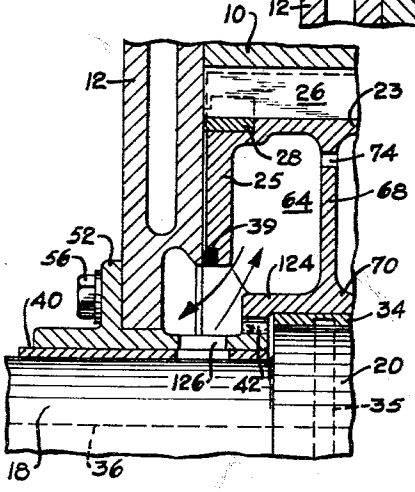


FIG. 12

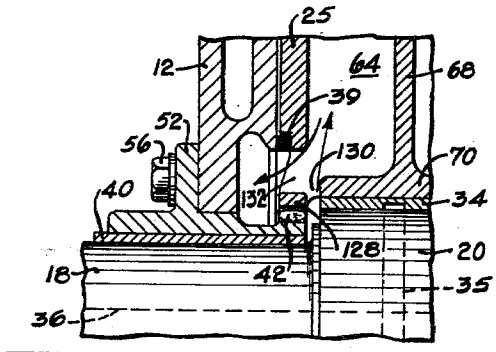


FIG. 13

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United States Patent Office 3,176,915 Patented Apr. 6, 1965

3,176,915

COOLING SYSTEM FOR A ROTARY MECHANISM

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Filed July 2, 1962, Ser. No. 206,753
34 Claims. (Cl. 230—210)

This invention relates to a rotary mechanism for fluid pumps, fluid motors, combustion engines or the like and more particularly to means for cooling the rotary mechanism:

The invention is best understood when described with reference to a rotary combustion engine and, although not limited to such use, is so described herein. A rotary combustion engine as described herein may be of the type disclosed in United States Patent 2,988,065, issued on June 13, 1961.

The invention is concerned with circulating a coolant through the interior of the rotor in order to keep the rotor temperatures from becoming excessive. Attempts have been made at cooling the interior of the rotor by continuously feeding oil into the rotor interior which keeps the rotor full of oil, at all times. This method is not satisfactory in all cases because keeping the rotor full of oil adds a substantial amount of weight to the rotor and has to be appropriately counter-balanced which then raises the total weight of the engine considerably. Furthermore, when the rotor runs full of oil high heat transfer is difficult to obtain.

The invention overcomes the drawbacks of the previous arrangements by directing a small amount of oil into compartments in the rotor at predetermined rotative positions of said rotor. The oil is substantially scavenged from each rotor compartment before the rotor reaches the predetermined position at which a small amount of oil is again directed therein whereby said oil impinges on the heated walls of the compartments within the rotor and the rotor thereby runs with only a relatively small amount of oil therein during its rotation. By directing a small amount of oil on the interior surface of each compartment and periodically scavenging it, greater heat transfer is obtained since oil is thrown against the heated metal surface of the compartments and spread over said surface in a thin film-like fashion providing greater cooling per unit area. This permits the rotor to run at cooler metal temperatures with consequent reduction of thermal distortion, reduction of deposits tending to stick the rotor seals, and therefore improving seal operation. The fact that the oil is periodically scavenged from each rotor compartment permits the rotor to run with relatively little oil thereby minimizing the problem of additional weight in the rotor because of the oil therein and further requires a smaller total oil flow in the engine and a smaller oil pump may be used. This of course permits a lighter engine overall with consequent reduction in cost and improvement in operation also since the oil is present in the rotor and in contact with the heated walls for a relatively short time the tendency for the oil to deteriorate due to the heat is substantially lessened. In one embodiment of the invention, the lubricating oil supplied for lubricating the rotor bearing, upon leaving said bearing is directed to the rotor interior and is advantageously used as the coolant for the rotor which eliminates the need for a separate coolant passage means for the rotor. The added heating of the oil through the rotor in addition to the heat added at the bearing is not detrimental to the oil and is within a range in which the oil can readily handle the absorption of the rotor and bearing heat. This of course leads to more effective use of the oil supplied in the engine.

Accordingly it is one object of the invention to provide a novel and improved cooling system for a rotor mechanism.

It is another object of the invention to provide a coolant for the interior of a rotor in a rotary mechanism while maintaining said rotor substantially empty during its rotation.

It is an additional object of the invention to impinge a coolant on the interior surface of a rotor in a rotary mechanism in predetermined rotative positions of said rotor and to periodically drain said rotor at other predetermined positions during each complete rotation of said rotor.

It is further an object of this invention to provide a plurality of circumferentially-spaced, generally-radial web members in the interior of a hollow rotor forming a plurality of circumferentially-spaced compartments therein and means for sequentially directing a small amount of oil on to the interior surface of each of said compartments during predetermined rotative positions of the rotor compartments and for periodically at least partially draining said compartments before said rotor reaches said rotative positions.

It is further an object of this invention to provide a novel arrangement utilizing the same medium and source used in lubricating the moving parts in a rotary mechanism as the cooling means for cooling the rotor in said mechanism.

Other objects and advantages of the invention will become apparent upon reading the following detailed description of the construction and method of operation with the accompanying drawing. The novel features which are considered as characteristic of the invention are set forth in particular in the appended claims.

In the drawings:

FIGURE 1 shows a sectional view of a rotary mechanism showing one embodiment of the invention,

FIGURE 2 is a view taken along line 2—2 of FIGURE 1 but with a coolant supply nozzle on the left side similar to the nozzle on the right side of FIGURE 1,

FIGURES 3A–3G are diagrammatic illustrations of the oil circulation in the rotor at different states of rotor rotation,

FIGURE 4 is an enlarged view of the eccentric and shaft of FIGURE 2 illustrating the preferred location of the jet nozzle,

FIGURE 5 is a partial sectional view of the rotor and eccentric illustrating another embodiment of the invention,

FIGURE 6 shows a rotor bearing, eccentric and shaft for use in FIGURE 1 and illustrates the preferred location of an oil flat in the eccentric,

FIGURE 7 is an enlarged sectional view showing the oil flat of FIGURE 6 taken along line 7—7 of FIGURE 6,

FIGURE 8 shows another rotor embodiment of the present invention,

FIGURE 9 is a view taken along line 9—9 of FIGURE 8, the rotor being shown empty,

FIGURE 10 is a sectional view showing another rotor which may be used in the present invention, and

FIGURES 11–13 are sectional views showing other rotors which may be used in the present invention.

In FIGURE 1 there is shown a rotary combustion engine having an outer body generally designated at **10** composed of a pair of end walls **12** and **14** interconnected by a peripheral wall **16** forming a cavity therein. As shown, for example, in FIGURE 2, the profile of the inner surface of the outer body peripheral wall **16** preferably is basically a two lobed epitrochoid. A shaft **18** is rotatably mounted co-axial with the cavity and has an

eccentric portion 20 formed thereon upon which is rotatably mounted a rotor 22 whose outer peripheral wall 23 forms three circumferentially spaced apex portions 24*a*, 24*b*, and 24*c* as shown in FIGURE 2. Seal strips 26 are mounted partly in grooves in the peripheral wall 23 provided in each apex portion and partly in intermediate seal bodies 28 mounted in said grooves and extend from one end face of the rotor to the other end face for continuous sealing engagement with the inner surface 30 of the outer body peripheral wall 16 thereby forming a plurality of working chambers 32 (see FIGURE 2) which upon relative rotation of the rotor 22 and outer body 10 vary in volume. Side seals 29 are mounted in each end wall of the rotor adjacent the periphery thereof and abut the intermediate seal bodies 28 to provide a continuous seal adjacent the periphery of the rotor on each side thereof.

A suitable plain sleeve type bearing 34 is supported in the rotor 22 for rotatively supporting the rotor on the eccentric 20. An annulus 35 may be cut out from the inner contour of the bearing 34 to communicate with an oil supply passage 36 in the shaft 18 through an oil hole 38 to aid in supplying oil to and distributing oil around the contacting surfaces of the bearing and eccentric. Oil seals 39 are provided in each end wall of the rotor and engage the inner faces of the end walls 12 and 14 to prevent any oil from leaking into the working chambers 32. Suitable bearings 40 are also mounted in the outer body end walls 12 and 14 as shown in FIGURE 1, for supporting the rotating shaft 18 in the outer body. An externally-toothed gear 42 is secured to the outer body end wall 14 by screws 56 and is disposed in mesh with an internally-toothed gear 44, mounted on the rotor 22 at the adjacent end face of the rotor said gear 44 forming an inner extension of the rotor end wall 27. The internal gear 44 has a plurality of circumferentially-spaced bosses 46 formed thereon which abut corresponding bosses 48 on the interior of rotor and screws 50 passing through the bosses 46 and threaded into the bosses 48 serve to secure the gear 44 to the rotor. In between the boss portions 46 formed on gear 44, there are ribs or webs 47 formed in the gear 44 which mate with radial web members or partitions in the rotor to aid in defining compartments within the rotor, as will be explained in greater detail below. The gear 42 has a radially-extending annular flange 52 which abuts a boss 54 on end wall 14 and screws 56 pass through this flange to secure the gear 42 to the end wall 14. The gears 42 and 44 serve to help rotatively position the rotor with respect to the epitrochoidal surface of the peripheral wall 16. The internal gear 44, instead of being fastened to the rotor by screws as shown in FIGURE 1, may be made integral with the rotor by suitable casting or machined from the end surface of wall 27. In the embodiment having a two-lobed epitrochoid and a three-lobed rotor the ratio of rotation of the shaft with respect to the rotor is 3:1 wherein for each rotation of the rotor about its axis the shaft rotates three times around its axis. An intake port 58 (see FIGURE 2) is provided for admitting air and/or fuel-air mixture, and ignition means 60 may be provided for igniting the mixture and an exhaust port 62 is provided for expelling the burnt gases so that the stages of intake, compression, expansion and exhaust may be carried out.

As stated above, the primary object of this invention is to provide a novel and simple cooling mechanism for the interior of the rotor. In order to carry out this object the rotor is made hollow and the entire interior of the rotor is divided into a plurality of circumferentially-spaced compartments 64 by substantially radial webs or partitions 66. The web members 66 extend radially from the outer wall 23 of the rotor and are integral with the hub portion 70 and extend axially between the end walls 25 and 27 of the rotor to provide support for the rotor outer walls and thereby strengthening the entire rotor. The compartments 64 are preferably further divided by an annular web 68 disposed transverse to the rotor axis and extending from the rotor hub portion 70 to tne peripheral wall 23 of the rotor. Openings 74 (see FIGURE 1) may be provided in wall 68 adjacent to its outer periphery to aid in equalizing the amount of coolant on both sides of the rotor by allowing the coolant to flow axially between axially adjacent compartments 64. As shown in FIGURE 1, the hub portion 70 terminates short of the rotor end walls 25 and 27 and also on the gear side short of the gear member 44 forming the axial width of an annular gap 78 which is formed at each end of the rotor hub portion for allowing the coolant to flow in and out of the compartment 64. The annular gap 78 is disposed adjacent the bottom of the compartments 64 and preferably extends radially outwardly a short distance therefrom, which is determined by the fact that the inner diameter of the rotor end walls is greater than the outer diameter of the hub 70, to facilitate flow of coolant out of the compartments. Preferably, as illustrated in FIG. 1, the radius of the hub portion 70 is a minimum at its ends adjacent the rotor end walls 25 and 27 so that the hub portion slopes radially inwardly in a direction toward said rotor end walls. By this means, the coolant will be prevented from collecting at the radially inward portion of each of the compartments 64. The gear member 44 is designed so that the root diameter of its gear teeth is larger than the outer diameter of the hub 70 so that the gear member has a larger effective diameter than that of said hub portion and therefore does not block the gap 78 and thereby interfere with the flow of coolant through gap 78 out of compartments 64. As indicated by the arrows in the gap 78 on the gear side of the rotor in FIGURE 1, gap 78 extends from the root portion of the gear 44 to the rotor hub 70 and forms an opening such that the coolant may flow both radially inwardly and axially away from the rotor when flowing out of the compartments 64.

As further shown in FIGURE 1, the gear 44 which forms an extension of end wall 27 has an axially extending rim 72 which is cut out at portions, indicated at 76, so that the gear will not restrict the flow of oil in and out of compartments 64 in that the cut-outs 76 in the gear rim 72 are substantially aligned with the inner ends of the adjacent compartments 64 and gaps 78 to provide openings therefor.

As illustrated in FIGURE 1 the webs 66 extend radially inwardly at least to the compartment openings 78 so that liquid coolant in draining from a particular compartment cannot spill into an adjacent compartment over their separating web 66 before draining through its opening 78.

As viewed in FIGURE 1, a lubricating oil passage 36 is provided in shaft 18 for transporting lubricating oil from a suitable source as, for example an oil pump. The passage 36 which normally carries lubricating oil to the rotor bearings may also be utilized in supplying lubricating oil to the rotor compartments 64 which may be used as the cooling medium for said rotor. For this purpose a passage 80 may be provided in the eccentric, said passage communicating at its inner end with the passage 36 while at its outer end the passage opens through the side of the shaft eccentric 20 on the gear side of the rotor and terminates in a nozzle 82 which is oriented so as to direct a jet of lubricating oil radially outwardly into the compartments 64. A single source such as passage 80 and nozzle 82 may be used for directing lubricating oil into one side of the rotor in combination with openings such as at 74 for allowing flow between both sides of the rotor or with a rotor having no intermediate radial wall such as wall 68. In the embodiment in FIGURE 1, there is shown on the side of the rotor opposite the gear side a passage 84 communicating at its inner end with passage 36 and having substantially at its outlet a nozzle 86. The nozzle 86 is positioned so as to direct a jet of lubricating oil from shaft passage 36 radially outwardly onto the interior surface of the compartment 64 on its respective side of the rotor. When two nozzles 82 and 86 are used such as

shown in FIGURE 1 the ports 74 may be eliminated, if desired. Also, two passages on the eccentric, such as passage 80, one on each side of the eccentric 20, may be used with a pair of nozzles 82 for directing lubricating oil to both sides of the rotor instead of the arrangement shown in FIGURE 1. Thus, FIGURE 2 illustrates a nozzle 82 on the left side (FIGURE 1) of the rotor eccentric. Similarly an arrangement such as the passage 84 and nozzle 86 may be used on each side of the rotor although on the side of the gears 42 and 44 an arrangement such as the passage 80 and nozzle 82 is preferred.

FIGURES 3A–3G diagrammatically illustrate the rotor and shaft in successive positions of rotation such that the rotor has rotated 30° (90° of shaft rotation) for each position to the next. As the rotor rotates each point on the rotor traces an epitrochoidal path. In any position of the rotor each particle of oil in a rotor compartment 64 adjacent a portion of the compartment wall has substantially the same motion as that of said wall portion and therefore is responsive to substantially the same acceleration force as that produced on said adjacent compartment wall portion during rotor rotation. It may be further shown that the acceleration of the various points on the rotor is at any instant the same as those which would result from the rotational motion of the rotor about a point which may be called the instantaneous center of acceleration. In each of FIGURES 3A–3G the point I indicates the instantaneous center of acceleration of the rotor for the particular position illustrated. Hence in each of these figures at least each particle of lubricating oil in the compartments 64 adjacent to the compartment walls is subject to a centrifugal force radially outward away from the instantaneous center of accelation I. In FIGURES 3A–3G, the long arrows drawn from the instantaneous center I to the compartments in apex portion 24*a*, generally indicate the direction of the centrifugal force of points in the rotor along each line and may be further said to indicate the general direction of the centrifugal force of the particles of oil in each respective compartment. For convenience of illustration the arrows are drawn to only one group of compartments but it should be understod that like arrows could be drawn from the center I to each of the compartments in the rotor. Because the rotor, as illustrated, rotates in a counter-clockwise direction, the oil will also rotate in a counter-clockwise direction about its compartment 64 in response to the above-mentioned centrifugal forces. The movement of the oil in the compartments is illustrated by the short arrows in the oil contained in the compartments.

As viewed in FIGURE 3A, the rotor and shaft are assumed to be at 0 degrees of rotation. The oil in the compartments 64 can be seen to be flowing about the compartments in a counter-clockwise direction. In particular, in apex portion 24*a* the oil is plastered against the leading wall of the compartments A*a*, B*a*, C*a* and D*a* due to its previous counter-clockwise movement about its compartments and in response to the centrifugal forces acting on the oil directed from the instantaneous center I, the oil is flowing inward along the leading wall where it will eventually drain out of the compartments through gap 78 (see FIGURE 1) and thence into an annular gutter 88 from which it is drained off through suitable drain passages 90. Compartment E*a* has already substantially fully drained of its heated oil and upon further rotation of the nozzle 82 will be sprayed with fresh cooling oil. It should be particularly noted, that due to the movement of the oil in the compartments A*a*–E*a* even though all of the oil in all of said compartments might not fully drain therefrom, portions of the walls of each compartment are substantially clear of all heated oil and present a substantially bare metal surface where a fresh supply of cool oil is sprayed into said compartments, as is shown being introduced into compartment B*b* of apex portion 24*b*. The oil already introduced into compartments B*b*–E*b* of said apex portion 24*b* is flowing in a counter-clockwise direction, as outlined above and will flow from the trailing wall of each compartment outwardly along the inner surface of the outer wall, as illustrated in compartment C*c* of apex portion 24*c*, toward its leading wall from where it will drain as in the case of compartment A*a*–E*a* of apex portion 24*a*.

In FIGURE 3*b* the rotor has rotated 30 degrees from 0 degree position of FIGURE 3A and the shaft has rotated 90 degrees since the shaft rotates three times to every one rotation of the rotor. It can be seen therein that the nozzle 82 has rotated past compartments B*b* and A*b* of apex portion 24*b* and compartment E*a* of apex portion 24*a*, and has sequentially sprayed oil into said compartments as it rotated by these compartments and is presently spraying oil into compartment D*a* of apex portion 24*a*. The status of compartment E*c* of apex portion 24*c* is such that the oil is draining therefrom and the trailing wall and outer wall present a bare metal surface onto which the fresh oil from nozzle 82 will be sprayed when it reaches this compartment. The oil just introduced into the compartments of apex portion 24*b* can be seen to be starting its counter-clockwise movement in said compartments.

When the rotor rotates 60 degrees, shown in FIGURE 3C, from the 0 degree position (180 degrees shaft angle) the nozzle 82 has reached compartment B*a* of apex portion 24*a* and is spraying fresh oil on the bare metal surfaces of the walls of this compartment and compartment A*a* has substantially drained of heated oil and the walls of this compartment also present bare metal surfaces for the introduction of fresh cooling oil. The compartments in apex portion 24*c* are in the state of draining in response to the forces directed inward, as illustrated, which is similar to the state of the compartments in apex portion 24*a*, as was present in the 0 degree position of FIGURE 3A. Again the walls of the compartments in apex portion 24*c* as was the case of apex portion 24*a* in the 0 degree position (FIGURE 3A), are substantially free from heated oil in preparation to the passing of nozzle 82 and the remaining oil in the compartments is in the process of draining from said compartments.

Eventually the nozzle 82 will sequentially spray all of the compartments of the rotor with oil as it rotates with respect to said compartments as can be seen from the 0 degree position, where the compartments of apex portion 24*b* are illustrated as being supplied, to the 30 and 60 degree position where the compartments of apex portion 24*a* are illustrated as being supplied and finally to the position of 90 and 120 degrees (FIGURES 3D and 3E) where the compartments of apex portion 24*c* are illustrated as being supplied. At approximately 135 degrees of rotor rotation (405 degrees shaft rotation), not shown, the nozzle will have again reached the compartments of apex portion 24*b* where they will again be sprayed with oil in the manner outlined above and as further illustrated in FIGURE 3F at the 150 degree rotor rotation position.

In between the supplying positions of each compartment the oil circulates in the compartments and drains therefrom so that the compartments go through a cycle of filling, draining and filling again. For example in FIGURE 3A compartment B*b* of apex portion 24*b* is in the process of just being sprayed with oil from nozzle 82. After this compartment is supplied, FIGURE 3B, the oil begins to circulate in a counter-clockwise direction about the compartment in response to the forces as described above, and at the position illutsrated in FIGURE 3E it can be seen that the compartment is in the process of draining being similar to that shown with respect to the compartments of apex portion 24*a* in the 0 degree position of rotor rotation (FIGURE 3A). When the nozzle again reaches compartment B*b* of apex portion 24*b* it will have substantially completely drained of all its heated oil so that, as described above, the fresh cooling oil sprayed from nozzle 82 will impinge on a substantially bare metal surface. In FIGURE 3G, 180 degrees rotor rotation, the

nozzle 82 has again reached compartment B*b* and as in the 0 degree position, oil is again being sprayed therein. It should also be noted that the compartments in apex portion 24*a* are diametrically opposite at 180 degrees from the 0 degree position but are in exactly the same state with respect to the oil in the compartments. Therefore, through 180 degrees of rotor rotation or one-half a revolution of the rotor in an engine having a two-lobed peripheral wall inner surface 30 each of the compartments is supplied once and will drain once, as outlined above, and therefore, for each complete revolution or 360 degrees of rotor rotation each compartment will be filled twice and drained twice.

It can be further shown that the number of times each compartment is supplied and drained will be the same as the number of lobes in the cavity. So, for a two-lobed outer body peripheral wall the supply and drainage each occur twice and for a three-lobed outer body peripheral wall the supply and drainage will each occur three times.

From the foregoing it can be seen that as compartments are being supplied other compartments are being drained at the same time. This is particularly advantageous in that only a small amount of weight is added to the rotor because of the oil and therefore, the allowance to be made for counter-balancing added weight to the rotor is very small. Also since the total engine oil flow is low in the present invention a smaller oil supply mechanism may be used thereby further reducing the overall engine weight and cost. Furthermore, by directing the oil on the heated substantially bare metal surface of the compartments better transfer of heat to the oil is obtained because the relatively cool oil is forced directly onto said heated surfaces and said oil spreads out over a substantially large heated area whereby greater cooling per unit area is obtained. An additional advantage is obtained by the fact that the web members 66 whose primary function is to strengthen the rotor by supporting its outer walls, are advantageously used to form the cooling compartments and no additional special structure need be added for providing a cooling mechanism. The quantity of oil in the compartments 64 of FIGURES 3A–3G has been exaggerated for purposes of illustration.

From the foregoing description of FIGURES 3A–3G it can be seen that the compartments are sequentially supplied with cooling oil each at a predetermined position of rotation of the rotor and said compartments are sequentially substantially drained of said cooling oil such that each compartment is substantially drained of cooling oil before it reaches the position where cooling oil is again supplied to that compartment whereby the rotor runs with relatively little oil therein during operation.

As stated above, with the engine illustrated the rotor 22 is rotatably mounted on the shaft eccentric 20 and the ratio of rotation between these elements is 3:1 wherein the shaft eccentric rotates three times for each revolution of the rotor. Since the nozzle 82 is mounted in the shaft eccentric 20, the nozzle will rotate with the shaft at three times the rotor speed and will sequentially pass the compartments during its rotation. The passageway 80 and nozzle 82 are so arranged in the shaft eccentric that, during their rotation as they pass each compartment, the jet of oil sprayed into the compartment is always initially directed against substantially bare metal surfaces. The preferred location of the passageway 80 and nozzle 82 in the shaft eccentric has been found to be approximately 70 degrees in the direction of shaft rotation from the point P of maximum radius of the shaft eccentric from the shaft axis as illustrated in FIGURE 4. However, the location may range from 0 degree or on a line through the point P of maximum eccentricity, to 100 degrees from said point P in the direction of shaft rotation. At any point beyond these two limits, it has been found that the oil draining from said compartments tends to interfere with the oil being introduced therein and the oil supply is therefore less efficient than within the limits of the above mentioned range.

FIGURE 5 shows an embodiment of the invention wherein another means for directing the oil into compartments 64 may be used. Instead of the nozzles 82 shown in FIGURE 1, a nozzle 83 is positioned in the outlet of passage 80 such that the nozzle 83 opens through a side wall of eccentric 20. The nozzle 83 produces a jet of oil which discharges against a deflecting plate member 85 fastened to the adjacent side wall of eccentric 20 by suitable means such as screws 87. The deflecting plate member 85 is shaped so that the jet of oil coming from nozzle 83 is directed into the compartments 64 through their inner openings 78. The circulation of the lubricating oil in compartments 64 of FIGURE 5 and the drainage therefrom is the same as that disclosed above in relation to FIGURES 3A–3G and reference may be made thereto for a more complete description of these features.

Due to the eccentric motion of the rotor with respect to the axis of the outer body and its rotation about its own axis, the plain sleeve type bearing 34 must absorb varying loads over its circumference so that on one side the clearance between the bearing 34 and shaft eccentric 20 will be very small and the bearing 34 may be considered loaded while on its other side the clearance will be larger and the bearing may be considered unloaded. As stated above, an annulus 35 preferably is formed in the bearing for allowing oil to flow from oil hole 38 in the eccentric around the bearing and axially outward between their contacting surfaces. Accordingly most of the oil supplied to the bearing will spill out from the bearing ends at the unloaded side of the bearing. The position of the unloaded portion of the bearing will of course vary over its circumference as the rotor rotates about its axis and moves with respect to the shaft eccentric.

It has been found however that at least some of the oil spilling from the ends of the rotor bearing 34 can be used for cooling the compartments. Some of this oil will be thrown outward into the compartments in response to the acceleration forces and aid in the cooling of said compartments while other portions of this oil will drain off into annular gutter 88. The unloaded region of the rotor bearing 34 generally extends from a region about 10 degrees from the point P of maximum eccentricity in the direction of rotor rotation and spans about 85 degrees in said direction. If sufficient oil spills from the ends of the bearing in its unloaded region the nozzles 82 and/or 86 may be entirely dispensed with. The amount of oil escaping from the ends of the plain sleeve type bearing 34 about the shaft eccentric may be increased by providing a flat on the shaft eccentric in the unloaded region of the bearing as will be described in FIGURES 6 and 7.

As explained above, the same oil supplied to the bearing 34 may be used to cool the rotor by being thrown outward by the acceleration forces into the compartments 64 after it drains from between the ends of the rotor bearing 34 and shaft eccentric. The circulation of the oil in the rotor compartments will be the same as described in FIGURES 3A–3G. As also explained above, additional oil may be supplied to the compartments 64 by providing an oil flat in the unloaded portion of the bearing for allowing oil to flow substantially unrestricted in an axially outward direction from the bearing oil supply annulus 35.

As viewed in FIGURES 6 and 7 the oil flat indicated at 92 is formed by flattening out a portion of the eccentric 20 thereby providing a passageway for the oil to flow axially-outward from the bearing annulus 35. As shown in FIGURE 7 the oil flat 92 preferably extends axially beyond both ends of the bearing 34 but terminates short of the ends of the shaft eccentric 20. The oil spill-

ing out from the flat and bearing ends will therefore be directed radially outward, as illustrated by the arrows in FIGURE 7. As also shown in FIGURE 7, the axial ends of the bearing 34 may be chamfered. The oil flat may be located in the eccentric within a range of 30 degrees to 100 degrees in direction of rotation from the point P of maximum eccentricity without any substantial interference with the draining oil but is preferably located 70 degrees in advance of the point P of maximum eccentricity as described above in the case of the jet nozzle 82 shown in FIGURE 4, but not necessarily so. The oil is supplied to the oil flat from the annular groove or annulus 35 in the bearing 34, said groove being supplied with oil as in FIGURE 1 by the shaft supply passages 36 and 38. Particularly by providing an oil flat, such as shown at 92, to increase the oil spillage out the ends of the bearing sleeve, the amount of oil spilling out the bearing ends can be made sufficient for purposes of cooling the rotor so that the nozzles 82 and 86 of FIGURE 1 can be dispensed with.

A further rotor embodiment of the invention is shown in FIGURES 8 and 9 wherein the same numerals used in FIGURE 1 are used therein to refer to similar elements. The modification shown therein makes use of a pair of rotor end walls 94 and 96 along with circumferentially-spaced web members 66 through which the loads placed upon the rotor may be transmitted to the rotor hub 70. The end walls 94 and 96 are connected at their upper ends with the rotor peripheral wall 23 and at their lower ends with the hub portion 70, thereby forming a substantially box-shaped rotor. The rotor, generally indicated at 98, is hollow and is divided by the circumferentially-spaced web members 66, as in FIGURE 1, into a plurality of compartments 100, said compartments being similar to compartments 64 of FIGURE 1 except for the absence of a transverse dividing partition 68. The webs 66 have extending portions 67 which extend beyond the rotor end walls 94, as shown in FIGURE 8, to aid the oil in draining from the compartments 100. The end walls 94 and 96 are provided with a plurality of apertures 102 and 104 positioned near the bottom of each compartment 100 for admitting and draining cooling lubricating oil to and from the compartments 100 in order that the rotor may be cooled. The shaft 18 has a lubricating oil supply passage 36 located therein, as in the embodiment of FIGURE 1, and a bearing oil supply hole 38 communicates with said passage 36 to supply lubricating oil to the rotor bearing 34. The bearing 34 has a cut out portion or annulus 35 to receive oil flow from the hole 38 and distribute said flow around the bearing from whence it flows axially between the contacting surfaces of bearing 34 and the eccentric 20, and then spills out from the bearing ends. As previously described this same oil may be further advantageously used as the coolant for the rotor.

Some of the oil released from the sides of the bearing will, in response to outward acceleration forces, be thrown radially outward through the apertures 102 and 104 into the compartments 100 and splashed against the inner surface of said compartments. As in the case of the embodiment of FIGURE 1, when the acceleration forces reverse, the oil will be drained from said compartments through its same apertures 102 and 104 much in the same manner as described in connection with FIGURES 3A–3G. Reference may be made to FIGURES 3A–3G for examples of the circulation of the oil to and from the compartments in response to the acceleration forces. Thus, it can be seen, that in the embodiment of FIGURES 8 and 9 a coolant supply passage is formed at the unloaded side of the bearing and that additional utilization of the oil supplied to the bearing surfaces for cooling the rotor is brought about and further, the need for a separate supply passage for providing coolant to the rotor may be eliminated. Of course, if desired or found necessary a flat may be provided in the eccentric preferably in the unloaded region of the bearing as described in relation to FIGURES 6 and 7 for increasing the coolant supply to the rotor.

As previously described each point on the rotor traces an epitrochoidal path during rotation of the rotor. The particular epitrochoid traced by any point on the rotor is dependent upon the ratio of the distance of the point from the rotor axis, R, to the distance between the rotor axis and the shaft axis generally known as the eccentricity E. This ratio may be referred to as the K factor and for larger values of K, the shape of the epitrochoid tends to smooth out at the intersection of the lobes. For example, in the present invention as shown in FIGURE 2, since the eccentricity, E, remains constant, a point at the tip of any of the apex portions will trace the inner surface of the peripheral wall and the distant R may be said to be a maximum in the embodiment shown therein. At any point inward from the apex point the distance R will decrease giving a smaller K factor and the shape of the epitrochoid will have a more distinct inward deviation at the intersection of the lobes or in other words the reversal of the curvature of the epitrochoid becomes greater at the intersection of the lobes, for lesser values of K. It has been found that when using an epitrochoidal inner surface 30 (FIGURES 1–2) of the outer body having a relatively large K factor, since the reversal of curvature at the lobe junctions is less, the reversal of the acceleration forces at the rotor apex portions is less and the oil might tend to accumulate at these apex portions instead of being forced out by the acceleration forces. FIGURE 10 illustrates a modification which provides adequate cooling of the rotor apex portions even though said apex portions generate an epitrochoid of relatively large K factor.

In the embodiment of FIGURE 10 there is shown a sectional view of a rotor 106 which is similar to the rotor of FIGURE 1 and like parts will be referred to by the same reference numerals as used in FIGURE 1. The rotor 106 is hollow and has a plurality of divided compartments which are basically the same as compartments 64 of the rotor of FIGURE 1. As stated above, at points wherein the K factor is largest the oil may tend to accumulate which therefore may affect the efficiency of the cooling system at these points. In the rotor 106 the compartments at the areas of largest K factor or maximum distance from the rotor axis are modified so as to prevent the accumulation of oil therein. Therefore, compartments 110 in the apex regions 112 are designed so as to terminate short of the rotor apex portions 112 instead of extending out to these portions as in the rotor of FIGURES 1 and 2 thereby reducing the K factor of the outer areas of these compartments 110 and preventing the accumulation of oil therein. In order to provide cooling for the apex regions 112 separate closed compartments 114 are provided, said compartments having a metal such as sodium, potassium or a mixture thereof sealed therein, said metal having a low melting point such that it is liquid at engine operating temperatures. The low melting point metal will absorb the heat from the peripheral wall at the apex portions 112 and transmit the heat through the inner walls of compartments 114 to the cooling liquid being circulated in apex compartments 110. Fins 116 may be provided on the inner surface walls of the compartments to increase the cooling area for absorbing the heat from the low melting point metal contained therein. The compartments 114 are not completely full of the above-mentioned metal and when the metal is in a liquid state it will circulate within the compartments 114. Thus it can be seen that accumulation of oil in the apex regions of the rotor is prevented which results in an increase in cooling efficiency in the rotor.

In FIGURES 11–13 there are shown modifications of the rotor in which separate openings are provided for supplying and draining oil in each of the compartments. In FIGURE 11 there is shown a rotor otherwise similar to that of the embodiment of FIGURE 1 and like numerals are used to refer to similar elements. However, in the

embodiment of FIGURE 11 the internally-toothed gear or rotor gear **118** is integral with the rotor, as it may be, as generally explained above in connection with FIGURES 1 and 2, and is formed on a radially inward extension of rotor end wall **25**. The gear **118** of FIGURE 11 has a diameter which is less than that of the hub portion **70** which is the opposite case of that in the embodiment of FIGURE 1. Openings **120** are provided in the outer portion of gear **118** at each rotor compartment in order that the oil draining from the compartments may exit therefrom without substantially interfering with the oil being supplied to the compartments. In this embodiment the oil is supplied through an opening **122** formed between the gear **118** and hub portion **70** as indicated by the arrow pointing into the compartment **64** in FIGURE 11. The supply of the oil may be by means of a suitably located oil jet (not shown) similar to oil jet **86** in FIGURE 1 or by the rotor bearing spill as also described in connection with FIGURE 1 and FIGURES 6 and 7. As stated in relation to FIGURES 3A–3G the oil moves counterclockwise in the compartments **64** and when the acceleration forces are directed inward the oil will tend to flow inward in the compartments along the inner surface of the rotor end walls and will drain from the compartments through openings **120** into the annular gutter **88** and finally out through suitable drain passages **90** as shown by the arrow in FIGURE 11. As can be seen that from this embodiment, separate openings may be provided for supplying and draining the oil in each compartment.

The embodiment of FIGURE 12 is similar to that of FIGURE 11. However in this embodiment the internally toothed gear **124** is formed as an integral part of the rotor hub portion **70**, instead of an integral part of the adjacent rotor end wall, as in FIGURE 11. However, the rotor internal gear **124** has an inner diameter which is less than the outer diameter of the rotor hub. The fixed gear **42** meshes with the gear **124** and has an axially extending shank portion with openings **126** to permit oil to be supplied radially outwardly therethrough by a suitable oil supply means. For example an oil jet (not shown) similar to the oil jet **86** may be used. The arrow pointing into compartment **64** of FIGURE 12 indicates the path of the oil supply through openings **126**, and into compartment **64** by means ofa gap between the rotor end wall and hub portion which is similar to gap **78** in FIGURE 1. The arrow leading out of compartment **64** indicates the drainage path for the oil through the gap and out of the rotor in the same manner as explained above.

In FIGURE 13 there is shown an embodiment in which the internally-toothed gear or rotor gear **128** is formed integral with the web members **66**. In this embodiment openings **130** to each compartment, for the supply of oil by suitable means are formed between the gear **128** and the rotor hub **70** as indicated by the arrow heading into compartment **64** shown in FIGURE 13. Drainage openings **132** are formed between the gear **128** and the rotor end wall so that the oil will drain through said openings **132** as shown by the arrow leaving compartment **64** in FIGURE 13. Again, as in the case of FIGURES 11 and 12 the gear **128** has a smaller diameter than the hub **70** but does not restrict the flow of oil to and from the compartments **64** due to the openings provided for said flow of oil. Thus, it can be seen in the embodiments of FIGURES 11–13 that the internally-toothed gear may be made integral with the rotor and with a diameter smaller than the outer diameter of the rotor hub and suitable separate openings may be provided adjacent the radially inner portion of the compartments for the flow of oil to and from said compartments.

From the above detailed description it can be seen that the acceleration forces produced due to the eccentric rotation of the rotor are used to provide a novel cooling means for the rotor by causing the coolant to be impinged on the inner surfaces of said rotor and also for draining the rotor of said coolant. The coolant may be said to be circulated along the walls of the rotor compartments and pumped out of the compartments by the forces produced by the rotor itself such that the rotor interior is thereby never completely full of coolant. It can also be seen that the motion of the oil in the rotor is accomplished by a minimum amount of work due to the utilization of the forces produced in the rotor during rotation and no special mechanism need be added to ensure the flow of oil in and out of the rotor. Furthermore, due to the fact that only a relatively small amount of oil is used, the turbulence of the oil or its "cocktail shaking" effect in the rotor with resultant substantial power consumption required to circulate the oil is substantially eliminated so that little, if any, power is lost in circulating the oil in the rotor.

Another advantage resulting from the present invention is that the efficiency of the cooling system increases as the speed of the engine increases. This is highly desirable in that at higher engine speeds the engine produces more heat and more efficient cooling is required while at low engine speeds the cooling is usually not as critical since the heating-up of the engine is normally less than at higher speeds. In the present invention, at higher engine speeds the oil is circulated proportionately faster due to the faster changing of the acceleration forces and flow of oil into and out of the compartments resulting in a substantially complete changing of the oil in the rotor proportionate to the speed of the rotor. The heated oil will therefore flow out of the rotor faster at higher speeds and likewise fresh oil will be introduced faster which provides for better heat transfer and higher cooling efficiency. Furthermore, better cooling is provided in that small amounts of fresh cooling oil are thrown against substantially bare metal surfaces of the compartments which results in a better transfer of heat from the metal rotor to the cooling oil and consequently provides a cooler running rotor. It should also be noted that each of the means disclosed for providing cooling oil to the compartments may be used interchangeably or in combination in each of the rotors disclosed. For example, an oil flat feed such as disclosed in FIGURES 6 and 7 may be used with the rotor disclosed in FIGURE 1 instead of the jet nozzle or in combination therewith and the same is true of the rotors disclosed in FIGURES 8, 10 and 11–13. Also in the embodiments wherein jet nozzles are used the invention is not to be limited to the number or type of nozzles used. Although in each of the embodiments illustrated the devices have been shown as having intermeshing gears on only one side thereof, the invention also contemplates the use of intermeshing gears on both sides, if desired, and the rotor gears may be made integral with the rotor or separately attached thereto.

While the invention has been specifically set forth in detail in the above description, the invention is not to be so limited thereby and various modifications and alterations may be made by those skilled in the art without departing from the spirit and scope of the invention defined in the following claims.

We claim:

1. In a rotary mechanism for fluid pumps, fluid motors, combustion engines or the like comprising an outer body with a cavity therein, a rotable shaft mounted in said outer body co-axial with the axis of the peripheral wall of the outer body cavity and having an eccentric portion thereon, a rotor journalled on said eccentric portion for rotation about its axis while said rotor axis describes a planetary motion relative to the axis of said outer body whereby each point on said rotor is subjected to acceleration forces which successively change in direction relative to the axis of said rotor, said rotor having a peripheral surface forming a plurality of circumferentially-spaced apex portions for sealing engagement with the inner surface of said peripheral wall and thereby forming working chambers between the rotor peripheral surface and the peripheral wall of said cavity which upon relative rotation of said outer

body and said rotor vary in volume, said rotor having an outer wall including a hub portion enclosing a hollow interior and having a plurality of circumferentially-spaced, generally-radial partitions therein forming compartments in said rotor, means including openings through the rotor outer wall and disposed adjacent to said hub portion at the radially inner portions of said compartments for supplying a liquid coolant sequentially into said rotor compartments and for sequentially draining said coolant from said rotor compartments as the rotor rotates about its axis such that each compartment receives a major portion of its liquid coolant during a portion of each revolution of the rotor and each compartment drains a major portion of said coolant during other portions of each revolution of the rotor.

2. In a rotary mechanism as recited in claim 1 wherein said partitions extend radially inwardly at least substantially to said openings.

3. In a rotary mechanism as recited in claim 1 wherein said means for supplying coolant to said cavities includes a coolant supply passageway in said shaft and means at the discharge end of said passageway for directing said coolant outwardly therefrom toward said compartment openings.

4. In a rotary mechanism as recited in claim 3 wherein said means for directing said coolant comprises a nozzle for directing a jet of coolant toward said compartment openings.

5. In a rotary mechanism as recited in claim 4 wherein said cavity has two lobes and said rotor has three circumferentially-spaced apex portions and said nozzle is located in a range from 0 degree to 100 degrees about the shaft axis as measured in the direction of rotor rotation from the point of maximum eccentricity of the shaft eccentric portion.

6. In a rotary mechanism as recited in claim 4 wherein said cavity has two lobes and said rotor has three circumferentially-spaced apex portions and said nozzle is located approximately 70 degrees about the shaft axis as measured in the direction of rotor rotation from the point of maximum eccentricity of the shaft eccentric portion.

7. In a rotary mechanism as recited in claim 3 wherein said means for directing said coolant toward said compartments comprises a deflecting member, said deflecting member being positioned adjacent to the outlet of said passages and being formed such that when the coolant supplied from said passages strikes said deflecting member said coolant is deflected toward said compartments.

8. In a rotary mechanism as recited in claim 1 wherein said rotor outer wall comprises a peripheral wall interconnected with a pair of end walls and said rotor hub portion, said rotor hub portion connected with said end walls at their inner ends and said end walls and said rotor compartments each having opening means for permitting said coolant to flow in and out of said compartments, each said compartment opening means being located in the region of the radially inner end of the adjacent end wall portion of the rotor and the adjacent axial end of the rotor hub with said opening means being disposed to permit both radial and axial flow into and out of the compartments.

9. In a rotary mechanism as recited in claim 1 further comprising a plain sleeve-type bearing member disposed between said hub portion of said rotor and said shaft eccentric for rotatively supporting the rotor on the shaft eccentric, passage means for supplying lubricating oil to said bearing member such that said lubricating oil flows out from at least one end of the bearing member and is thrown out by centrifugal force into said rotor compartments to function as said liquid coolant.

10. In a rotary mechanism as recited in claim 9 and in which said shaft eccentric is provided with a passageway between said eccentric and said bearing member and running to at least one end of the bearing member, said passageway being formed such that said lubricating oil is directed radially outwardly from at least one end of said passageway.

11. In a rotary mechanism as recited in claim 10 wherein said cavity has two lobes and said rotor has three circumferentially-spaced apex portions and said passageway formed on said eccentric is located in a range of 30 degrees to 100 degrees about the shaft axis as measured in the direction of rotor rotation from the point of maximum eccentricity of the shaft eccentric portion.

12. In a rotary mechanism as recited in claim 10 wherein said cavity has two lobes and said rotor has three circumferentially-spaced apex portions and said passageway formed on said eccentric is located at approximately 70 degrees about the shaft axis as measured in the direction of rotor rotation from the point of maximum eccentricity of the shaft eccentric portion.

13. In a rotary mechanism as recited in claim 1 wherein the profile of said outer body cavity is a multi-lobed configuration, said rotor compartments each being supplied with said coolant a plurality of times during each complete rotation of said rotor and drained a plurality of times during each complete rotation of said rotor, said supplying and draining of said compartments each occurring in proportion to the number of lobes in said cavity.

14. In a rotary mechanism as recited in claim 1 wherein the profile of said outer body cavity preferably is basically a two-lobed epitrochoid and said rotor has three circumferentially-spaced apex portions, said rotor compartments each being supplied with said coolant twice and at least partially drained twice during each complete rotation of said rotor.

15. In a rotary mechanism as recited in claim 1 wherein said rotor compartments are further defined by a pair of rotor end walls, a rotor peripheral wall and a transverse wall disposed across said partitions and parallel to said end walls, said coolant being supplied to said compartments at both sides of said rotor.

16. In a rotary mechanism as recited in claim 15 wherein said means for supplying coolant to said compartments includes first passage means in said rotatable shaft for transporting said coolant toward said rotor, second passage means connected to said first passage means with said second passage means having an opening disposed at each side of said rotor and directed toward the openings to said rotor compartments such that said coolant is sequentially supplied to each side of the rotor compartments as the rotor rotates relative to said shaft.

17. In a rotary mechanism as recited in claim 1 wherein said outer body has an externally-toothed gear secured thereto and co-axially disposed about the shaft and said rotor has an internally-toothed gear secured to and in meshing engagement with said externally toothed gear, said rotor gear being disposed at one end face of the rotor adjacent to the rotor inner periphery and having a rim portion extending axially toward the rotor from its gear teeth for attachment to the rotor, said gear rim portion having a plurality of cut-out portions in its inner side in substantial alignment with said compartment openings whereby said liquid coolant may flow from said supply means through said cut-out portions and into said compartments and drained out of said compartments through said cut-out portions.

18. In a rotary mechanism as recited in claim 1 wherein at least one closed compartment is provided at the outermost region of each apex portion of the rotor, said closed compartment containing a low melting point metal sealed therein.

19. In a rotary mechanism as recited in claim 1 wherein said means including openings through the rotor outer wall comprises a plurality of openings disposed adjacent the radially inner portion of each compartment, and wherein substantially all of the coolant in each compartment is supplied therein through at least one of said openings and is drained therefrom through at least one other of said openings.

20. In a rotary mechanism as recited in claim 19 wherein said rotor outer wall comprises said hub portion and a peripheral wall interconnected with a pair of end walls, said rotor also having an internally-toothed gear member secured on at least one side thereof and one of said openings in said compartment being disposed between said gear and an end wall of said rotor and another of said openings in said compartment being disposed between said gear and said hub portion of said rotor.

21. In a rotary mechanism as recited in claim 1 wherein said rotor outer wall comprises said hub portion and a peripheral wall interconnected with a pair of end walls, an internally-toothed gear member secured to at least one side of said hub portion, an externally-toothed gear member secured to an end wall of said outer body and having an extending shank portion surrounding said shaft for meshing engagement with said internally-toothed gear member, at least one opening for each of said compartments being disposed between said internally-toothed gear member and one of said rotor end walls and at least one other opening disposed in said extending shank portion of said externally-toothed gear member.

22. In a rotary mechanism for fluid pumps, fluid motors, combustion engines or the like comprising an outer body having a cavity, a shaft co-axial with said cavity and having an eccentric portion, a hollow rotor disposed within said cavity and having a hub portion journaled on the shaft eccentric portion and having a peripheral wall disposed radially outwardly of the hub portion, the rotor peripheral wall having a plurality of circumferentially-spaced points having sealing cooperation with the inner surface of the cavity peripheral wall to form a plurality of working chambers between the rotor and outer body which vary in volume upon rotation of the rotor relative to the outer body, said rotor also having axially-spaced end walls extending radially inwardly from its peripheral wall and circumferentially-spaced webs extending inwardly from said peripheral wall between said end walls to divide the interior of the rotor into a plurality of circumferentially-spaced compartments, at least one of the ends of the rotor hub portion terminating short of the adjacent rotor end wall and the inner diameter of said end wall being greater than the outer diameter of the adjacent end of its rotor hub portion to leave an annular gap therebetween communicating with said compartments and forming an opening therefor; cooling means for said rotor comprising means for causing lubricating oil to flow sequentially into said rotor compartments through primarily a portion of said annular gap and means for at least partially draining each of said compartments through said gap periodically during each revolution of said rotor.

23. In a rotary mechanism as recited in claim 22 further comprising an annular transverse wall generally perpendicular to the rotor axis and extending from said hub to said peripheral wall and intersecting said webs, said annular transverse wall extending over the entire circumference of said rotor and thereby dividing said compartments.

24. In a rotary mechanism as recited in claim 22 wherein both ends of said rotor hub portion terminate short of the adjacent rotor end walls and the inner diameter of each end wall is greater than the outer diameter of the adjacent end of the rotor hub portion to leave an annular gap on each side of said rotor in communication with said compartments whereby oil may be supplied to a drain from the compartments on both sides of said rotor.

25. In a rotary mechanism as recited in claim 23 further comprising ports in said annular transverse wall adjacent the outer periphery of said wall providing openings between compartments on opposite sides of the wall whereby, when said oil flows into said compartments, said oil may flow therebetween thereby tending to equalize the amounts of oil on each side of said rotor.

26. In a rotary mechanism as recited in claim 22 wherein said outer body has an externally-toothed gear secured thereto and said rotor has an internally-toothed gear in meshing engagement with said externally-toothed gear, said rotor gear having an effective diameter greater than the adjacent part of the rotor hub portion whereby said flow of oil in and out of said compartments through said gap is substantially unobstructed by said rotor gear.

27. In a rotary mechanism as recited in claim 23 wherein said means for causing lubricating oil to flow sequentially into said rotor compartments includes passage means in said shaft having an opening disposed at each side of said rotor with said passage means openings being directed toward an annular gap at each side of said rotor for supplying lubricating oil to each side of said rotor compartments.

28. In a rotary mechanism for fluid motors, fluid pumps, combustion engines and the like having an outer body with a cavity therein, a shaft mounted in said outer body co-axially with said cavity and having an eccentric portion thereon, a hollow rotor, a plain sleeve type bearing disposed between said rotor and shaft eccentric portion for rotatably mounting said rotor on said eccentric portion for rotating of said rotor about its axis while said rotor axis describes a planetary motion relative to the axis of said outer body whereby acceleration forces are generated which periodically reverse in direction relative to said rotor, said rotor having an outer wall including a peripheral wall forming a plurality of circumferentially-spaced apex portions for sealing engagement with the inner surface of said outer wall, a pair of end walls interconnected with said peripheral wall and a rotor hub portion forming a cavity therein, a plurality of web members extending between said rotor walls and hub portion thereby forming a plurality of circumferentially-spaced compartments in said rotor cavity each of said compartments having opening means therein adjacent their inner ends; means for supplying lubricating oil to said bearing to a region intermediate the ends of the bearing for flow axially therefrom and out the bearing ends from which said lubricating oil is thrown radially outwardly through said end wall openings into each of said compartments during at least predetermined rotative positions of said rotor wherein during further rotation of said rotor said lubricating oil circulates within said compartments and the acceleration forces causing said lubricating oil to be periodically at least partially drained from said compartments through said openings before said rotor rotates to said predetermined rotative position during each revolution of said rotor.

29. A rotor for a rotary mechanism having a hollow outer body, a shaft coaxial with said outer body and having an eccentric portion on which the rotor is to be journaled for rotation within said hollow outer body to form a plurality of variable volume working chambers between said rotor and the inner surface of said outer body; said rotor comprising a peripheral wall interconnected with a pair of axially-spaced end walls and a hub journal portion defining a cavity between said walls and said hub portion, said rotor peripheral wall having a plurality of circumferentially-spaced apex portions for sealing cooperation with an outer body, said rotor also having a plurality of circumferentially-spaced webs extending radially inwardly from said rotor peripheral wall to said hub portion between said rotor end walls to interconnect said walls and hub portion and to divide said rotor cavity into a plurality of circumferentially-spaced compartments, said rotor end walls extending radially inwardly from said rotor peripheral wall and at least one of said rotor end walls and the adjacent end of said rotor hub portion terminating short of each other thereby defining an annular gap between said hub portion and said end wall communicating with said rotor compartments and forming an opening therefor so that a coolant may be supplied

and drained to and from each of said rotor compartments through said annular gap.

30. A rotor for a rotary mechanism as recited in claim 29 wherein each of said end walls and said rotor hub portion define an annular gap on each side of said rotor communicating with said rotor compartments, said rotor including an internal gear having a shank portion secured to one side of said rotor for determining the rotative position of the rotor, said internal gear shank portion being provided with openings therethrough for permitting coolant to flow substantially unobstructed in and out of said compartments between said internal gear and said rotor and through the annular gap on the gear side of said rotor.

31. A rotor for a rotary mechanism as recited in claim 29 further comprising an annular transverse wall generally parallel to said rotor end walls and positioned intermediate thereof, said annular transverse wall extending from said hub portion to said rotor peripheral wall and intersecting each of said webs to divide said rotor compartments.

32. A rotor for a rotary mechanism as recited in claim 29 wherein one wall of said rotor hub portion forms the radially inner wall for each of said rotor compartments and said one wall extending axially across said rotor with said one wall being sloped radially inwardly in a direction toward at least said one rotor end wall for preventing a coolant from collecting at the radially inward portion of each of said compartments, said one rotor end wall having a radially inner diameter greater than the radially outer diameter of said rotor hub portion and said rotor hub portion adjacent said one rotor end wall terminating short of said one rotor end wall in an axial direction.

33. A rotor for a rotary mechanism having a hollow outer body, a rotatable shaft coaxial with said outer body on which the rotor is to be journaled for rotation within said hollow outer body and having an eccentric portion; said rotor comprising a peripheral wall interconnected with a pair of parallel axially-spaced end walls and including a hub journal portion defining a cavity within the rotor between said walls and hub portion, said rotor peripheral wall having a multi-lobed configuration with a plurality of circumferentially-spaced apex portions for sealing cooperation with the inner surface of said outer body, a plurality of circumferentially-spaced webs extending radially inwardly from said rotor peripheral wall to said rotor hub portion and axially between said rotor end walls to interconnect said walls and hub portion and to divide a plurality of circumferentially-spaced compartments in said rotor cavity, and said rotor including openings in the outer surface thereof to said compartments with said openings being located closer to the radially inner portion of said compartments than the radially outer portion so that during rotation of said rotor a coolant may be supplied through said openings to said compartments and substantially completely drained therefrom through the same openings.

34. A shaft assembly for a rotary mechanism having a hollow outer body and a multi-lobed rotor in sealing engagement with the inner surface of said outer body to form a plurality of variable volume working chambers, said rotor having a plurality of circumferentially-spaced compartments therein having openings adjacent their radially inner portions for supply and discharge of a coolant to and from each of said compartments; said shaft assembly comprising, a rotatable shaft having an eccentric portion thereon, passage means in said shaft for transporting a coolant therethrough and passage means in said eccentric portion extending from said shaft passage means to each side face of said eccentric portion and nozzle means positioned in the end of said eccentric portion passage means at each side face of said eccentric portion with said nozzle means being oriented for discharging a jet of coolant radially outwardly relative to said shaft axis and said nozzle means being located in a range of 0 degree to 100 degrees about the shaft axis as measured in the direction of rotor rotation from the point of maximum eccentricity of the shaft eccentric portion.

References Cited by the Examiner

UNITED STATES PATENTS

3,007,460	11/61	Bentele et al.	230—210
3,016,184	1/62	Hart	230—210
3,091,386	5/63	Paschke	230—210

References Cited by the Applicant

UNITED STATES PATENTS

3,102,682	9/63	Paschke.

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