Aug. 10, 1965 V. BUSH ETAL 3,199,460

HYDRAULIC PUMP OR MOTOR

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FIG. 1.

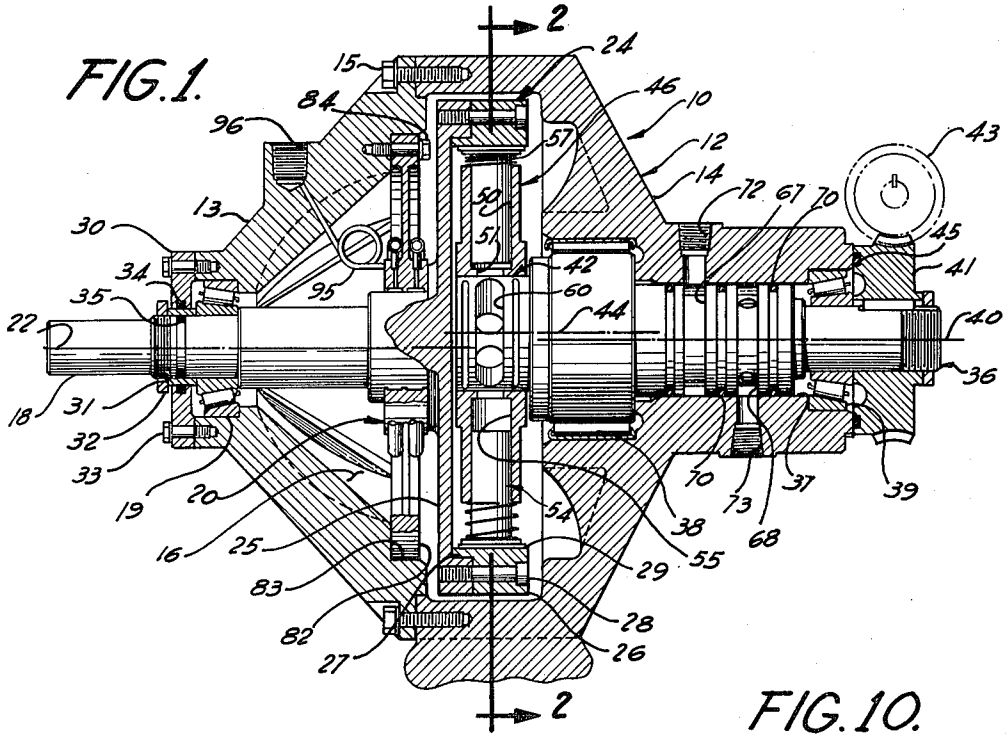


FIG. 10.

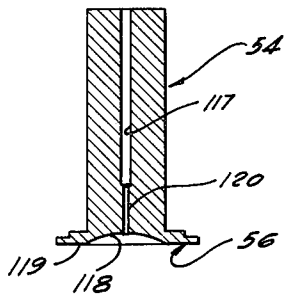
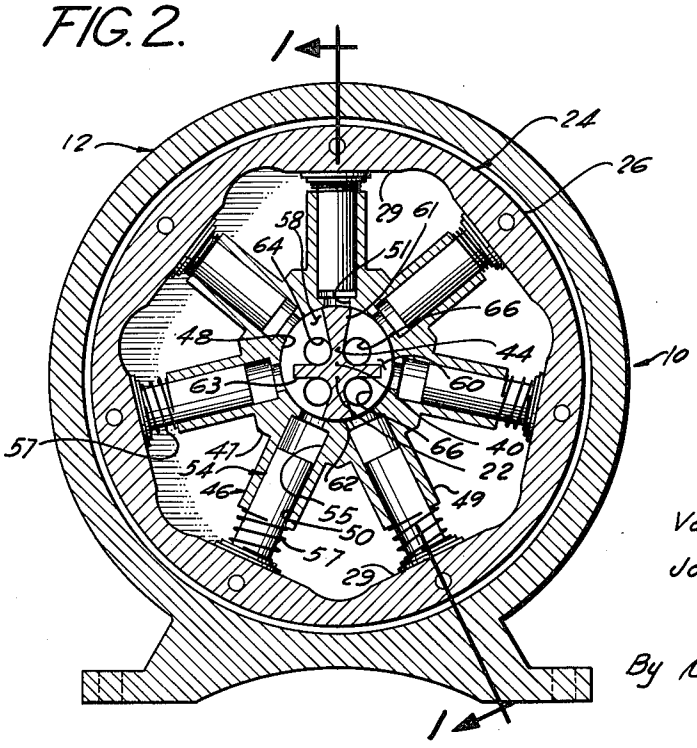


FIG. 2.



INVENTORS

Vannevar Bush

John A. Hastings

By A. G. Dowrat.

Attorney.

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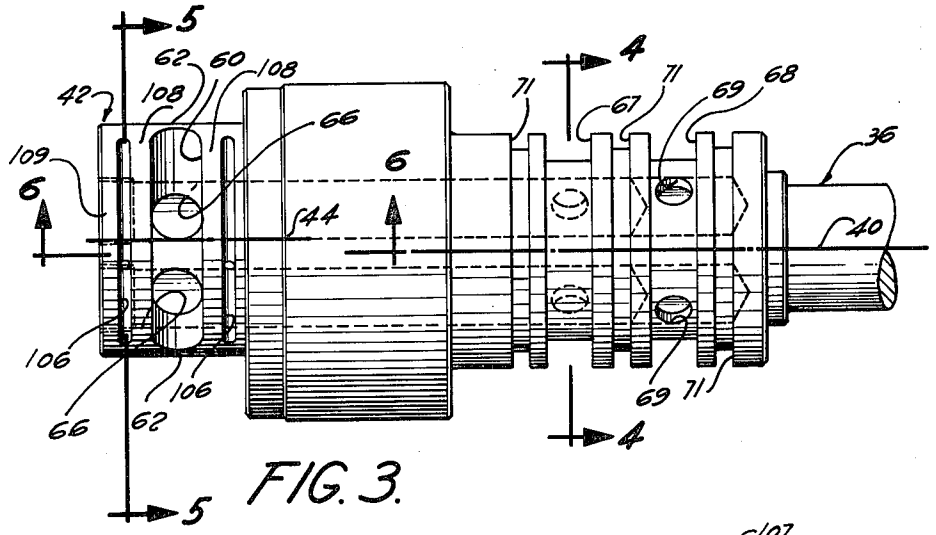


FIG. 3.

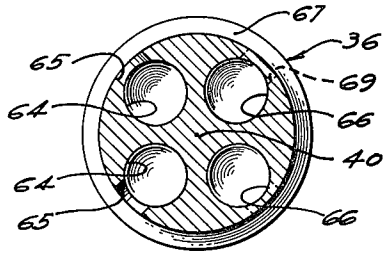


FIG. 4.

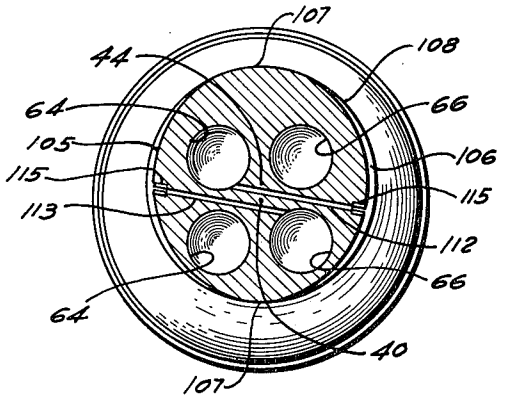


FIG. 5

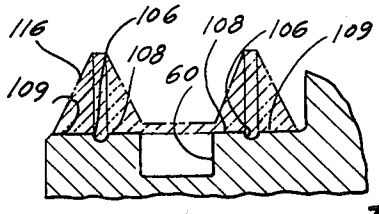
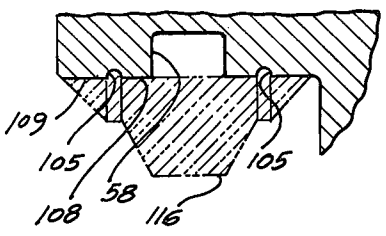


FIG. 6.



INVENTORS

Vannevar Bush

John A. Hastings

By A. G. Dowrat

Attorney

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HYDRAULIC PUMP OR MOTOR

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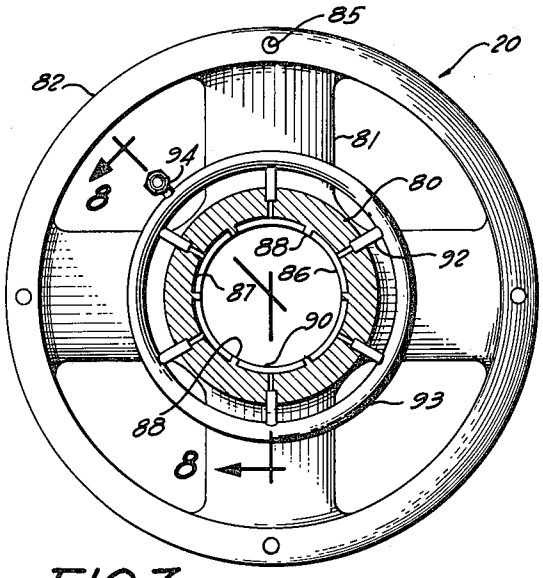


FIG. 7.

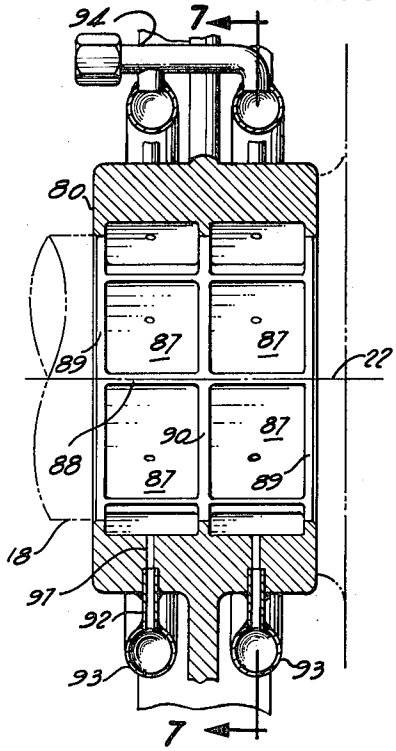


FIG. 8.

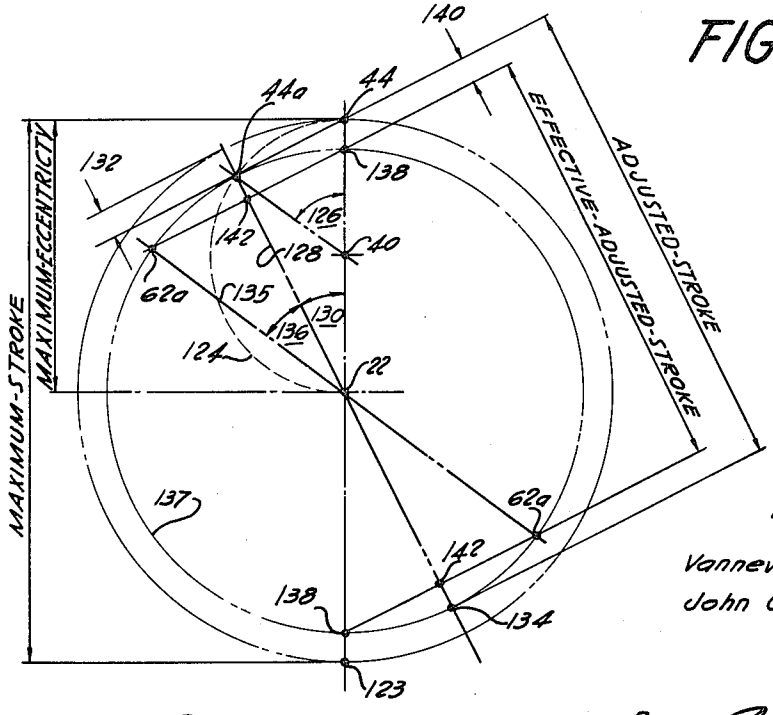


FIG. 9.

INVENTORS

Vannevar Bush

John A. Hastings

By A. G. Dowad.

Attorney.

United States Patent Office 3,199,460 Patented Aug. 10, 1965

3,199,460

HYDRAULIC PUMP OR MOTOR

Vannevar Bush, Belmont, and John A. Hastings, Bass River, Mass., assignors to Stewart-Warner Corporation, Chicago, Ill., a corporation of Virginia

Filed Jan. 11, 1962, Ser. No. 165,685
6 Claims. (Cl. 103—161)

This invention relates to hydraulic pump or motor units of the type having radially disposed fluid chambers each defined by a mating piston and cylinder intermediate to opposing reaction elements.

A major problem of any hydraulic pump or motor unit is film lubrication between adjacent mating members adapted to slide along one another, while simultaneously being pressed together with loads of large magnitude. The load supporting capacity of the film lubrication must be adequate to eliminate metal-to-metal contact at the maximum load and at any intermediate loads. Similarly, throughout all outputs and speeds of the unit, the power consumption caused by leakage or friction of the film must be maintained at a tolerable minimum. The success of the film lubrication, however, depends on a second problem, the problem of withstanding cocking forces or couples applied between the mating members.

A couple between two members separated from each other by a fluid film causes tilting of the members relative to one another until balanced by a resisting couple of equal magnitude in the opposite direction. This resisting couple is developed initially by a shift asymmetrically of the film thickness and pressures, causing localized pressure concentrations of the film. These localized pressure concentrations upon exceeding the maximum allowable film pressures, degenerate to localized metal-to-metal contacts. Thereafter any deficiency of the film support is made up by the direct metal-to-metal support. Consequently, any couple between mating members that causes localized pressure concentrations within the separating film to exceed the maximum allowable film pressure is extremely harmful to the unit and greatly shortens the operating life of the unit.

In a piston-type unit, the fluid pressure confined by the mating piston and cylinder exert an outward force on the piston in the direction of the longitudinal axis of the cylinder. The outward force is absorbed by spaced reaction elements opposing the piston and cylinder. The reaction elements are moved toward and away from one another by appropriate structure to transform between the energy of fluid pressure and work. A shaft member generally is associated with one of the members to harness the work in the form of rotating shaft torques. The shaft torque must be opposed by an equal and opposite reaction torque through the housing structure of the unit, which includes the reaction elements.

It is commonplace in existing piston-type units that the pressure force acting along the longitudinal axis of the cylinder always extends through the center of the shaft. A torque about a point is represented as a force acting in a direction at some normal distance or moment arm from the point, and is the product of the two. When the pressure force extends through the center of the shaft, it thus has no direct moment arm about the shaft. Thus the only way transformation between fluid pressure force and shaft torque can occur is by indirectly applied forces such as a couple between the adjacent mating members. Since this couple must equal the shaft torque and thus be of large magnitude, film support fails causing metal-to-metal contact.

It is also commonplace in conventional radial piston hydraulic units that the shaft is directly connected to the cylinder or member rotating on one of the reaction elements. The other reaction element is mounted for free

rotation about an axis spaced from the shaft axis. Rotation of the shaft only indirectly, through the interaction of the other members including the piston and cylinder, causes the movement of the reaction elements and thus reciprocation of each piston and cylinder. This further requires that couples be brought in play to transform between the pressure forces and shaft torque.

There is further a greatly increasing demand to use hydraulic equipment at pressures exceeding 5,000 p.s.i. and flow rate approaching 200 g.p.m. Also, to reduce the ratio of overall unit weight to output horse power, operating shaft speeds of 5,000 r.p.m. or higher have been tried and achieved in some commercial embodiments. However, under these severe operating conditions, the aforementioned problems relating to cocking of, and film support between, adjacent mating members become increasingly complex and important, and more often the direct cause of failure.

The bearing structure supporting the various rotating members must be capable of withstanding the large radial loads applied to the members without excessive torque loss because of friction. Centilever application of these radial loads causes slight transverse deflection and/or axial misalignment of the shaft. Thus self-aligning bearing means capable of compensating these tendencies must be used; particularly in a film lubrication bearing where mating surfaces must remain parallel.

Another practical drawback of existing commercial hydraulic pump or motor units is complex and expensive adjusting structure commonly used for varying the fluid flow per cycle of the unit. The limitations of commercial manufacturing plus the continuing battle to reduce costs require an easily produced structure capable of accurate and dependable operation.

Accordingly, an object of this invention is to provide a hydraulic pump or motor unit of the radial piston and cylinder type with improved disposition of the various members relative to each other to eliminate reaction couples between adjacent members caused directly by the confined fluid pressures.

Another object of this invention is to provide a hydraulic pump or motor unit having fluid film lubrication between all adjacent mating surfaces adopted to slide along one another while simultaneously being forced together by a large load, the fluid lubrication being pressurized from a high pressure source through a fluid restrictor to establish and maintain a balanced film at all operating conditions of the unit.

Another object of this invention is to provide a hydraulic film bearing capable of supporting a rotating shaft member subjected to a large radial load while accommodating moderate angular misalignment of the longitudinal shaft axis from its normal axial position.

Another object of this invention is to provide a hydraulic pump or motor unit having simple economical structure including a double eccentric mounting operable to varying the fluid flow per cycle of the unit.

These and other objects will be more fully appreciated after a complete disclosure of the subject invention given in the following specification and the accompanying drawings forming a part thereof, wherein:

FIGURE 1 is a longitudinal section view of the subject invention as embodied in a hydraulic pump or motor unit, the view being taken generally from line 1—1 of FIGURE 2;

FIGURE 2 is a front section view of the subject invention as taken generally from line 2—2 of FIGURE 1;

FIGURE 3 is a side elevational view, similar to that shown in FIGURE 1, of a pintle-crank member used in the subject invention;

FIGURE 4 is a section view as taken from line 4—4 of FIGURE 3;

FIGURE 5 is a section view as taken from line 5—5 of FIGURE 3;

FIGURE 6 is an enlarged section view as taken from line 6—6 of FIGURE 3, the figure including operating fluid pressure forces and distributions on the member;

FIGURE 7 is a front elevational view, partly shown in section from line 7—7 of FIGURE 8 and of reduced size compared thereto, of a self-aligning film bearing unit of the subject invention;

FIGURE 8 is an enlarged section view as taken generally from line 8—8 of FIGURE 7;

FIGURE 9 is a diagrammatic representation of the adjusting structure of the subject invention, the view being a greatly enlarged elevational view similar to FIGURE 2; and

FIGURE 10 is a longitudinal center section view of a piston used in the subject invention.

Referring now to FIGURES 1 and 2 of the drawings, the disclosed hydraulic unit 10 includes a housing 12 formed by adjacent cup-shaped members 13 and 14 secured together at their periphery by bolts 15 and defining an internal cavity 16. A drive shaft 18 is supported by bearing units 19 and 20 to rotate about its longitudinal axis 22. A cage 24 secured to the shaft is formed by a web 25 and an annular shoe ring 26 held thereto by engaging shoulders 27 and bolts 28. A plurality of inwardly facing shoe surfaces 29 are disposed on ring 26 symmetrically about the center axis 22. The cage 24 and shoe surfaces 29 rotate with shaft 18 about the longitudinal center axis 22, which can also be considered as the center axis of the unit 10.

Annular end plate 30 is received over spacer ring 31 on the shaft 18 between nut 32 and the inner race of bearing unit 19 and is secured to housing 12 by bolts 33. Seals 34 and 35 between the moving shaft, spacer and housing prevent fluid leakage from the cavity 16.

A pintle element 36 (FIGURE 3) is supported within bore 37 of the housing 12 by spaced bearing units 38 and 39 to rotate about its longitudinal axis 40. Axis 40 is parallel to but offset from the longitudinal axis 22 of the shaft 18. Conventional means including a worm gear 41 keyed to pintle 36 and engaged by a driving pinion 43 (shown in phantom) operate to rotate the pintle. O-ring 45 between gear 41 and housing 12 seal the bore 37 as required. The pintle 36 has a cylindrical crank portion 42 disposed about a longitudinal center axis 44 parallel to but offset from the pintle axis 40 by the same offset as between the pintle axis 40 and the shaft axis 22.

Cylinder member 46 has a hub 47 defining a through-bore 48 which matably receives crank 42. The cylinder member 46 has a plurality of radially extending projections 49 each having an internal cylindrical opening defining a cylinder 50. An opening 51 extends between the periphery of through-bore 48 and the cylinder 50 and is approximately one-half the area of the cylinder. A cylindrical piston 54 (FIGURE 10) is received matably within each cylinder 50 and defines therewith a fluid chamber 55 communicating with the openings 51. The piston 54 has an outwardly disposed bearing surface 56 matable with the shoe surface 29 on the cage 24. Light compression springs 57 hold the pistons 54 against the cage 24 when the unit is not operating.

The crank 42 (FIGURE 3) includes opposing slots or ports 58 and 60 axially aligned with the through openings 51 in the cylinder member 46. The ports 58 and 60 are separated from one another by a rib 61 extending across the crank 42 and terminating at the outer periphery thereof on spaced lands 62 each of a given lap. A strengthening rib 63 extends transversely of the ports 58 and 60 to a position short of the periphery of the crank. Thus rotation of the cylinder member 46 on the crank 42 exposes each opening 51 alternately to the ports 58 and 60.

The pintle 36 has spaced bores 64 and 66 extending

axially thereof and communicating respectively with the opposing ports 58 and 60. Bores 64 communicate through radial passage 65 with annular recess 67, and bores 66 communicate through radial passages 69 with annular recess 68. The recesses 67 and 68 are sealed from each other by O-ring gaskets 70 received in grooves 71 and engaging the periphery of bore 37. Radial threaded taps 72 and 73 in housing 12 communicate respectively with annular recesses 67 and 68 for separate connection to the high pressure and low pressure fluid sources, as is well known in the art.

The subject hydraulic unit 10 operates in a manner that eliminates the reaction couples between adjacent members caused by transformation between fluid pressure in the chambers 55 and rotation of the shaft 18. The cage 24 and crank 42 form the reaction elements opposing the piston 54 and cylinder 50 (within cylinder member 46) on opposite sides of chamber 55. The cylinder member 46 rotates freely on the crank 42 at the same rate as the shaft 18 and cage 24 through the interactions of the mating of pistons 54 on the cage 24. The cylinder member 46 follows freely since it acts only to confine the fluid pressures in chambers 55 and to communicate the chambers alternately through the ports 58 and 60 with the high and low pressure fluid sources.

As noted above, the shaft 18, the cage 24, and the spaced shoe surfaces 29 all have a common center axis 22 which is parallel to but offset by some eccentricity from the center axis 44 of the crank 42. During one revolution of shaft 18 and cage 24, and thus of the cylinder member 46, the distance normal to each shoe surface 29 between said shoe surface and the crank center axis 44 varies between a maximum and a minimum. This variation in radial distance causes the reciprocation of the pistons 54 within the cylinders 50. It is apparent that if a plane were extended through the spaced center axes 22 and 44, the top dead centers of the pistons 54 in the cylinders 50 would occur on that plane, the maximum and minimum being 180° apart. Accompanying this radial reciprocation relative to the cylinder 50, each piston 54 translates to each side of its top dead center position relative to shoe surface 29 along the shoe surface a distance equal to the eccentricity of the unit.

The fluid pressure confined within each chamber 55 acts along the longitudinal center axis of cylinder 50, which axis extends through the center axis 44 of crank 42 and the centroid of piston bearing surface 56. At all positions other than the top dead center positions, this axial force is displaced from the center axis 22 of the shaft 18 and cage 24 to exert a turning moment or torque on the cage about its axis. The cage is one of the reaction elements and is also directly connected to the shaft for common related movement therewith. The cylinder member 46 follows the rotation of cage 24 freely and without resistance other than minor film friction losses. This offset application of the axial force directly on the moving reaction member, that is the driving or driven cage 24 connected to shaft 18, transforms between the fluid pressure forces and mechanical shaft torques. Furthermore, regardless of whether the unit is working as a pump or motor, or whether the shaft is rotating in one direction or the other, the transformation occurs directly, without reaction couples between the members caused by the fluid pressure.

It will be apparent that a cocking force of minor magnitude will be present, caused by fluid friction in the films between the adjacent members. However, the magnitude of this force and the resulting couple is negligible compared to the couple equal to the full shaft torque caused by operating fluid pressures; a couple which this invention eliminates.

FIGURES 7 and 8 show the fluid film bearing 20 in greater detail. The bearing consists of a hub 80 connected centrally at circumferentially spaced portions by ribs or spokes 81 to an outer rigid rim 82. The rim 82

is received within annular recess 83 of the housing 12 and held therein by a plurality of circumferentially spaced bolts 84 extending through apertures 85. The hub 80 has cylindrical centerbore 86 from .0005 to .002 of an inch larger than the cylindrical shaft 18 received therein, depending on shaft diameter. Thus there is radial clearance of approximately .00025 to .001 of an inch symmetrically around the shaft.

The inner periphery of the bore 86 has thereon a plurality of uniformly spaced recesses 87. The bearing shown is only by way of exemplification in which six equally spaced recesses 87 are disposed around the bore 86, each recess 87 being separated by axial land areas 88 and by circumferential outer land areas 89 and an intermediate land area 90. The bearing shown has two such axially spaced bearing sections separated from one another by the circumferential land area 90.

Each recess 87 has a radial bore 92 communicating therewith and with a source of high pressure hydraulic fluid confined within annular distributing tubes 93. Manifold 94 connected to tubes 93 communicates with tap 96 through flexible tubing 95. The tap 96 is maintained under high fluid pressure comparable to that of the unit 10, and can in fact be taken from the same source by a double check valve (not shown) connected, for example, between tap 96 and both taps 72 and 73. A restrictor 97 of fixed size orifice is secured in each bore 92.

The basic operation of the subject film bearing 20 is similar to those commonly known as hydrostatic bearings. The high pressure source of fluid at tap 96 continually presents high pressure through the radial passage 92 and the restrictor 97 to the circumferentially spaced recesses 87. When shaft 18 is centered in bore 86, the clearances from each recess between axial land areas 88 and circumferential land areas 89 and 90 to the adjacent recess or to cavity 16 are all uniform and, therefore, the fluid pressures in the recesses 87 are maintained generally constant.

When, however, a radial load is applied to shaft 18 to cause displacement of the shaft within the bore 86, the clearances between the confining land areas and the shaft in the direction opposing the load are reduced while the clearances on the remote side of the shaft are increased. The variation in clearance between the land areas surrounding each recess 87 causes a variation of fluid resistance from the recess to change correspondingly the fluid flow and fluid pressure. The restrictor 97 has such fluid resistance that through flow at normally balanced conditions causes a pressure drop across the restrictor of approximately one-half or one-third of the high pressure source at tap 96. The increased fluid flow from the remote recess, because of the increased clearance between the confining land areas and the shaft, causes restrictor 97 in the passageway 92 to have a greater effect on the pressure drop of the fluid delivered to the recess. Thus on a remote side of the shaft, the fluid pressure of the recess 87 is materially reduced from that of the balanced film condition.

Conversely, the displacement of shaft 18 within bore 86 causes a reduction in clearance between the land areas opposing the load and the shaft to increase the flow resistance. Thus through-flow decreases since the total resistance to flow increases. At reduced flows, the throttling effect of restrictor 97 on the fluid pressure admitted to the recess 87 is reduced to increase the fluid pressure within the recess.

Thus the increased and decreased radial clearance between the pressure confining land areas of the recesses 87 in line with the shaft displacement and thus the load, causes the fluid pressures within the recesses to change. The differential fluid pressures on the opposite sides of the shaft tend to center the shaft within the bore until balance with the applied load is attained.

Similarly, axial misalignment of shaft 18 can be adequately absorbed by the fluid pressure differences established in the axially spaced recesses defined by land areas 89 and 90. Thus when there is a force on shaft 18 tending to tilt the shaft in a plane extending through its longitudinal axis, the differential fluid pressures act within each of the recesses on the opposite sides of the shaft and at opposite longitudinal ends of the bore 86 to produce a counteracting force; thus avoiding direct metal-to-metal contact.

The counteracting force of the fluid film tilts the hub 80 somewhat to follow the shaft, the tilting being absorbed by flexure of the radial spokes or ribs 81. The transverse cross-section of each rib 81 has sufficient strength to support adequately any radial load acting on the bearing 20 to maintain the centerline of the bearing at the same general transverse position. The relatively high slenderness ratio, or the ratio of length of rib 81 compared to its transverse cross-section gives flexure to the rib to permit limited angular deflection of hub 80 with respect to rim 82. Thus, any axial misalignment of the shaft 18 can be absorbed by changes in the film thickness and resulting pressure difference, and by flexure of the radial ribs 81.

The load from fluid pressures confined in chambers 55 between crank 42 and the rotating cage 24 supported by shaft 18, produces a large bending moment and radial load on the projecting cantilever end of the shaft. The bearing 19 and the herein described fluid film bearing 20 support and maintain axially spaced points on the shaft 18 in generally fixed transverse positions. The high radial load, plus the moment of cage 24 on shaft 18, causes the shaft to bend in a curved manner generally between the confining spaced bearings 19 and 20. The misalignment of the shaft caused by the above deflection can now be adequately absorbed by the self-aligning film bearing 20 without metal-to-metal contact.

The resistance torque, or loss, of the bearing 20 is small because the force required to shear an oil film is a function of the area of the film and is inversely proportional to the thickness of the film; in this design a thin film exists only at the land surfaces, which are small in area. Moreover, the bearing is smaller in diameter than existing anti-friction bearings of long life and good reliability, and hence the loss is appreciably less.

The fluid pressures confined within each chamber 55 also act in part on the cylinder member 46 surrounding the opening 51, and are in part communicated through the radial openings 51 to the ports 58 and 60. The fluid in the ports 58 and 60 acts against the portion of the cylinder member 46 other than opening 51 in line with the ports, and leaks from between the crank 42 and bore 48 in the form of a fluid film. When the unit is operating, one of the ports will be under greatly higher fluid pressure than the other. Although the effects on cylinder member 46 of fluid pressures within the chambers 55 and ports 58 and 60 generally oppose each other, there still results a vector component force generally in a direction transverse to the rib 61 connecting the lands 62. Unless this phenomenon (commonly called separating force) is corrected, it causes the mating crank 42 and cylinder member 46 on the high pressure side to separate, thereby tending to bind the low pressure side in metal-to-metal contact.

FIGURES 3, 5 and 6 show a particular embodiment of means operable to overcome the separating force, thus eliminating metal-to-metal contact. Fine circumferential grooves 105 and 106 in crank 42 are formed adjacent the ports 58 and 60 respectively and extending generally parallel thereto. The grooves 105 and 106 are separated from each other by lands 107, and each from its adjacent port 58 or 60 by a narrow land area 108 generally between $\frac{1}{16}$ to $\frac{1}{4}$ of an inch across, depending on the diameter and length of the bore and radial clearance, etc. The grooves themselves are approximately $\frac{1}{32}$ to $\frac{1}{16}$ of an inch across and only a few thousandths of an inch deep. Land area 109 separates

the grooves from the outside edges of cylinder member 46 mating on the crank 42.

FIGURE 5 shows a passage 112 that intercommunicates axial bore 64 and groove 106 on one side of crank 42 while a passage 113 intercommunicates bore 66 and groove 105 on the opposite side of the crank. A restrictor 115 of high fluid resistance is disposed in each passage 112 and 113 and is operable to throttle the through-pressure by approximately one-third or one-half when exposed to balanced flow conditions.

Referring now to FIGURE 6, the pressure forces and distributions on opposite sides of crank 42 (ignoring the effect of openings 51) are shown by the appropriate areas, generally designated 116. Assume that port 58 and passages 64 are exposed to the high pressure fluid, while port 60 and passages 66 are exposed to the low pressure fluid. Thus, the separating force caused by the previous mentioned differentials in pressure acting on cylinder member 46 would ordinarily tend to cause binding of the right side of crank 42 (FIGURE 5) against the cylinder member. To counteract this tendency, the high pressure from bore 64 is communicated through passage 112 across restrictor 115 to groove 106. The film clearance is extremely small adjacent grooves 106 so the film pressure quickly builds up. Clearance adjacent port 58 initially is quite large so that leakage of the high pressure fluid across land area 108 to groove 105 reduces the pressure considerably, groove 105 being interconnected across passage 113 and restrictor 115 further throttling the pressure to port 60. The restrictors 115 materially reduce flow through the passages 112 and 113 so that actually little loss because of fluid flow results. The pressures at the grooves 105 and 106 then are dissipated approximately linearly across land areas 109 to the end of the cylinder member on the crank or across land area 108 to low pressure port 60.

The integrals of opposing fluid pressures acting over the appropriately chosen land and groove areas balance cylinder member 46 more symmetrically on the crank 42 to eliminate any direct metal-to-metal contact. The land and groove areas and their positioning on the crank from the ports can be accurately determined to counteract the separating force previously encountered.

It has been noted that each piston 54 has a generally larger bearing surface 56 matable with the shoe surface 29 on cage 24. Each piston further has a through-bore 117 (FIGURE 10) extending from the defined fluid chamber 55 to a recess 118 on the innerface of bearing surface 56 separated from the periphery thereof by land area 119. A restrictor 120 is fixed within the through-bore 117. As fluid pressure in chamber 55 forces the piston 54 and thus bearing surface 56 toward shoe surface 29, the fluid is simultaneously delivered at a reduced pressure via through-bore 117 and the restrictor 120 to recess 118 on the innerface of the bearing surface.

The outwardly acting radial forces on piston 54 will be balanced when the average fluid pressure acting within recess 118 and film pressure on land areas 119 produces a resultant opposing force of equal magnitude. Under any balanced condition, the clearance between the surfaces 29 and 56, or the film thickness will be of a certain value, and thus of a given fluid resistance.

If the film thickness should increase, because of reduced film resistance, flow would increase, the restrictor 120 would have a greater effect in throttling the fluid pressure to recess 118 between the surfaces than with balanced film thickness to reduce the average film pressure. The supporting capacity of the fluid film thereby is reduced with the reduced film pressure, permitting a reduction in film thickness, and automatically varying the film thickness and pressure until balance is established. Similarly, if the film thickness is reduced; the film flow would be reduced, so that restrictor 120 would have less effect in throttling the pressure, and the pressure introduced to recess 118 increases to increase the average film pressure.

Thus the adjacent surfaces are supported spaced apart by a fluid film regardless of any unbalancing tendency acting on the members. The restrictor 120 throttles the pressure to recess 188 as required to maintain balance for all applied forces. It has been observed that if at balanced film conditions the restrictor 120 throttles the pressure by one-third to one-half of that of the pressure source, the balanced film thickness does not vary appreciably with variation of chamber pressures, and thus of loads. Thus at balanced conditions of the film, the film thickness is generally independent of the operating pressure.

As was mentioned above, the longitudinal center axis 40 of pintle 36 is offset by a given distance from the longitudinal center axis 44 of crank 42. Similarly, pintle 36 is supported in housing 12 with its longitudinal center axis 40 offset a similar distance from the longitudinal center axis 22 of the shaft 18, cage 24 and shoe surfaces 29, or the center axis of the unit 10.

FIG. 9 shows an operational sketch, greatly enlarged but similar to the front section view (FIGURE 2), of the variable flow per cycle adjustmtnt feature for the unit. The unit center axis of the shaft 18, cage 24 and the shoe surface 29 is represented at 22; the center axis of pintle 36 is represented at 40; and the center axis of crank 42 at maximum eccentricity is represented at 44. The eccentricity of the unit 10 is the distance from crank center axis 44 to the unit center axis 22, as is shown for the maximum eccentricity along the maximum top dead center line 22, 40 and 44. The stroke of pistons 54 in cylinders 50 is twice the eccentricity, the maximum stroke being along top dead center line 123, 22, 40 and 44 as indicated.

Since pintle 36 can rotate in housing 12 about its center axis 40, the locus of points defined by the adjusted crank center axis 44 extends along line 124 in a circular path above pintle axis 40 and intersects the unit center axis 22. When pintle 36 is rotated by worm gear 41 through an angle 126 crank axis 44 is shifted to 44*a*. Line 128 between the adjusted crank axis 44*a* and machine axis 22, or the adjusted top dead center of the eccentric, rotates through an angle 130 from the maximum top dead center line 22, 40 and 44. Through elementary trigonometry it is apparent that angle 130 is one-half the angle 126. Thus for pintle rotation through any angle 126, the top dead center of the adjusted eccentric is shifted by angle 130 from the maximum top dead center along line 22, 40 and 44.

For an angle 126 of pintle rotation, the eccentricity is reduced from its maximum along straight line 22, 40 and 44, to its adjusted value designated by line 128 between 22 and 44*a*. The reduction of the eccentricity is shown by distance 132. Thus, as the shaft 18, cage 24 and shoe surfaces 29 rotate about the unit center axis 22, the contraction and expansion of the pistons 54 in the cylinders 50 varies by twice the adjusted eccentric distance 22, 44*a*, or by the adjusted stroke 134, 22, 44*a*. The eccentricity varies as a harmonic function from its maximum 22, 40, 44 to zero at 22 as the pintle 36 is rotated from the top dead center position through angle 126 equal to 180°; in fact as the cosine of one-half the angle 126 of pintle rotation.

The crank 42 is rotated through a similar angle as the pintle 36. Thus as pintle 36 is rotated an angle 126, the land areas 62 of crank 42 are similarly rotated an angle equal to 126, as represented in FIGURE 9 by points 62*a* defined by the intersection of line 135 extending from machine center 22 at an angle equal to 126 (shown as 130 and 136), and the adjusted eccentric circle 137.

Thus, radial openings 51 communicating between fluid chambers 55 and the ports 58 and 60, do not pass the lands 62 represented by the points 62*a* until cylinder member 46 is rotated relative to crank 42 an additional

angle 136 past the adjusted top dead center axis 134, 22 and 44. From points 138, across 44a and 134 respectively, to point 62a, represented by an angular rotation of cylinder member 46 on crank 42 through angles 130 and 136 equal to pintle rotation 126, the pistons 54 are reciprocated within the cylinders 50 a distance 140 without actually causing fluid flow through the unit 10. The fluid in each chamber 55 is caused to reverse normal flow during relative rotation of cylinder member 46 on crank 42 through an angle represented by 130; but is permitted to resume normal flow during the subsequent equal annular rotation represented by 136.

Thus the effective adjusted stroke of pistons 54 in cylinders 50 is not twice that of the adjusted eccentricity 22, 44a; but is that portion which is represented by line 142, 22, 142 on the adjusted top dead center line 134, 22 and 44a. The effective adjusted stroke varies from the adjusted stroke as the cosine function of one-half the angle 126 of pintle rotation from 0° to 180°. Since the adjusted stroke also varies as the cosine function of one-half of angle 126, the effective adjusted flow per cycle, consequently varies from maximum to zero as the square of the cosine function of one-half the angle 126 of pintle rotation from 0° through 180°.

Thus, for pintle adjustment through an angle 126, the adjusted effective flow of unit 10 is varied both by the double mounted eccentric shaft 18, and pintle 36 and crank 42, and by phasing lands 62 with respect to the top dead center positions. A graph showing the flow capacity per cycle as a function of adjusted pintle rotation from 0° to 180° traces a reverse S (it being the square curve of the first quarter cycle of a cosine function) having generally curved opposite ends and having a generally linear intermediate portion. This is quite different from similar graphs of conventional variable flow units wherein the adjustment is affected either singularly by varying the stroke or singularly by phasing the port lands with respect to the top dead center positions.

It will also be noted that the structure including the double eccentric relationship of the shaft 18, pintle 36 and crank 42 although easily fabricated is capable of withstanding rigorous service conditions. Since pintle 36 does not rotate rapidly, bearings 38 and 39 need not have the characteristics required of film bearing 20 for rapidly rotating shaft 18. A simple worm gear 41 keyed to pintle 36 can be used to adjust the crank 42 with respect to shaft 18 to vary the flow capacity, as previously described. The mounting of the pintle 36 is such that any adjusted position of the pintle can be adequately and accurately maintained by the worm gear 41 and its driving pinion 43; although other structure equally as simple could be used.

While a single embodiment has been shown, it is obvious that variations therein can be made which employ the basic concepts of the subject invention. Accordingly, it is desired that the invention be limited only by the claims hereinafter following.

What is claimed is:

1. A hydraulic pump or motor unit, comprising an annular cage having spaced shoe surfaces disposed symmetrically of its center axis, external drive means connected to said cage to move directly therewith during rotation of the latter about said axis, a pintle mounted on a rotational axis offset from said center axis, said pintle including integral therewith a cylindrical crank having its longitudinal center axis offset from the rotational axis by a distance similar to said first-mentioned offset and generally disposed at some eccentricity from the cage center axis, mating pistons and cylinders supported respectively, between the shoe surfaces and the crank and defining a plurality of fluid chambers, the distance between each shoe surface and the crank varying by twice the eccentricity during a rotation of the cage about its center axis, said pistons and cylinders being reciprocable thereby along axes extending radially from the crank center axis to vary the volumes of said fluid chambers, means operable to port the fluid chambers including separate inlet and outlet ports on the crank terminating in diametrically spaced relationship generally equidistant of a plane through the above-mentioned axes of the cage and pintle, and through the center axis of the crank when the latter is so positioned, and means to rotate the pintle about its rotational axis so as simultaneously to adjust the eccentricity of the cage axis from the crank axis and to phase the porting of the fluid chambers effective to vary the volumetric flow capacity per cycle of the unit.

2. A hydraulic pump or motor unit, comprising a housing, an annular cage having spaced shoe surfaces disposed symmetrically of its center axis, an external drive shaft connected to the cage symmetrically of the center axis operable to support said cage and to be in driving relationship therewith, a pintle supported by the housing on a rotational axis offset from said center axis, said pintle including rigidly secured thereto a cylindrical crank having its longitudinal center axis offset from the rotational axis and disposed radially in line with the shoe surfaces, mating pistons and cylinders supported respectively, between the shoe surfaces and the crank and defining a plurality of fluid chambers, the distance between each shoe surface and the crank varying by twice the eccentricity of the cage axis from the crank axis during a rotation of the cage about its center axis, said pistons and cylinders being reciprocable thereby along axes extending radially from the crank center axis and operable to vary the volumes of said fluid chambers, means to port the fluid chambers, means to rotate the pintle about its rotational axis to adjust the eccentricity of the cage axis from the crank axis and simultaneously phase the porting to the chambers, and self-aligning bearing means for supporting the shaft rotatably of the cage center axis, said bearing means including a rigid hub having a bore closely surrounding the shaft, means including a source of fluid under high pressure and fluid passages having direct communication therewith and with the inner-face of the bore operable to establish a fluid film between the hub and the shaft, each of said passages having a resistance to flow comparable to the flow resistance of the fluid film under normal film flow and clearance conditions, and flexible structure between the hub and housing flexibly supporting the former so as to accommodate a limited tilting thereof relatively to the normal longitudinal center axis of the shaft.

3. A hydraulic pump or motor unit, comprising in combination, a housing, an annular cage supported by the housing and rotatable about its longitudinal center axis, external drive means in direct driving relationship with said cage, a cylindrical crank supported by the housing and having its longitudinal center axis parallel to but offset from the cage center axis by a given eccentricity, said cage having a plurality of circumferentially spaced shoe surfaces radially spaced from the crank, a cylinder member having a through-bore matable receiving the crank, said cylinder member having a corresponding plurality of cylinders extending radially from the crank open toward the respective shoe surfaces, a piston member matably received in each of the cylinders and defining therewith a variable volume fluid chamber, each piston member symmetrically of and in line with the cylinder having a bearing surface of greater area than the area of the cylinder and disposed matably adjacent the respective shoe surface, the cylinder member being freely rotatable on the crank and having no mechanical connections to the cage or to the drive means other than through the piston members, so that the cage, cylinder member and piston members all tend to rotate in the same direction at the same speed about their respective centers, and that during each rotation each piston member and cylinder member reciprocate relative to one another by twice the eccentricity, means to port the fluid chambers, said crank having opposing separated grooves intermediate the mating through-bore periphery of the cylinder member, and means to supply the piston bearing surface and crank

grooves with fluid at pressures generally inversely proportional to the clearances of the interfaces defined proximate thereto for establishing fluid films therebetween, said last-mentioned means including a source of high pressure fluid, and structure forming a restricted through passage intercommunicating the pressure source and each interface and having a flow resistance comparable to that of the respective fluid film under normal film clearance conditions.

4. A hydraulic pump or motor unit, comprising an annular cage having spaced shoe surfaces disposed symmetrically of its center axis, a pintle supported to rotate about a rotational axis offset from said center axis, a cylindrical crank formed integrally of the pintle and having its longitudinal center axis offset from the rotational axis, mating pistons and cylinders supported between the shoe surfaces and the crank and defining a plurality of fluid chambers spaced radially of the crank, said pistons and cylinders being adapted to reciprocate relative to one another along axes extending radially from the crank center axis and to rotate with the cage during rotation of the cage relative to the crank effective to vary the volumes of said fluid chambers, means to port the fluid chambers including circumferentially aligned diametrically spaced separate inlet and outlet ports terminating on the periphery of the crank in radial alignment with the pistons and cylinders and generally symmetrical of a plane through all of said centers in the maximum throw position, said port means being adapted to communicate separately and successively with each fluid chamber upon each rotatable cycle of the cage and the pistons and cylinders relative to the crank, and means to rotate the pintle and integral crank equal rotational angles about the rotational axis effective to adjust the distance between the cage center axis and the crank center axis and simultaneously therewith to phase the port means relative to the adjusted throw position for varying the flow capacity per cycle of the unit.

5. In a hydraulic pump or motor unit having a cylindrical crank member including separate circumferentially aligned fluid inlet and outlet ports terminating on the exterior of the crank member in generally diametrically spaced relationship, and a cylinder member having a bore snugly receiving the crank member effective to cover the inlet and outlet ports and defining in part at least one expansible fluid chamber adapted to communicate separately with the fluid ports through an opening terminating at the bore periphery radially in line with the fluid ports upon each rotation of the cylinder member relative to the crank member, the combination including a pintle member supported rotatably by the unit to rotate about a given rotational axis, the crank member being securely fixed to the pintle member and having its longitudinal center axis offset from but parallel to the rotational axis of the pintle member, the crank member also having on its periphery at least two separate circumferential grooves spaced from one of the fluid ports and extending circumferentially to approximately in line axially of the bore with the circumferential terminations thereof, each of said grooves being of narrow width and shallow depth and being on the interface of the bore spaced from said one fluid port, means forming fluid passageways between the other of the fluid ports and the separate grooves, the fluid leakage from the grooves and the ports between the adjacent portions of the crank member and the bore periphery being effective to establish a fluid flow therebetween, means forming a restrictor in each of the passageway means of flow resistance comparable to that of the fluid film under normal film flow and thickness conditions, and means to rotate the pintle and crank member about the rotational axis to vary the fluid output per cycle of the unit.

6. In a hydraulic pump or motor unit having a cylindrical crank member including separate circumferentially aligned fluid inlet and outlet ports terminating on the exterior periphery of the crank member in generally diametrically spaced relationship, and a cylinder member having a bore snugly receiving the crank member effective to cover the inlet and outlet ports and defining in part at least one expansible fluid chamber adapted to communicate separately with the fluid ports through an opening terminating at the bore periphery radially in line with the fluid ports upon each rotation of the cylinder member relative to the crank member, the combination including a pintle member supported by the unit to rotate about a fixed rotational axis, the crank member being formed integrally with the pintle member and having its longitudinal center axis offset from but parallel to the rotational axis of the pintle, the crank member also having on its periphery separate circumferential grooves spaced endwardly of each of the fluid ports on both sides thereof and extending circumferentially to approximately in line axially of the bore with the circumferential terminations thereof, each of said grooves having a generally small cross area of approximately $\frac{1}{32}$ to $\frac{1}{16}$ of an inch width and approximately a few thousandths of an inch depth and being on the interface of the bore endwardly of the fluid port spaced generally $\frac{1}{16}$ to $\frac{1}{4}$ of an inch therefrom, means forming fluid passageways between each fluid port and the separate grooves adjacent the opposite fluid port, the fluid leakage from the grooves and the ports between the adjacent portions of the crank member and the bore periphery being effective to establish a fluid film therebetween, means forming a restrictor in each of the passageway means of flow resistance comparable to that of the fluid film effective to throttle the fluid passing therethrough under normal film flow and thickness conditions by one-third to one-half the original fluid pressure, and means to rotate the pintle and crank member about the rotational axis to vary the fluid output per cycle of the unit.

References Cited by the Examiner

UNITED STATES PATENTS

388,522	8/88	Beauchemin	103—161
1,502,310	7/24	Magie et al.	103—161
1,778,238	10/30	Wilsey	103—161
1,947,050	2/34	Kuzelewski	103—161
2,392,754	1/46	Mercier	103—161
2,393,128	1/46	Temple	103—161
2,515,861	7/50	Campbell	308—26
2,599,609	6/52	Carey	103—161
2,635,015	4/53	Babb	308—15
2,639,673	5/53	Hadekel	103—161
2,675,763	4/54	Muller	103—161
2,679,210	5/54	Muller	103—161
2,683,422	7/54	Richards	103—161
3,044,838	7/62	Winer et al.	308—122
3,082,696	3/63	Henrichsen	103—161
3,101,060	8/63	Bjorklund	103—161
3,110,527	11/63	Fox	308—122

FOREIGN PATENTS

953,223	11/56	Germany.
116,440	6/18	Great Britain.

LAURENCE V. EFNER, *Primary Examiner.*

JOSEPH H. BRANSON, JR., *Examiner.*