May 10, 1966 M. J. KRAWACKI 3,250,223

VANE IMPULSION APPARATUS

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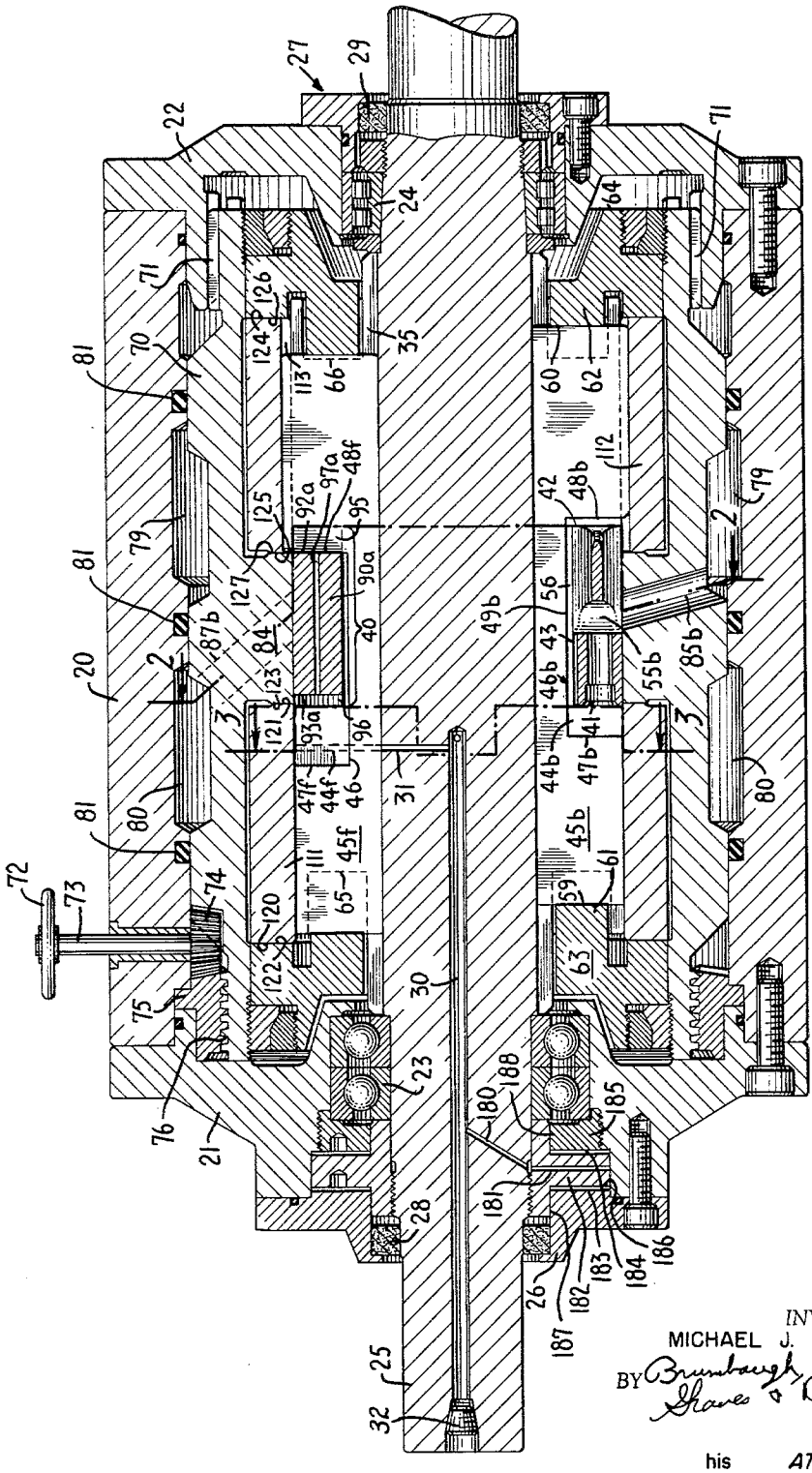


FIG. 1

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VANE IMPULSION APPARATUS

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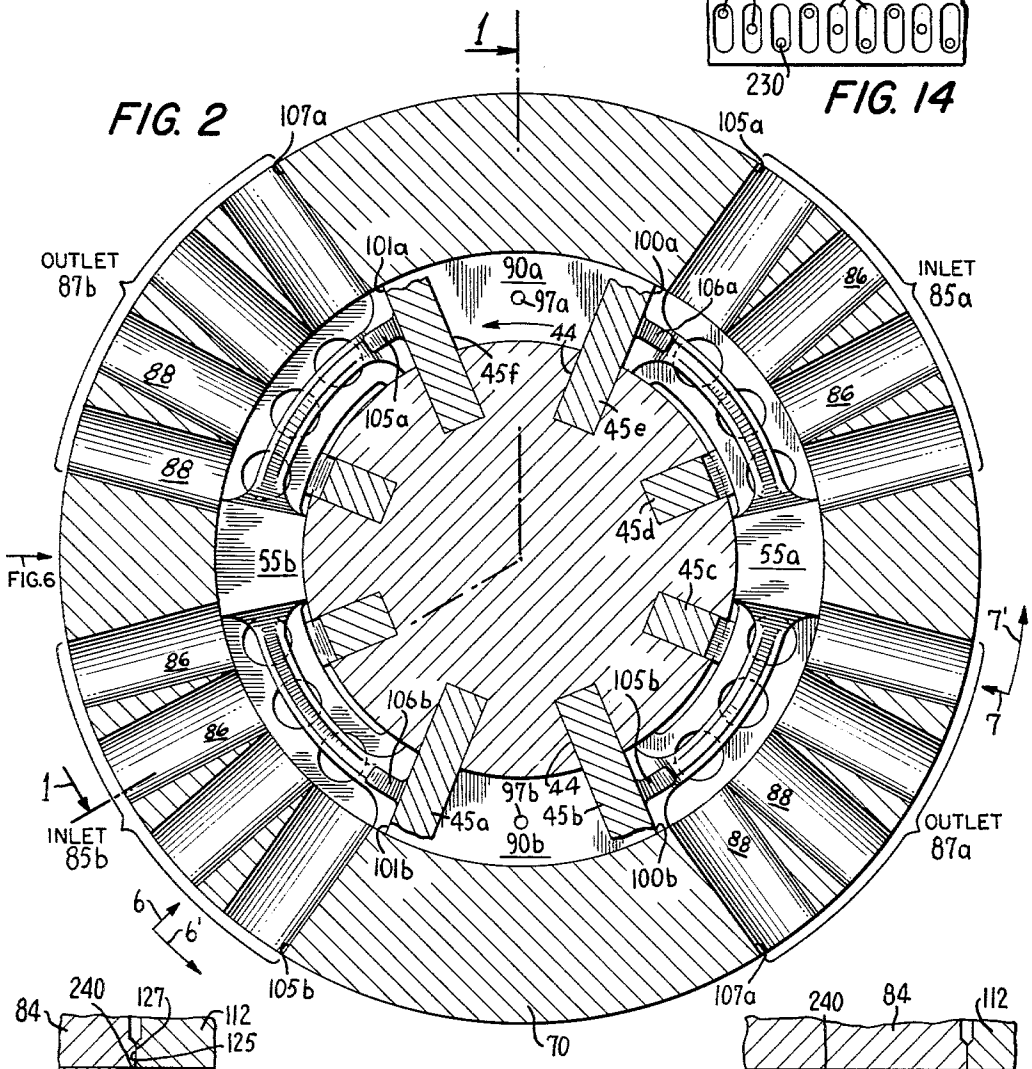


FIG. 2

FIG. 14

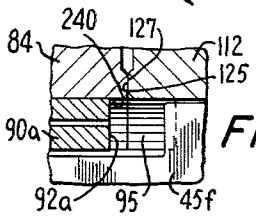


FIG. 15

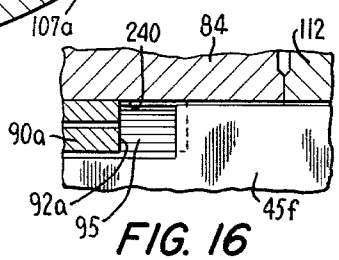


FIG. 16

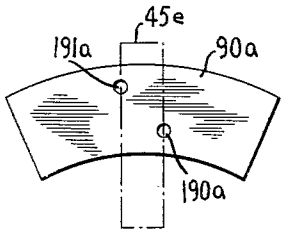


FIG. 8

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VANE IMPULSION APPARATUS

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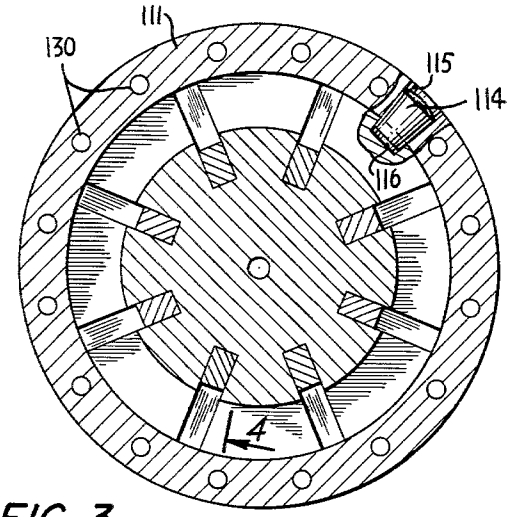


FIG. 3

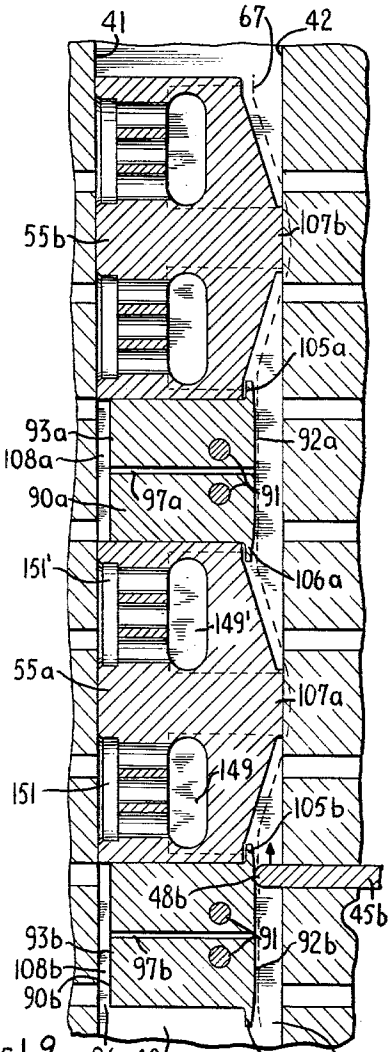


FIG. 6

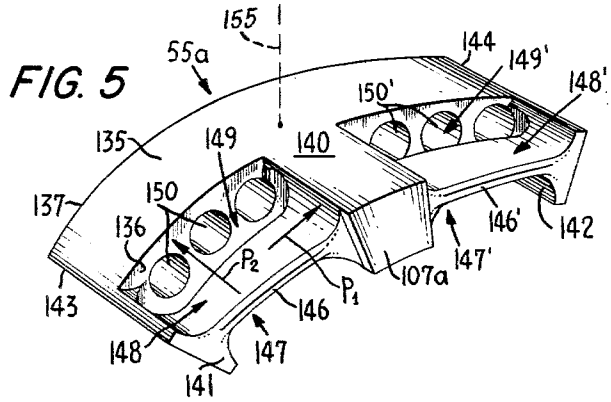


FIG. 5

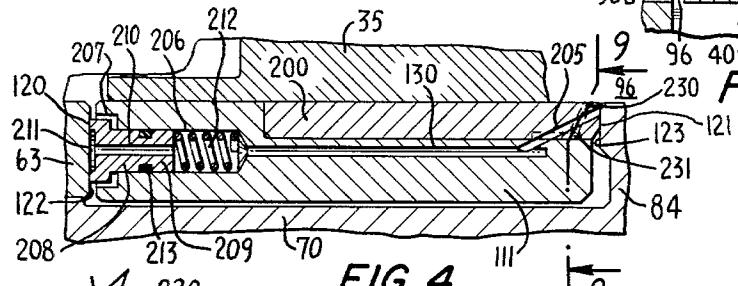


FIG. 4

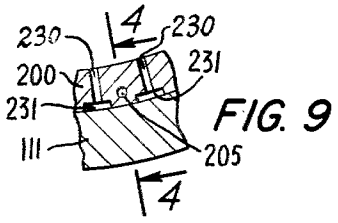


FIG. 9

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VANE IMPULSION APPARATUS

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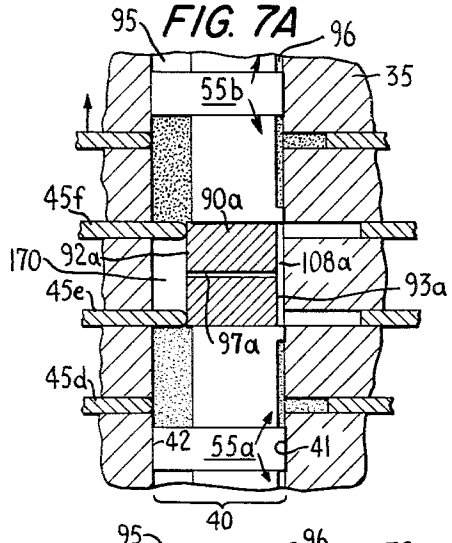


FIG. 7A

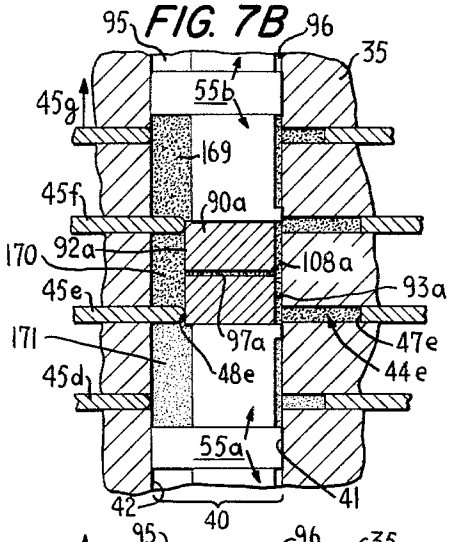


FIG. 7B

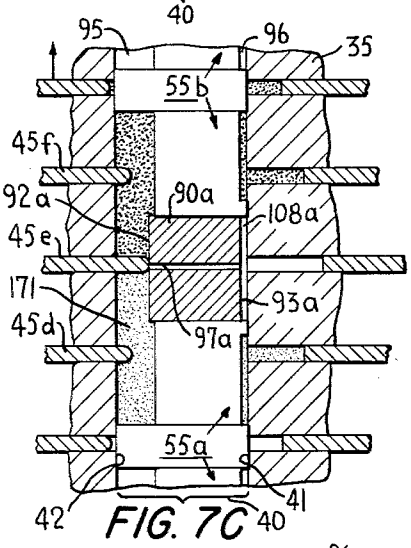


FIG. 7C

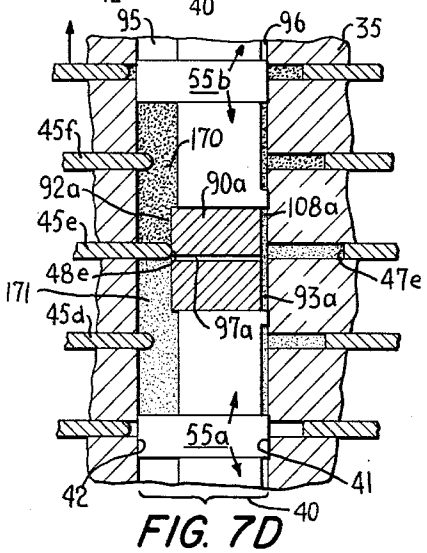


FIG. 7D

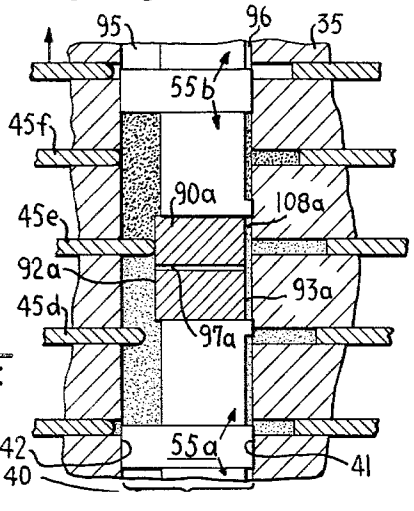


FIG. 7E

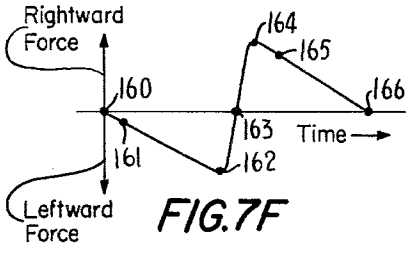


FIG. 7F

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VANE IMPULSION APPARATUS

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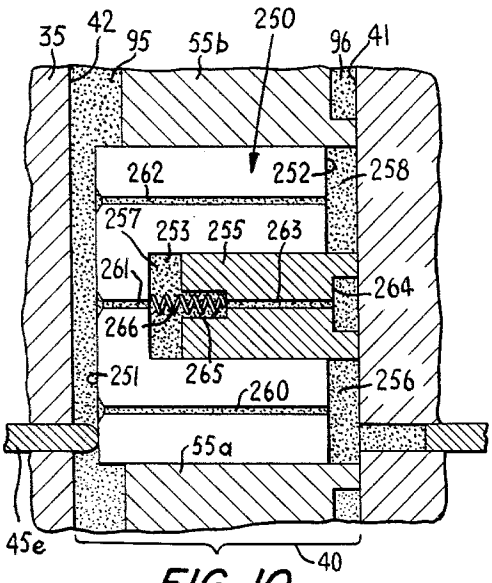


FIG. 10

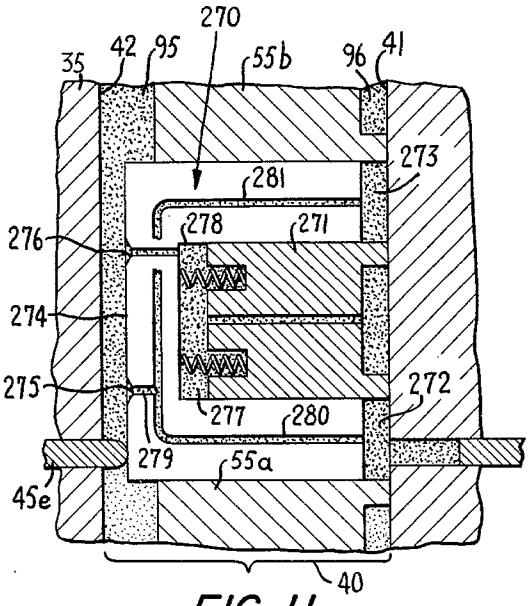


FIG. 11

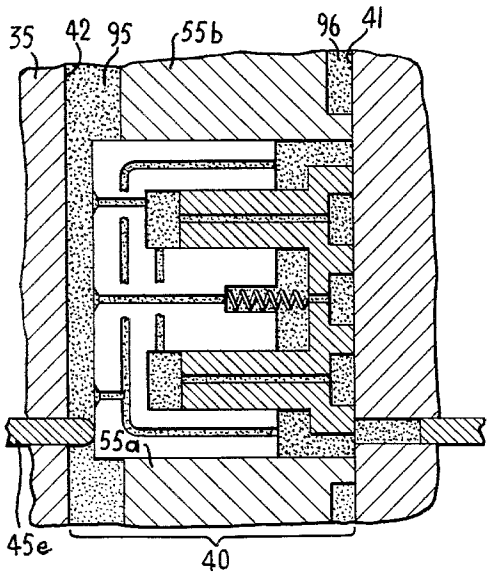


FIG. 12

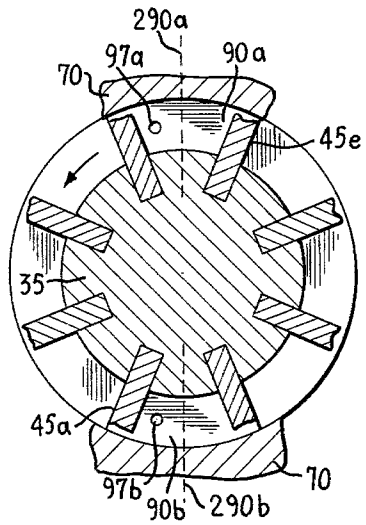


FIG. 13

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United States Patent Office 3,250,223 Patented May 10, 1966

3,250,223

VANE IMPULSION APPARATUS

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Filed May 14, 1962, Ser. No. 194,450
32 Claims. (Cl. 103—139)

This invention relates generally to vane impulsion apparatus and more particularly to improvements in, for example, the apparatus of such sort which is described in my co-pending U.S. application Serial No. 853,912 (filed November 18, 1959), now Patent No. 3,033,122, and in British Patent 860,566.

By vane impulsion apparatus is meant apparatus wherein a transfer of energy takes place between an expansible or inexpansible fluid and one or more mechanical parts referred to herein as "vanes." Ordinarily, a plurality of vanes are used. The machine which carries those vanes is comprised of "rotor" and "stator" units between which there is relative rotational movement, and which are coaxial in the sense that they have a common axis, and that one unit is surrounded by the other.

The term "rotor" is used herein in a broad sense to refer to the vane-carrying one of the two mentioned units rather than to indicate that the unit referred to is rotatable in an absolute sense, e.g., with respect to the foundation on which the machine is mounted. In other words, "rotor" as used broadly herein is to be taken as pertaining to relative rotation (with the "stator") rather than as pertaining to absoluate rotation. Thus, "rotor" may be considered as descriptive of the vane-carrying one of the mentioned two units even though that vane-carrying unit is stationary relative to the foundation for the machine (although rotating relative to the "stator"), and/or even though such vane-carrying unit is disposed outside the "stator" so as to surround it.

Returning to a description of the basic character of vane impulsion apparatus, the vanes thereof are carried in a plurality of slot formed in the rotor to be angularly spaced around an annular fluid-receiving channel of which at least part is provided by an annular groove formed in the rotor and surrounding its axis. Each of those slots runs transversely of such channel and opens into such channel on at least one side thereof, the arrangement of slots and vanes being such that each vane is transversely movable within its slot between a position at which the vane obstructs the channel and a position at which the vane leaves the channel unobstructed. While one mode of such transverse movement of the vanes is axial translatory movement, the terms "transverse" and "transversely" are used herein in a sense broaded than "axial" and "axially." Thus, according to the usage herein, the term "transversely movable" as applied to the vane is descriptive not only of axial translatory vane movement, but, in addition, of other types of vane movement as, say, radial translatory, combined radial and axial translatory, angular movement in a plane transverse to the fluid receiving channel, or combined angular and translatory movement in such plane.

The mentioned fluid-receiving channel has received therein at least one reaction block unit which (in the said relative rotational sense) is fixed as to position in the direction around the rotor axis, and which is disposed in the channel to obstruct angular flow of fluid therein. Preferably a plurality of reaction block units are used. Inlet and outlet fluid ports communicate with the channel on angularly opposite sides of the centerline of each block unit. In operation, the rotor rotates relative to the block units. During such rotation, means are employed to move each vane within its slot so that, at certain times, the vane is positioned in the channel to obstruct ffow of fluid therein in the angular direction (i.e., in the direction around the channel), and so that, at other times, the vane is positioned out of the channel to thereby be enabled to pass each reaction block unit.

When a vane is in fluid-obstructing position, the fluid in the channel produces a differential in the fluid pressure exerted on the angularly opposite sides of the vanes. If the relative rotation of the rotor and the one or more reaction block units is such as to move each vane in the same direction as the force exerted thereon by the pressure differential, the described machine acts as a motor. Conversely, if such relative rotation moves each vane in the direction opposite to that of the force exerted on the vane by the mentioned pressure differential, the described machine acts as a pump.

Among the objects of my invention are, in connection with vane inpulsion apparatus, to provide alone or in combination any one or more of the following: ability to vary the volumetric fluid capacity of the apparatus without any accompanying dumping of fluid through a route between the inlet and outlet ports which bypasses the channel in which the fluid does work or has work done thereon, utilizing fluid pressure to assist in maintaining the vanes in fluid-obstructing position, minimizing or compensating the net fluid pressure force exerted axially on the rotor, damping of transient variations in such force, minimizing or compensating for the next fluid pressure force exerted axially and/or radially on parts of the apparatus other than the rotor; improvements in reaction block units and in means for maintaining such units in appropriate position, improved sealing-in of the pressurized fluid within an annular channel of the apparatus, and minimization of wear and other undesirable effects due to outward throwing of the vanes by centrifugal forces.

The first named object (variation in the volumetric fluid capacity of the apparatus) is realized according to the invention by providing for the apparatus an abutment means which is axially adjustable relative to the rotor, and which cooperates with the groove in the rotor to define therewith an annular, fluid-receiving channel bounded by a rotor-provided side, a rotor-provided bottom, and a side wall furnished by the abutment means. When the abutment means is axially adjusted in position relative to the rotor the cross sectional area of the channel is correspondingly varied to thereby vary the volumetric capacity of such channel.

Preferably but not necessarily, the rotor has formed therein a two-sided groove, the abutment means is arcuate in form and fits around at least part of the groove bottom, and the said abutment means is axially disposed intermediate the two sides of the groove to divide it into two annular channels on axially opposite sides of the abutment means. Of those two channels, one is a working channel in which the vanes interact with pressurized fluid to provide a pumping effect or a motor effect as the case may be, and the other channel is a balancing channel which likewise contains pressurized fluid, but in which such fluid does not interact with the vanes. Instead, the main purpose of the pressurized fluid in the balancing channel is to offset the axial pressure exerted on the rotor by the working channel fluid so as thereby to reduce the unbalance of the axial forces acting on the rotor. In contradistinction to the balancing channels proposed for prior art devices, the balancing channel of apparatus according to the invention has means therein to obstruct flow of fluid from the one or more inlet ports directly through the balancing channel to the one or more outlet ports. Thus, such apparatus is adapted to obtain a variation in its volumetric fluid capacity without the necessity of resorting to a dumping of some of the fluid, i.e., allowing a certain fraction of the stream of fluid between the inlet and outlet ports to flow therebetween through a balancing channel path which bypasses the

working channel, and in which the fluid neither does work nor has work done thereon.

Apparatus embodying the above-described aspects and other aspects of the invention may take a large variety of forms of which some are indicated in the aforementioned U.S. patent application and in the aforementioned British patent. While certain of the aspects of the present invention are confined to vane impulsion apparatus of variable volumetric fluid capacity, others of such aspects are useful both in connection with such variable capacity apparatus and in connection with vane impulsion apparatus characterized by a constant volumetric capacity or displacement for fluid. Moreover, while the invention will be described in terms of a vane impulsion pump, the invention is equally applicable to a vane impulsion motor.

For a better understanding of how the above-mentioned objects and other objects of the invention may be realized, reference is made to the following description of exemplary embodiments of the invention, and to the accompanying drawings wherein:

FIG. 1 is a front elevation in cross section of a variable capacity, vane impulsion pump embodying the present invention, the view being taken in cross section in the manner indicated by the arrows **1—1** in FIGURE 2;

FIG. 2 is a view (taken in cross section transverse to the axis of the FIG. 1 machine in the manner indicated by the arrows **2—2** in FIG. 1) of a portion of the FIG. 1 machine;

FIG. 3 is a view (taken transverse to the axis of the FIG. 1 pump in the manner indicated by the arrow **3—3** in FIG. 1) of another portion of the FIG. 1 machine;

FIG. 4 is a front elevation taken in cross section (as indicated by the arrows **4—4** in FIG. 9) of the bottom of the left hand sleeve in the FIG. 1 pump;

FIG. 5 is an isometric view of one of the reaction block units of the FIG. 1 machine;

FIG. 6 is a developed view of a cross section at the mean diameter of the rotor groove and associated parts of the FIG. 1 pump, such view being taken as indicated as FIG. 2 by the arrows **6, 6'** of which **6** represents the viewing direction and the location of the bottom of the FIG. 6 view, and **6'** represents the direction of development of such view;

FIGS. 7A–7E are developed, schematic, cross-sectional views of a portion of the rotor groove (and associated parts) of the FIG. 1 machine, such view being taken as indicated in FIG. 2 by the arrows **7, 7'** of which **7** represents the viewing direction and the location of the bottom of the FIG. 7 view, and **7'** represents the direction of development of such view;

FIG. 7F is a graph of the net axial force developed at the time of operation of the FIG. 1 machine represented by FIGS. 7A–7E;

FIG. 8 is a schematic front view of a modification of one of the seal blocks of the FIG. 1 machine;

FIG. 9 is a view in cross section (taken transverse to the axis of the FIG. 1 machine in the manner indicated by the arrows **9—9** in FIG. 4) of a fragmentary portion of the sleeve of which other details are shown in FIG. 4;

FIGS. 10, 11 and 12 are developed, schematic, fragmentary views (partly in cross section) of modifications other than that shown by FIG. 8 of the seal blocks of the FIG. 1 machine;

FIG. 13 is a schematic, cross sectional view (taken in a plane normal to the axis of the FIG. 1 machine) of another modification of such seal blocks;

FIG. 14 is a developed schematic, fragmentary view of the exterior surface of the tubular liner appearing in FIG. 9 as the upper cross-sectioned member of that last-named figure, and

FIGS. 15 and 16 are fragmentary views (partly in cross section and taken in a plane through the axis of the FIG. 1 machine) of modifications in the seal blocks and right hand vanes of the FIG. 1 machine.

In the description to follow, elements which are counterparts will, when practical, be designated by the same reference numeral but by different suffixes for that numeral. On occasion a reference numeral without a suffix may be used in a collective sense to designate one or more of several counterpart elements. The drawings herein are to be taken as disclosing qualitatively rather than quantitatively the dimensions and character of the shown parts.

General structure

Referring now to FIGS. 1, 2 and 3, the reference numeral **20** designates a hollow, cylindrical, axially stationary housing which has secured to opposite ends thereof a pair of end plates **21, 22**. Suitable means (not shown) are provided to drain leakage fluid from the entire interior of the housing. Mounted within the plates **21** and **22** respectively, are a pair of bearing assemblies **23** and **24** for a rotatable, axially stationary shaft **25** passing through central openings in the end plates and surrounded at those end plates **21, 22** by the stuffing boxes **26, 27** and their accompanying glands **28, 29**. The shaft **25** has formed therein a central fluid bore **30** closed at its left end by plug **32** and connected to the operating zone of the machine through equiangularly spaced conduits **31** (one being shown) extending radially through the shaft.

An axially central portion of the shaft **25** is formed to provide as an integral part thereof a rotor **35** of which the general shape is that of a cylindrical drum. Formed in the rotor **35** at an axially central position thereof is an annular groove **40** of rectangular cross section and having sides **41, 42** and a bottom **43** provided by the rotor. Also formed in the rotor **35** are eight slots **44** (FIG. 2) spaced equiangularly around the rotor and extending axially from end to end of the rotor. As shown by FIGS. 2 and 3, those slots extend radially into the rotor beyond the bottom **43** of the groove **40**.

Each of the eight slots **44** contains an axially movable, rectangular vane **45** having formed therein a central rectangular notch **46** with sides **47, 48** and a bottom **49**. The width of each vane notch between its sides **47, 48** is somewhat greater than the width of the rotor groove **40** between the sides **41, 42** thereof. Moreover, the bottom **49** of each vane notch is radially disposed somewhat inward of the bottom **43** of the rotor groove **40**. As shown in FIG. 6 by the notch side **48*b*** for the vane **45*b***, the notch side **48** for each vane **45** may have a camber.

Disposed in and on diametrically opposite sides of the rotor groove are a pair of reaction block units **55*a*, 55*b*** (FIGS. 2 and 6) whose structure will be later described in further detail. For the present, it suffices to say that each of units **55*a*, 55*b*** is of arcuate form to rest upon and make intimate contact with the groove bottom **43** over an angular interval thereof, that an axial passage **56** for fluid is formed between the inner curved surface of each unit **55*a*, 55*b*** and the bottom **49** of each vane notch when such vane is angularly positioned within the angular interval occupied by the unit, and that each of units **55*a*, 55*b*** extends from side to side of the rotor groove **40** to obstruct flow of fluid therein in the angular direction, i.e. in the direction around the rotor axis.

The vanes **45** are moved axially within their slots **44** by a pair of annular camming faces **59, 60** disposed at opposite ends of each vane to contact the opposite end faces thereof. Those camming faces are the end faces of a pair of annular camming flanges **61, 62** projecting axially outwards of a pair of angularly stationary camming ring assemblies **63, 64** to extend into a pair of annular recesses **65, 66** formed in the opposite ends of the rotor **35**. The cam faces **59, 60** are shaped to produce, during rotation of the rotor, an axial shifting of the vane **45** in the manner indicated by the dotted line **67** (FIG. 6) which qualitatively illustrates the locus of the instan-

taneous axial positions assumed during such rotation by the notch side 48 of each vane. The axial movement of the vanes may, of course, be produced by means other than camming means, as, say, hydraulic vane-shifting means or a wobbler plate.

As so far described, the pump of FIG. 1 is similar in structure and advantages to the apparatus disclosed in the aforementioned U.S. patent application and in the aforementioned British patent.

Coming now to a description of some of the differences between the present apparatus and the apparatus I have previously disclosed in such U.S. application and in such British patent, in the now-described apparatus the camming ring assemblies 63, 64 are positioned at the opposite ends of and threadedly secured to a tubular casing 70 disposed with a sliding fit within the housing 20. While the casing 70 is restrained from angular movement relative to housing 20 by intermeshing, axially running splines 71 formed in the interior of the end cap 22 and on the exterior of the casing, the casing 70 is axially adjustable in position relative to the housing 20. Such axial adjustment is effected by the turning of a hand wheel 72 on a shaft 73 extending through housing 20 to a pinion gear 74 meshing with a ring gear 75 which surrounds the casing. The interior of the ring gear 75 has helical threading 76 meshing with corresponding helical threading formed on the exterior of casing 70. Hence, by the turning of the hand wheel 72 in the appropriate direction, the casing 70 may be axially shifted either leftward or rightward.

The interior surface of housing 20 and the exterior surface of casing 70 are each hollowed out to provide an inlet distribution chamber 79 and an outlet distribution chamber 80 which are each bounded partly by housing 20 and partly by casing 70, and which each extend annularly around the casing. Those two chambers are adapted to conduit means (not shown) to be connected to a fluid-providing system so that inlet chamber 79 receives fluid from that system and the outlet chamber 80 discharges fluid thereto. Inasmuch as the FIG. 1 machine is a pump, the fluid pressures in the inlet and outlet chambers are relatively low and relatively high, respectively.

A plurality of O rings 81 surround the casing 70 and are axially spaced along the interface between such casing and the housing 20 in such manner that each of chambers 79 and 80 has such a ring disposed to either side thereof. The O rings 81 act as seals which obstruct axial flow of fluid through the interface between the casing and the housing. Because of this sealing effect of the rings 81, the casing 70 may be axially adjusted in position relative to housing 20 while, meantime, the chambers 79 and 80 are each maintained fluid-tight in respect to leakage of fluid therefrom through the mentioned interface.

Disposed at an axially central position in its bore, the casing has an instanding annular flange 84 encircling the rotor groove. Inlet chamber 79 is connected to groove 40 by a pair of oppositely disposed fluid inlet ports 85*a*, 85*b* which each consist of an angularly spaced set of conduits 86 passing from the chamber 79 through flange 84 to the groove 40. Likewise, outlet chamber 80 is connected to the groove 40 by a pair of oppositely disposed fluid outlet ports 87*a*, 87*b* which each consist of an angularly spaced set of conduits 88 passing from the last named chamber through the mentioned flange and to groove 40. As shown in FIG. 2, the reaction block unit 55*a* has an inlet port and an outlet port (ports 85*a* and 87*a*) disposed on angularly opposite sides of the centerline thereof, and, similarly, the reaction block unit 55*b* has inlet and outlet ports 85*b* and 87*b* disposed on angularly opposite sides of its centerline. It will be noted that the inlet and outlet ports alternate around the rotor groove, the two inlet (low pressure) ports being directly opposite each other and the two outlets (high pressure) ports also being directly opposite each other.

The flange 84 is extended inwardly by a pair of oppositely disposed, arcuate seal blocks 90*a* and 90*b* of which each is secured to the inner cylindrical surface of flange 84 by a pair of pipe plugs 91 (FIG. 6) received in tapered radial holes (not shown) extending through flange 84 and projecting from those holes inwardly of the flange to be threadedly received in the seal block. The two seal blocks together form what is referred to herein as an abutment means. The arcuate bottom face of each seal block fits around and is in intimate contact with the bottom 43 of the rotor groove 40. In the axial direction, the width of each seal block between its axially opposite front and rear faces 92 and 93 is less than the width of the rotor groove 40 between its sides 41 and 42. Accordingly, the seal blocks 90*a* and 90*b* divide the rotor groove 40 into right hand and left hand annular fluid-receiving channels 95 and 96 (FIG. 6) on axially opposite sides of the seal block. For reasons later apparent, the channel 95 is referred to herein as the working channel, whereas the channel 96 is referred to herein as the balancing channel.

The seal block 90*a* has formed therein a fluid-conducting passage 97*a* extending axially along the centerline of that block between its front face 92*a* and its rear face 93*a*. A similar passage 97*b* is formed in the seal block 90*b*. Those two passages permit fluid pressure to be communicated between the portion of working channel 95 into which each passage opens and the portion of the balancing channel 96 into which such passage opens.

Referring to FIGS. 2 and 6, the reaction block units 55*a*, 55*b* extend from side to side of the groove 40 by passing axially through the opposite gaps by which the oppositely disposed seal blocks 90*a*, 90*b* are separated from each other around the groove. Within the gap occupied thereby, the unit 55*a* fits slidably in the axial direction within the margins 100*a*, 100*b* which belong to respectively, the seal blocks 90*a*, 90*b*, and which bound the two angularly opposite sides of such gap. The unit 55*b* is similarly fitted slidably in the axial direction within the magins 101*a*, 101*b* of, respecitvely, seal blocks 90*a* and 90*b* and disposed on either side of the gap for unit 55*b*. Those axial sliding fits of the reaction block units 55*a*, 55*b* with the angularly stationary seal blocks 90*a*, 90*b* serve to maintain the mentioned units angularly stationary in the direction around the groove despite the fact that such reaction block units have no connection by a pin or the like to the casing 70. It has been found desirable to eliminate such connection in order to avoid the problem created if such connection were present of a canting of each of the units 55*a*, 55*b* around its connection when, during operation, the unit is subject to the differential in fluid pressure developed on angularly opposite sides thereof by fluid in the vicinities of the inlet and outlet ports on opposite sides of the unit. While the reaction block units 55*a* and 55*b* are axially floating relative to the casing 70 and to the seal blocks attached thereto, such units are maintained fixed as to axial position relative to the working and balancing channels 95 and 96 by the sides 41 and 42 of the rotor groove 40. Accordingly, the seal blocks 90*a*, 90*b* can be adjusted in axial position without disturbing the axial positioning of the reaction block units. As taught in my aforementioned copending U.S. application Serial No. 853,912, each of the reaction block units can be anchored to the casing 70 by a coupling which is comprised of a radial pin and an axial keyway, the said coupling being one which holds the unit angularly stationary relative to the casing but which permits axial sliding of the unit relative to the casing.

As will be noted, the forward ends (FIG. 6) 107*a*, 107*b* of, respectively, the block units 55*a* and 55*b* obstruct angular flow of fluid in the working channel 95, whereas the rear portions of such units obstruct angular flow

of fluid in the balancing channel **96**. Those rear portions cooperate with the seal blocks **90***a*, **90***b* and with the side **41** of rotor groove **40** to convert angular subdivisions of the balancing channel **96** into two chambers **108***a*, **108***b* which are disposed behind the rear faces **93***a*, **93***b* of, respectively, the seal blocks **90***a*, **90***b*, and which approach closely to being fully fluid-tight. Those two chambers **108***a*, **108***b* receive fluid from the working channel **95** through, respectively, the passages **97***a*, **97***b* in the seal blocks **90***a*, **90***b*.

The front face **92***a* of seal block **90***a* has tabs **105***a*, **106***a* (FIG. 6) projecting angularly outwards in opposite directions from the angularly opposite ends of that front face to overlap in the working channel **95** with the angular intervals occupied by the reaction block units **55***a*, **55***b*. The seal block **90***b* has similar tabs **105***b*, **106***b*. Those tabs act as "vane-catchers" to prevent the vanes **45** from inadvertently being axially displaced by the camming faces **59**, **60** (FIG. 1) so far towards the fronts of the reaction block units **55***a*, **55***b* that the notch sides **48** of the vanes bear against those units. Such inadvertent excess axial displacement of the vanes might occur when the mentioned camming faces become worn from use, and the vanes consequently fit with some play between such faces. The reason for avoiding any such undue vane displacement and consequent bearing of the vanes on the reaction block units is that the vanes will tend to become jammed when they so bear.

The arcuate extents occupied by the reaction block units **55***a*, **55***b* and by the seal blocks **90***a* and **90***b* are related to certain significant angles (best observed in FIG. 2) which are incorporated in the design of the shown machine, and of which some are as follows: The inter-vane angle (that between the centerline of any two adjacent vanes); the port spread (the angle bisected by the centerline of a reaction block unit and extending from the edge far from that unit of the inlet port adjacent thereto to the edge far from that unit of the outlet port adjacent thereto); the interport angle (that between the edges near to each other of the inlet and outlet ports on either side of a seal block). Some other important angles (best observed in FIG. 6) are: The dwell interval (that angular interval over which each vane is stationary or nearly so in its axial position); the camming interval (that angular interval over which each vane is in axial motion in the course in its angular movement of passing by a reaction block unit and then returning to the dwell position for the vane).

Within a port spread, each vane can be considered as subjected at all times to equal fluid pressures on its angularly opposite sides, wherefore the vane is free to move within its slot. On the other hand, such vane when within an interport angle is subjected on its opposite sides to a fluid pressure differential which tends to force the vane into friction contact with a side of its slot to thereby hinder axial movement of the vane. Because of those considerations, the camming interval (during which the vane is moved) is approximately coextensive with (but may be somewhat greater than) the port spread, and the dwell interval (during which the vane is axially stationary) is approximately coextensive with (but may be some what less than) the interport angle. With such relations existing between the port spread, interport angle, camming interval and dwell interval, evidently the camming intervals can be increased relative to the dwell intervals if, consonantly, the port spreads are increased by decreasing the interport angles until the value of each of the latter angles approaches or (at the limit) reaches about that of the intervane angle. The intervane angle is about the lower limit for the interport angle because, if the interport angle were substantially less, then the interport angle would not at all times contain a vane, and, accordingly, there would be created an intermittent fluid short circuit between the inlet and outlet ports on either side of the interport angle.

As explained in detail in my aforementioned U.S. application and aforementioned British patent, subject to the limitation that the interport angle must not be decreased to the point where there would be created the mentioned intermittent short circuit, it is preferable that the interport angle be made as small as practical in order to render the port spread as large as practical so as thereby to permit a maximizing of the camming interval relative to the dwell interval. Such maximizing of the camming interval is advantageous because it lessens the cam-imparted accelerating forces on the vanes and the resultant wear produced by the vanes on the cams, whereby the machine can be operated at higher rotor velocity to have a higher fluid capacity.

In the presently described machine, the principle of maximizing the camming intervals is followed in that, as shown in FIGS. 2 and 6, the dwell intervals and interport angles are each only slightly greater than the intervane angle, wherefore the camming intervals and port spread of the machine are about as large as it is practical to make them in a machine having two reaction block units and eight vanes. Such maximizing of the camming intervals and the port spreads by minimizing of the dwell intervals and interport angles is a feature of such machine which permits the arcuate extents of the reaction block units **55***a*, **55***b* to be maximized relative to the arcuate extents of the seal blocks **90***a*, **90***b*. Thus, in the presently described machine which has two reaction block units and two seal blocks, the arcuate extents of the former are almost three times as great as those of the latter. To so maximize the arcuate extent of each reaction block unit relative to that of each seal block is advantageous for reasons later set forth.

Returning now to the elements of the FIG. 1 machine, a pair of seal sleeves **111**, **112** are disposed within casing **70** on axially opposite sides of the flanges **84** to each fit around rotor **35** with a small enough clearance that only a minimal amount of fluid leakage can take place through the interspace between the exterior of the rotor and the interior of each sleeve. The right hand sleeve **112** has formed on the inside thereof a plurality of keyways **113** of which each receives as a key the radially outward portion of a respective one of the vanes **45** carried by the rotor **35** in its slots **44**. Hence, the sleeve **112** is keyed by the vanes **45** to the rotor **35** to rotate therewith while, nonetheless, being axially adjustable relative to the rotor. When, during the operation a vane is positioned so that part thereof is inserted in working channel **95** and is subjected on its angularly opposite sides to fluid pressure forces of different value, the seating of the outward portion of the vane in a keyway **113** of sleeve **112** is advantageous because the inserted part of the vane is thereby supported on three sides (on two sides by slot **44** and on one side by the sleeve keyway) against the net pressure force exerted on that inserted part.

While the sleeve **111** is likewise keyed to rotor **35** to rotate with it but to be axially adjustable relative thereto, in the case of the sleeve **111** the keying is accomplished by (FIG. 3) a keying pin **114** received in a tapered hole **115** extending radially through sleeve **111**, the pin being disposed in that hole to have the inner end of the pin project into an axially running keyway **116** formed in the exterior surface of the rotor **35**.

The sleeve **111** is axially fitted between the camming ring assembly **63** and the flange **84** (FIG. 1) such that annular planar rear and front faces **120** and **121** on the sleeve register with respectively matching annular planar faces **122**, **123** on, respectively, the assembly **63** and the side of flange **84** nearest sleeve **111**. Similarly, the sleeve **112** fitted between the camming ring assembly **64** and the flange **84** such that annular planar rear and front faces **124** and **125** on the sleeve register with respectively matching annular planar faces **126** and **127** on, respectively, the assembly **64** and the flange **84**.

Each of the sleeves **111** and **112** has formed therein

a plurality of angularly spaced passages **130** (FIG. 3) permitting communication of fluid pressure from the front end of the sleeve to behind the rear face thereof. Those passages form part of separate means which are later described in detail, and which operatively maintain the front face of each sleeve in close up registration with the adjacent matching face on the side of flange **84** to provide a fluid seal by the registering faces. As an aid to establising such fluid sealing relation, both the front face of each sleeve and the matching face of the flange are optically or nearly optically flat surfaces. The seal which is so formed is such as to preclude all passage of fluid between the registering seal and flange faces except for a minor amount of capillary leakage. Hence, the sleeves **111** and **112** provide radially outward fluid seals for, respectively, the balancing channel **96** and the working channel **95**.

General operation

The operation of the FIG. 1 machine is described herein in terms of only the vane **45***b* thereof. It will be understood, however, that the other vanes of the machine operate in a similar manner.

The machine is run as a pump by employing an external power source (not shown) to rotate the shaft **25**, the rotor **35** and the vanes carried by that rotor. Assume (FIGS. 2 and 6) that the initial position of vane **45***b* is its shown position wherein the angular movement of the vane is about to bring it into the region underneath the high pressure outlet port **87***a*. As the vane moves into that region, it is subjected to equal (high) values of fluid pressure on the angularly opposite sides thereof. Thus, the vane **45***b* is free to move in its slot and is, in fact, cammed rightwardly (locus line **67** of FIG. 6) for the purpose of passing the front end **107***a* of the reaction block unit **55***a*. While the vane **4***b* is so underneath ports **87***a* and is so moving rightwardly, the vane does not work because, in its rotation, it is neither moving with or against a net fluid pressure force exerted thereon in the working channel **95**.

The vane **45***b* continues to be so cammed rightwardly until its angular motion brings it to the center line of the reaction block unit **55***a*. There, the axial position of the vane is such that the rotor groove **40** and the axial cross section of unit **55***a* are completely framed within the notch **46** of the vane (cf. vane **45***b* as shown in FIG. 1). Accordingly, the vane **45***b* is enabled to pass the reaction block unit.

After so passing the centerline of unit **55***a* the vane **45***b* is cammed leftwardly until it returns to its dwell position at that point where the vane in its angular movement is a little beyond the end of projection **106***a* on seal block **90***a*. While the vane is so moving leftwardly, it is mostly under inlet port **85***a*, and so is in a condition where the vane is subjected to equal (low) fluid pressures on its angularly opposite sides and is free to move in its slot. While being cammed leftwardly, the vane does not work because it is angularly moving neither with nor against a net fluid pressure force exerted thereon in channel **95**.

Soon after the vane has reached the point a little beyond the end of projection **106***a* and has there assumed its dwell position, the vane angularly moves beyond the port **85***a* into the interport angle (FIG. 2) between that inlet port **85***a* and the outlet port **87***b*. While in that interport angle, the notch edge **48***b* of the vane bears against the front face **92***a* of the seal block **90***a* to cause the part of the vane in working channel **95** to act therein as an angularly moving fluid seal. When the vane so seals the working channel, the fluid supplied thereto from inlet port **85***a* is impelled by vane **45***b* towards the outlet port **87***b* to discharge into the latter port at a pressure higher than that characterizing the fluid at the inlet port. Thus, within the mentioned interport angle, the vane **45***b* has a pumping action.

The angular movement of the vane in its axial dwell position continues until the vane reaches a point near the end of the projection **105***a* on the seal block **90***a*. At this point, the vane has angularly moved 180° from its assumed initial position and, therefore, has completed one axial camming cycle consisting of one camming interval followed by a dwell interval. During the next 180° of rotation, the vane completes a second similar camming cycle. At the end of the second cycle, the vane has undergone one full revolution. Thereafter, the vane starts a new revolution in which the vane again experiences two camming cycles similar to the ones described. The same is true for every further revolution undergone by vane **45***b*.

During those angular intervals within the which the notch side **48***b* of vane **45***b* bears against the front face of one or the other of the seal blocks **90***a*, **90***b*, the shown cambered shape (FIG. 6) of the notch side **48***b* aids in maintaining a lubricating wedge of fluid between the vane and the seal block contacted thereby.

The sleeves **111** and **112** rotate with the rotor and with the vanes carried thereby by virtue of the described coupling in a keyed manner of the sleeves to the rotor. Thus, the front and rear faces on the sleeves undergo rotation relative to the angularly stationary, registering faces on the camming ring assemblies **63**, **64** and on the flange **84**.

In connection with sleeve **112**, that sleeve is impelled at any moment to rotate with rotor **35** only by those particular vanes of the rotor which at that moment are working, i.e. are being subjected to an unbalance in the fluid pressure forces acting angularly thereon. This being so, the remainder of the vanes are free to move axially in the keyways **113** in the sleeve **112**. Thus, the use of the vanes **45** to key the sleeve **112** to the rotor **35** does not interfere with the axial shifting of the non-working vanes by the camming faces **59**, **60**.

So far, the operation of the FIG. 1 machine has been discussed in terms of a working channel **95** having a constant volumetric capacity for fluid. The capacity of that channel may, however, be varied by turning the hand wheel **72** (FIG. 1) to initiate the previously described chain of events which produces an axial shifting of casing **70** leftward or rightward as desired. When the casing is so shifted, it carries with it the camming ring assemblies **63**, **64**, the sleeves **111**, **112**, the flange **84**, the seal blocks **90***a*, **90***b* and the vanes **45** in the rotor, but the rotor **35**, itself, its groove **40** and the reaction block units **55***a*, **55***b* meanwhile remain axially stationary. Of course, instead of the casing etc. being axially movable and the rotor etc. being stationary, the rotor etc. may be axially movable and the casing etc. be stationary as taught in Lehne U.S. Patent 1,042,696. Some incidental effects of the axial shifting of the casing are to slide both sleeves over the rotor, to produce relative axial movement between the seal blocks **90***a*, **90***b* and the reaction units **55***a*, **55***b* and, lastly, to shift the axial position of the seal blocks in rotor groove **40** relative to the sides **41** and **42** thereof.

As the seal blocks are so shifted leftwardly or rightwardly within the groove **40**, the cross sectional area of the working channel **95** is decreased or increased within those angular intervals of the channel which are bounded by the seal blocks, and in which the vanes have a pumping action. Accordingly, a leftward and a rightward shift of casing **70** serve, respectively, to increase and to decrease the effective volumetric capacity for fluid of the working channel **95**. Since such a shift of the casing does not affect the position of the vanes relative to the seal blocks, after any such shift the vanes continue as before to bear against the seal blocks during the dwell intervals for the vanes. Moreover, since a shifting of casing **70** does not affect the positions of the sleeves **111** and **112** relative to the flange **84**, after any such shift the sleeves continue as before to register in fluid sealing relation with the flange. While an axial shifting of casing **70** does produce a change in the mean axial position of the vanes rela-

tive to the reaction block units, the notches **46** of the vanes are made wide enough to enable the vanes to pass those units for any axial position to which the casing is set. Likewise, while the shift of the casing produces a shift of the inlet and outlet ports relative to working channel **95**, those ports so communicate with that channel that a satisfactory interchange of fluid between the channel and the ports continues to take place for any axial position to which the casing is adjusted.

As the casing **70** and the seal blocks **90***a*, **90***b* are shifted to vary the effective cross sectional area of working channel **95**, a concomitant result is a variation in the opposite sense of the cross sectional area of the balancing channel **96**. While such variation in channel **96** is not of importance in the FIG. 1 machine, the presence in the first instance of channel **96** is important. This is so because the axial pressure of the fluid in channel **96** opposes the axial pressure of the fluid in channel **95** to thereby render the rotor **35** and the casing **70** close to balanced in the axial direction in respect to the fluid pressure forces thereon. Moreover, such axial balancing of pressure forces is realized without any accompanying wasteful diverting of fluid from the working channel **95** to the balancing channel **96**. How such axial balancing without fluid dumping is obtained is a matter later discussed in detail.

The reaction block unit

FIGURE 5 illustrates the details of the reaction block unit **55***a*, the structure of unit **55***b* being similar. As shown, the unit **55***a* comprises an E-shaped structure of which one element is an arcuate cross arm **135** having a front face **136**, a rear face **137** and a circular curvature concentric with the axis of the rotor of the FIG. 1 machine. Projecting axially outwards of the front face **136** is a central arm **140** which furnishes in the unit **55***a* the part which is the true reaction block, i.e.—the part which acts in working channel **95** to obstruct angular flow of fluid therein.

The center arm **140** is flanked on either side by two shorter arms **141** and **142** disposed at opposite ends of cross arm **135** and projecting axially outwards of the front face **136** thereof. The axially extending, outer faces of those two side arms **141** and **142** form the marginal side walls **143** and **144** of the unit **55***a*.

The outer end of the side arm **141** is connected by a web **146** to a portion of center arm **140** back of the front end **107***a* thereof. The web **146** is of smaller radial thickness than the arms **140**, **141**, is disposed radially intermediate the bottom and top faces of those last named arms, and is spaced forwardly of the front face **136** of cross arm **135**, whereby web **146** has a fluid-conducting space **147** beneath it, another fluid-conducting space **148** above the web and a third fluid-conducting space **149** between the web and the cross arm **135**. A similar web **146'** bordered by like fluid-conducting spaces **147'**, **148'**, **149'** is connected between the side arm **142** and the center arm **140**.

To the left (FIG. 5) of the center arm **140**, the cross arm **135** is perforated by three fluid passages **150** extending between the fluid-conducting space **149** and a pocket **151** formed (FIG. 6) in the rear face **137** of the cross arm **135**. A similar set of three fluid passages **150'** is disposed rightwards of center arm **140** and extends through the cross arm **135** and between the fluid-conducting space **149'** and a second pocket **151'** formed in such rear face **137**.

Considering the unit **55***a* and its disposition within groove **40** (FIGS. 2 and 6), the openings of the outlet and inlet ports **87***a* and **85***a* to groove **40** are angularly in registration with, respectively, the fluid-conducting spaces **149** and **149'** provided by the unit. Thus, the port **87***a* is in fluid communication with groove **40** by way of spaces **147–149**, and, similarly, the port **85***a* is in fluid communication with such groove by way of the spaces **147'–149'**. Because of the relatively large axial extents of the spaces **148** and **149** taken together and of the spaces **148'** and **149'** taken together, fluid communication is maintained between such ports and the groove **40** for any axial position to which those ports are shifted as a result of axial adjustment in position of the casing **70**.

Inasmuch as the FIG. 1 machine is a pump, high and low fluid pressures are manifested in groove **40** underneath, respectively, the outlet port **87***a* and the inlet port **85***a*. Both such fluid pressures act axially on the front expanse of unit **55***a* to tend to force it against the side **41** of groove **40**. The same high and low fluid pressures are, however, transmitted through passages **150**, **150'** to, respectively, the pockets **151**, **151'** to there exert on the rear expanse of unit **55***a* a force urging it towards the side **42** of groove **40**. Thus unit **55***a* is largely balanced with respect to the axial pressure forces thereon. Because the effective area over which the same effective pressure acts on the rear and front expanses of unit **55***a* is somewhat less for the rear than for the front expanse (due, among other things, to the fact that the combined cross sectional area of pockets **151**, **151'** (in a plane normal to the rotor axis) is less than the cross sectional area of the entire unit), there is exerted on unit **55***a* a moderate net pressure urging the unit towards groove side **41**. It will be recalled that the unit **55***a* has no connection to casing **70** which would constrain that unit from axial movement relative either to the casing or to the rotor. Hence, such net pressures causes the unit to bear with moderate firmness against groove side **41** irrespective of thermally or wear induced changes in the dimensions of the rotor groove or unit or in the relative axial positions of the rotor and casing. Such bearing of the unit against that groove side is desirable since it stabilizes the unit in the angular position assumed thereby around a radially extending axis of turning **155** (FIG. 5) for that unit, and since it assures that the rear end of the unit will provide a good fluid seal between the chambers **108***a* and **108***b* disposed in channel **96** behind, respectively, the seal block **90***a* and the seal block **90***b*.

Besides acting on the reaction block unit **55***a* in an axial translatory manner, the mentioned high and low pressures tend to exert moments on the unit around axis **155**. Considering the effect of the difference in those two pressures on unit **55***a*, such pressure differential creates the shown net force P_1 (FIG. 5) which acts against the central arm **140**, and which tends to rotate unit **55***a* counterclockwise around the axis **155**. Simultaneously, however, the same pressure differential creates the shown net force P_2 which acts between arms **140** and **141** on the face **136** of cross arm **135** to tend to rotate the unit **55***a* clockwise around the axis **155**. Thus unit **55***a* is designed to produce an approximate equality in the values of the opposing moments around axis **155** which the forces P_1 and P_2 set up. Hence, the tendency of unit **55***a* to cant around axis **155** is minimized.

The stability in positioning of reaction block unit **55***a* both axially and angularly around axis **155** is further improved by the relatively long arcuate extent of the cross arm **135**. As was earlier pointed out, such arcuate extent is almost three times as great as that of each of the seal blocks **90***a*, **90***b*.

The fluid in pockets **151**, **151'** of unit **55***a* is trapped in those pockets by the groove side **41** against which the unit is held by the described net axial pressure acting thereon. A similar fluid trapping takes place in the pockets formed in the rear of unit **55***b*. The fluid received in chambers **108***a*, **108***b* behind, respectively, the seal blocks **90***a*, **90***b* is trapped in those chambers by the side walls of the reaction block units (e.g., walls **143** and **144** of unit **55***a*) which bound the angularly opposite sides of each of those chambers. With both the mentioned pockets and the mentioned chambers acting as fluid traps, excepting for a slight amount of leakage, there is no fluid flow between the inlet and outlet ports through the bal-

ancing channel 96. Accordingly, the FIG. 1 machine operates without any fluid dumping, i.e.—without any wasteful diversion of fluid from the working channel through a by-pass path which includes the balancing channel, and through which a substantial amount of fluid flows between the inlet and outlet ports.

Axial balancing by reaction units and seal blocks

In the described machine, it is preferable to render the rotor balanced in respect to axial pressure forces thereon for the reason among others of avoiding the necessity of using thrust bearings to absorb a net axial force on the rotor. Likewise, it is preferable to have the casing **70** axially balanced in order to minimize the effort needed to adjust the axial position of the casing. The mode of obtaining such axial balancing of the rotor and casing will now be discussed.

Within the angular intervals of groove **40** which are occupied by the reaction block units **55***a* and **55***b*, the pressurized fluid in the groove exerts (FIG. 6) a rightwardly directed force on the groove side **42**. This force is opposed by a leftwardly directed force exerted on the groove side **41** by the pockets (e.g., **151**, **151'** of **55***a*) formed in the rear faces of the reaction block units. Because the total cross sectional area of those pockets in a plane normal to the rotor axis is somewhat less than the total cross sectional area of the entire units **55***a*, **55***b* (and ignoring leakage effects in channel **96**), the mentioned leftward force is evidently somewhat less than the rightward one. Such leftward fluid force is, however, supplemented by the described net pressure force acting on the units **55***a*, **55***b* to hold those units against the groove side **41** and transmitted from those units to such groove side. Therefore, over the angular intervals occupied in groove **40** by the reaction block units the total leftward force on groove side **41** is nearly or exactly equal to the total rightward force exerted on groove side **42**, wherefore, the rotor is balanced with respect to the axial pressure forces acting thereon.

Inasmuch as the units **55***a*, **55***b* are not secured to casing **70** by any connection which would transmit axial force from the former to the latter, over those same angular intervals occupied by those units **55***a*, **55***b*, the pressurized fluid in groove **40** exerts no axial force on the casing, and, hence, such casing is likewise balanced over those angular intervals. Because the angular interval occupied by each reaction block unit is, as stated, almost three times larger than that occupied by each seal block, it follows that both the rotor and the casing are axially balanced over almost three-quarters of the circumference of the rotor groove. Therefore, any axial unbalance, of the rotor and casing can ocur only within the remaining one-quarter (or thereabouts) of the groove circumference which is taken up by the angular intervals occupied by the seal blocks.

FIGURE 7A depicts schematically an instant in the operation of the FIG. 1 machine when the vanes **45***f* and **45***e* are at opposite ends of the seal block **90***a*. At this instant (represented by point **160** in FIG. 7F), the same intermediate pressure exists in the interspace **170** ahead (in working channel **95**) of vane **45***e* and in the chamber **108***a* behind seal block **90***a*. Hence, at such instant, the seal block **90***a* is, for practical purposes, axially balanced in respect to the fluid pressure forces thereon.

In FIG. 7B, the angular movement of rotor **35** has carried vane **45***f* to a location where the high pressure fluid manifested under outlet port **87***b* (FIG. 2) and within interspace **169** ahead (in channel **95**) of vane **45***f* has been enabled to flow into chamber **108***a* through the passage **56** (FIG. 1) defined between the bottom **49***f* of the vane notch **46***f* in vane **45***f* and the bottom **43** of the rotor groove **40** (cf. vane **45***b* in FIG. 1). Once in that chamber **108***a*, the high pressure fluid is further enabled to flow through the passage **97***a* in seal block **90***a* to the mentioned interspace **170**. At this time, the advancement of the vane **45***f* from its FIG. 7A position to its FIG. 7B position decreases the area of the front face **92***a* (of seal block **90***a*) bordering interspace **170** to a size which is somewhat less than that of the area of the rear face **93***a* (of seal block **90***a*) which is exposed to the high pressure in chamber **108***a*. Therefore, at the instant depicted by FIG. 7B, there is exerted on seal block **90***a* a leftwardly-directed, net axial pressure force represented in FIG. 7F by the point **161**.

Subsequently, the continued advancement of vanes **45***f* and **45***e* causes a further progressive decrease in the area of front face **92***a* which is exposed to the high pressure in interspace **170**. Meanwhile, the entire area of rear face **93***a* is continuously exposed to the high pressure in the chamber **108***a*. Hence, the net leftward axial force on seal block **90***a* builds up in amplitude to the peak represented by point **162** in FIG. 7F.

Very shortly after such peak is reached, the machine reaches the condition depicted in FIG. 7C wherein the vane **45***e* becomes centered in relation to the passage **97***a* through the seal block, the pressure in chamber **108***a* drops to an intermediate value and the axial unbalance on seal block **90***a* transiently drops to the zero value represented by point **163** in FIG. 7F. When, however, the vane **45***e* advances only slightly farther to its position shown in FIG. 7D, the uncovering by such vane advance of the passage **97***a* permits a flow of low pressure fluid from the interspace **171** (behind vane **45***e*) through passage **97***a* into chamber **108***a*. At this instant, the high pressure fluid in interspace **170** is still acting on almost half of the area of the front face **92***a* of seal block **90***a*. Hence, as represented by point **164** in FIG. 7F, the seal block is subjected to a peak value of a net axial force which acts rightwardly.

The continued advance of vane **45***e* produces a progressive decrease and increase in the areas of front face **92***a* which are exposed to, respectively, the high pressure in interspace **170** and the low pressure in interspace **171**. It follows that there is a dropping off from its peak value of the net rightward force on seal block **90***a*. Thus, for example, for the condition represented by FIG. 7E, such force has dropped to the value represented in FIG. 7F by the point **165**. Such dropping off of the net rightward axial force continues until vane **45***e* has reached the same position relative to seal block **90***a* as that shown for vane **45***f* in FIG. 7A. At this instant (represented by point **166** in FIG. 7F), the net axial pressure force on seal block **90***a* has gone through one complete cycle and is now starting a new cycle identical with the one just described.

From the above, it will be seen that each of seal blocks **90***a*, **90***b* transmits to casing **70** an axial force characterized by a sawtooth variation in amplitude and by alternation in its direction of action. The rotor **35** is subjected to an axial force similar to that on casing **70** excepting that the force on the rotor is 180° out of timephase with the force on the casing in that the former force acts rightwardly when the latter force acts leftwardly, and conversely. Because both forces are developed only in the relatively small angular intervals occupied by the seal blocks, and because the alternation in direction of both forces reduces to zero the average value of each thereof over all but a very short time period, neither force is, at worst, particularly troublesome. To the extent they might be found troublesome, the effects of those forces can be reduced as desired by increasing the number of vanes per seal block and/or by increasing the rotor speed.

When the number of vanes per seal block is increased, the inter-vane angle is decreased to permit a corresponding decrease in the angular interval which is occupied by each seal block, and of which the lower limit of practical size varies directly with such intervane angle. Evidently, however, the amplitude of the mentioned axial forces varies directly with the size of the seal block angular intervals from which those forces arise. Hence, an in-

crease in the number of vanes per seal block and a concomitant decrease in the seal block intervals will serve to decrease the amplitude of the transient axial forces exerted on the casing and on the rotor.

As a further effect of increasing the number of vanes per seal block, for a given rotor velocity, such an increase produces an increase in the repetition frequency of the sawtooth axial forces. The higher such frequency, the lesser the effect of such forces on the rotor mass and on the casing mass which become progressively less responsive to such cyclical forces as the repetition frequency thereof increases. Increasing the rotor velocity is, of course, another way by which the repetition frequency of such sawtooth forces may be increased.

Still another way to minimize the effect of the sawtooth axial force on the rotor is to damp out the transient variations of such force by means now to be described.

Damping mechanism

As earlier mentioned, the FIG. 1 machine contains equiangularly spaced conduits **31** which extend through rotor **35** to connect the interspace between that rotor and sleeve **111** to the central bore **30** in shaft **25**. The bore **30** is in turn connected by way of a radial conduit **180** through shaft **25** to a passage **181** formed at an axially central position in an annular damper disc **182** threadedly mounted on the shaft to rotate therewith. As shown, the passage **181** extends radially through disc **182** to terminate at a clearance **186** between the radially outward cylindrical surface of disc **182** and a registering cylindrical surface formed on the inside of end plate **21**.

The left and right hand annular sides of damper disc **182** are spaced from, respectively, an annular surface formed on the interior of stuffing box member **26** and an annular surface of a ring member **185** threadedly received in end plate **21**. Such spacing creates a pair of annular, fluid-receiving, damping chambers disposed on axially opposite sides of the disc **182**. To increase the fluid tightness of those chambers, the rotating disc **182** has radially inward hub portions projecting axially outwards in opposite directions from the center of the disc and having exterior cylindrical seal surfaces separated only by the small clearances **187** and **188** from, respectively, an interior cylindrical seal surface on the stuffing box member **26** and an interior cylindrical seal surface on the ring member **185**.

In operation, fluid from the interspace between rotor **35** and sleeve **111** is supplied through conduit **31**, bore **30** and conduits **180** and **181** to the clearance space **186**, from thence into the damping chambers **183** and **184**, and from those chambers through the clearances **187**, **188** to sump regions (not shown) for the fluid. Because the fluid pressure at the mentioned interspace is substantially higher than the fluid pressure in such sump regions, there occurs from the former to the latter a small but steady flow of fluid by which the damping chambers **183**, **184** are continuously replenished with fluid. If desired, a fluid collection recess may be formed in one of the reaction block units, and fluid may be conveyed from such recess to chambers **183**, **184** through conduit means (not shown) passing through elements **70**, **20** and **21**.

The described mechanism acts as follows to provide damping. When the rotor **35** is subjected to a transient axial force urging the damper disc **182** leftward, the fluid in damping chamber **183** absorbs that force by being discharged from **183** either through the clearance **187** or through the clearance **186** to the chamber **184**. Conversely, when the rotor is subjected to a transient axial force urging disc **182** rightwardly, the fluid in chamber **184** absorbs that force by being discharged from the last named chamber either through clearance **188** or through clearance **186** back to chamber **183**. Since the chambers **183** and **184** are continuously being replenished with fluid, the damper disc **182** and its assoicated chambers are continuously effective to absorb either rightwardly or leftwardly acting transient forces on the rotor of the FIG. 1 machine.

Pressure effects on the vanes

Turning back to FIGS. 7A–7E, within the angular intervals occupied by the reaction block units **55***a*, **55***b*, the pressure effects on the vanes are the same as those described in my aforementioned U.S. application and aforementioned British patent. Thus, when within such an interval, because any such vane, (e.g. **45***d*) has equal fluid pressure forces acting axially in opposite directions on the opposite sides of its notch, each vane at that time will be axially balanced in respect to the fluid pressure forces thereon. Further, any such vane when within such an interval has equal fluid pressures on the angularly opposite sides thereof, and, hence, the vane is angularly balanced in respect to fluid pressure forces. Still further, any such vane may be radially balanced with respect to such forces by providing one or more fluid passages extending in the vane between the radially inner and outer margins thereof in the manner described in my aforementioned application and aforementioned British patent.

When the vanes are in the angular intervals occupied by the seal blocks, the pressure effects become more complex. Considering, for example, the vane **45***e* as shown in FIGS. 7A–7E, that vane is axially balanced or nearly so when in the position therefor which is shown by FIG. 7A. An instant later, however, when vane **45***e* is in the position shown by FIG. 7B, the vane becomes exposed on its angularly opposite sides to a pressure differential caused by a charging of the interspace **170** with high-pressure fluid while the fluid in interspace **171** remains at low pressure.

Such pressure differential forces a small amount of fluid from the interspace **170** to the interspace **171** through a leakage path between the notch side **48***e* of vane **45***e* and the front face **92***a* of seal block **90***a* against which the notch side **48***e* bears. The fluid in that leakage path acts on the vane notch side with an effective pressure which is less than that in interspace **170** (because, in the leakage path, the pressure is characterized by a dropping gradient from interspace **170** to interspace **171**), but which, nonetheless, exposes the notch side **48***e* to an axial force tending to move vane **45***e* leftwardly. That force is in part a function of the profile of the chamber of notch side **48***e*. Such leftward urging of the vane by the intermediate pressure in the leakage path is, however, more than overcome by the fact that, simultaneously, the rotor slot **44***e* (which contains vane **45***e*) becomes charged with high-pressure fluid which acts axially on the vane notch side **47***e* to tend to move vane **45***e* rightwardly. The result is that the vane is subjected to a net axial pressure force which urges the vane rightwardly, and which maintains the notch side **48***e* of the vane in contact under moderate force with the front face **92***a* of the seal block **90***a*.

Thus, as each vane reaches the beginning of one of the angular intervals in which it works, the vane is automatically positioned by the pressure condition to which it is subjected to bear as it should against the front face of the seal block with which the vane cooperates during the working interval. Such automatic positioning of the vane by pressure is desirable since it assures that the vane will bear against the seal block (so as to be properly positioned for working) despite some play between the vane and the camming faces **59**, **60** (FIG. 1) by which the vane is mechanically positioned. If there is any such play, the described "vane-catching" projections **106***a*, **106***b* (FIG. 6) on the front faces of the seal block prevent the described net pressure force from displacing a vane so far that its vane notch overshoots the axial position of the front face of a seal block being angularly approached so as, thereby, to cause the vane upon angularly reaching such block to be jammed against its side.

Turning now to FIGS. 7C and 7D, as the vane **45***e* moves past the passage **97***a* in seal block **90***a* (FIG. 7D), the rightwardly acting pressure force on vane notch side **47***e* drops sufficiently in value to be overcome by the leftwardly directed pressure force still being exerted on the vane notch side **48***e*. Thus, the net axial pressure force one the vane notch sides of vanes **45***e* is such as to tend to disengage the notch side **48***e* of that vane from the face **92***a* of the seal block **90***a*. Even so, the vane still remains in its proper working position (i.e., with its notch side **48***e* against seal block face **92***a*) because of an additional pressure effect which was initially developed when the vane first started to work (FIG. 7B), and which continues until the vane **45***e* arrives at the location shown for vane **45***f* in FIG. 7A to thereby reach the end of its working interval. Such additional pressure effect is that, as stated, the vane **45***e* when working is subjected on its angularly opposite sides to a pressure differential developed by the high and low pressure fluid in, respectively, the interspaces **170** and **171**. Such pressure differential urges vane **45***e* against the side of its slot **44***e* with enough friction that the force necessary to move the vane **45***e* leftwardly against such friction is greater than the net pressure force which acts on the notch sides of the vane and which tends to displace it leftwardly. Accordingly, throughout its entire working interval, the vane **45***e* is maintained properly positioned by the pressure effects thereon.

As the vanes of the FIG. 1 machine are moved axially during the described camming intervals therefore, the notch sides of each vane impel back and forth the fluid in the rotor slot within which the vane is seated. If such fluid is inexpansible and should happen to become trapped without any outlet between a notched side of the vane and the center solid portion of a reaction block unit towards which that notch side is moving, then that trapped fluid would block further axial movement of the vane to produce a locking thereof. Such vane-locking problem is avoided in the present machine in the same way as in the machine described in my aforementioned U.S. application and my aforementioned British patent, namely, by permitting the fluid impelled by the vane notch side to flow from one side to the other of the reaction block unit through the passage **56** (FIG. 1) formed between the bottom margin **49** of the vane notch **46** and the bottom of the reaction block unit.

If the presently described machine should happen to be stopped in the position shown in FIG. 7C wherein the vane **45***e* blocks the flow of fluid through the passage **97***a* in seal block **90***a*, there would occur an undesirable "dead center" effect. Such effect may be avoided by modifying the seal blocks to have the construction illustrated in FIG. 8. A fluid passage is provided through the seal block of that figure by two holes **190***a*, **191***a* instead of by the single hole **97***a* which is shown in FIG. 1. The centers of the holes **190***a*, **191***a* are spaced apart by the distance between the two angularly opposite sides of a vane. Hence, there is no angular position of a vane relative to a seal block in which the vane will block entirely the flow of fluid through the seal block.

Structure and operations of the seal sleeves

FIG. 4 is a cross sectional view taken through a bottom portion of the sleeve **111** as seen in FIG. 1, the cross section plane for FIG. 4 being specifically shown in FIG. 3. The purpose of FIG. 4 is to show in detail the preferred structure for the seal sleeves. It is to be understood, however, that such sleeves need not have as complex a structure as that shown, but instead, may, for example, comprise sleeves of which the equiangularly spaced fluid passages **130** (FIG. 3) are simple holes extending in a straight line between the front and rear faces of each sleeve. Alternatively, the seal sleeves **111**, **112** may be simple sleeves which have no fluid passages therein at all.

Coming now to the sleeve structure shown by FIG. 4, the sleeve **111** contains at the end thereof towards flanges **84** a tubular liner **200** of different material (as, say, harder material) than the rest of the sleeve. It is such liner which provides for sleeve **111** the annular face **121** which (FIG. 1) registers against the matching annular face **123** of flange **84** to form therewith a fluid seal around the outer periphery of the balancing channel **96**. By providing such face **121** by an element of hard material, that sleeve face will wear better as it rubs against the flange face **123** in the course of the rotation of sleeve **111** relative to the angularly stationary flange **84**.

Although very little fluid passes between the registering faces of the sleeve and of the flange, the capillary leakage which does occur in the interface therebetween will act against the area of sleeve face **121** with a pressure which urges the sleeve **111** outwardly so as to tend to open up the fluid seal formed by those faces. Even so, because the pressure in the leakage path through the interface between faces **121**, **123** has a dropping gradient in the radially outward direction, the effective value of such pressure on any angularly positioned part of the area of sleeve face **121** is always less than the fluid pressure at the same angular position in balancing channel **96**. Accordingly, the tendency of such leakage path pressure to force sleeve **111** leftwardly can be more than overcome in a manner as follows.

As stated, the sleeve **111** contains a plurality of axially running passages **130** (FIG. 3). As shown by FIG. 4. each such passage is connected by a slanted conduit **205** to the interface between sleeve face **121** and flange face **123** at a zone of such interface which is relatively close to the balancing channel **96**. At the place where the conduit **205** taps into the interface, the pressure of the fluid has a value close to that of the fluid pressure manifested in channel **96**, i.e., is at a value which is still relatively high compared to the effective pressure at the same angular location by which the fluid in the interface acts on the sleeve face **121** to urge the sleeve **111** leftwardly.

Within the sleeve **111**, the passage **130** extends rearwardly to an enlargement thereof provided by a deep counter bore **206** of greater diameter than such passage, and by, also, an outer shallow counter bore **207** of greater diameter than the counter bore **206**. Received within the enlargement is a bushing **208** having a stem portion **209** in counter bore **206** and a larger diameter head portion in counter bore **207**. The bushing **208** has formed therein an axially running central passage **210** which continues the passage **130**, and which extends from the rear of bushing **208** to a pocket **211** formed in the front face **120** of the bushing (i.e., formed in what is, in effect, the rear face of the sleeve assembly). As shown, a compression spring **212** is inserted in counter bore **206** behind the bushing **208** to urge it outwardly so as to maintain its front face **120** in contact with the annular face **122** formed on the camming ring assembly **63**. As further shown, an O ring **213** is carried by the stem portion of the bushing and is compressed between that stem portion and the interior of counter bore **206**.

The described sleeve balancing structure operates as follows. In the absence of pressure in balancing channel **96**, the resilient force of spring **212** maintains the face **120** of bushing **208** against the face **122** of camming ring assembly **63** and, also, the face **121** of sleeve **111** against the face **123** of flange **84**, When the channel **96** contains pressurized fluid, such fluid under only slightly less pressure than it has in **96** is transmitted through conduit **205**, passage **130** and passage **210** (in bushing **208**) to the pocket **211** formed in the bushing. The fluid which so reaches such pocket produces on bushing **208** and the counter bore **206** an axial force which acts with the spring **212** to urge the entire sleeve **111** rightwardly.

Thus, there is exerted on the sleeve **111** a total rightward force equal to the sum of the forces developed by the pressurized fluid in all the bushing stem bores plus

the sum of the resilient forces developed by all of the springs **212** contained by sleeve **111**. While the total of the cross sectional areas (in a plane normal to the rotor axis) of all such bushing stem bores may be less than the area of the sleeve face **121** which is urged leftwardly by the effective pressure in the interface between that sleeve face **121** and the flange face **123**, because such effective pressure is less in value than the average value of the fluid pressures developed in all the bushing stem bores, the total rightward force on sleeve **111** from such stem bores may be made equal to or even greater than the leftward force exerted on the sleeve face **121**. Hence, even when the balancing channel contains pressurized fluid, the sleeve face **121** may be urged towards the flange face **123** with about the same value of force as that by which the sleeve face was so urged when there was no pressurized fluid in channel **96**, so that only the springs **212** in sleeve **111** urged the sleeve face towards the flange face. It follows that, when pressurized fluid is in balancing channel **96**, such fluid will not open up the fluid seal provided by sleeve face **121** and flange face **123** but, instead, such fluid may be employed to make that seal even tighter.

As the sleeve rotates, any particular conduit **205** thereof will be carried in alternation through the angular intervals occupied by seal blocks and the angular intervals occupied by reaction block units. Because there are different fluid pressure values in the reaction unit intervals and in the seal block intervals, the balancing structure associated with such a particular conduit **205** will experience cyclical fluctuations in the fluid pressure to which the structure is subjected. Such pressure fluctuations for any one such structure do not, however, detract from the described holding by fluid pressure of the sleeve face **121** against the face **123** of flange **84**. This is so for the reason that such holding effect is produced by the average of the individual holding actions of all the balancing structures in the sleeve, and considering such average action, there is over any time period and at any instant in that period, a canceling out with each other of the force variations arising from the pressure fluctuations manifested in the individual balancing structures, whereby such force variations are largely eliminated from the average action.

The pressurized fluid in channel **96** acts radially outwards on sleeve **111** so that, at high pressure, the end of the sleeve towards flange **84** is subjected to stress which may become unduly high and lead to a deformation of the sleeve in the form of radial expansion of the front end thereof. Since the liner **200** (FIG. 4) of the sleeve may be of a hard material which is likely to be brittle, it is desirable to relieve such radial stress by offsetting the outwardly acting force by a force acting radially inwards on the exterior of the liner.

Such inward force is developed by providing in the liner **200** a plurality of equiangularly spaced holes **230** (FIG. 9) angularly disposed around the sleeve at positions intervening those of the conduits **205**. As shown in FIG. 9, each hole **230** extends from the interior surface of the liner through the liner to a small pocket **231** formed in the liner's exterior surface. Each such pocket is rendered fluid tight by being outwardly bounded by the front end portion of sleeve **111**.

As the width of the balancing channel **96** is varied by axially adjusting casing **70** and the seal blocks **90***a*, **90***b* in the manner heretofore described, a corresponding variation takes place in the inner peripheral area of liner **200** which is subjected to the high pressure in that channel. Such areal variation in turn produces a corresponding variation in the total radial force exerted outwardly on the liner. In order to keep the offsetting inward force exerted on the liner (from pockets **231**) in step with such variation in the outward force exerted thereon, the holes **230** have axial locations which vary from one hole to the next in the manner shown by FIG. 14.

If channel **96** has been adjusted to be narrow so that the outward force on liner **200** is correspondingly small, most of the axially distributed holes **230** will open inwardly on the interface between rotor **35** and sleeve **111** to thereby be cut off from the high pressure fluid in channel **96**. Thus, for a narrow channel **96**, high fluid pressure can be considered as manifested only in those pockets **231** which are connected to channel **96** through the minority of such holes **230** whose inward openings are not covered by rotor **35**. Because those last named pockets are a minority of all the pockets, the total inward force on liner **200** is small so as to be proportioned to the small outward force thereon. As, however, the width of channel **96** becomes progressively larger to thereby increase the outward force on the liner, the number of uncovered holes **230** becomes progressively larger to activate a greater number of pockets **231** to thereby match such increasing outward force on liner **200** with a progressively increasing inward force thereon.

FIG. 15 depicts a modification of the vane **45** and of the seal blocks which provides another way of overcoming the problem of the radially outward force exerted on the seal sleeves. In the instance of FIG. 15 the force in question is the force exerted on sleeve **112** from pressurized fluid in the working channel **95**. As shown by FIG. 15, the seal block **90***a* is so proportioned in its axial dimension and is so secured to the flange **84** that the front face **92***a* of the seal block is overhung by a portion of the inner cylindrical surface **240** of the flange. Thus, the outward bound for channel **95** is now provided only in part by the front end portion of sleeve **112**, the remainder of such outward bound being provided by the flange **84**. Evidently such construction relieves the radial stress on sleeve **112** in that, of the total outward force developed by the pressurized fluid in channel **95**, a substantial fraction of that force is received by the flange **84** which is massive and well adapted to absorb the force so exerted thereon.

By increasing the overhang of flange **84** relative to the face **92***a* of seal block **90***a* so as to increase the outward bounding of channel **95** by that flange and to correspondingly decrease the outward bounding of such channel by the sleeve **112**, the outward force on that sleeve can be reduced as desired. For example, as shown in FIG. 16, the flange **84** may provide the entire outward bound for the working channel **95**, in which case there is no significant outward fluid pressure force on the sleeve **112**.

In FIGS. 15 and 16, the modified sleeve **112** is similar to sleeve **111** in that the sleeve **112** is coupled to rotor **35** by a key and a keyway (neither shown) similar to the key **114** and keyway **116** described in connection with sleeve **111** (FIG. 3). A further similarity is that in the sleeve **112** of FIGS. 15 and 16 the rotor-carried vanes **45** no longer extend into keyways **113** formed (FIG. 1) on the inside of sleeve **112**. The reason for the omission of such keyways is that, for the construction of FIGS. 15 and 16, the outer margins of the vanes **45** must be radially inward of the inner cylindrical surface **240** of flange **84** (and, thus inward of the inside surface of sleeve **112**) in order for the vane notch edge **48** of each vane to abut against the front face **92***a* of seal block **90***a* during the working interval for the vane.

By not having the radially outer margin of each vane seated in a keyway in sleeve **112**, there is lost the previously described feature of a three-sided support for the portion of the vane inserted into working channel **95**. The absence of such three sided support feature can, however, be fully compensated for by making the vane more massive.

The use of the discussed seal sleeves in a machine of the sort described has numerous advantages of which some are as follows. First, since fluid between the rotor **35** and the seal sleeves **111**, **112** does not produce on the rotor the viscous drag which would occur and vary inversely with the rotor-sleeve fit if those sleeves were

stationary, there can be a closer fit between such rotor and the surrounding sleeves to the end of minimizing axial fluid leakage between the rotor and the sleeves. Note that the fluid between the rotor and each sleeve tends to couple that sleeve and the rotor together for rotation, wherefore such fluid may be used in lieu of other means as a keying means between such sleeve and the rotor. Further, since the vanes do not project radially outward of the rotating assembly, the machine is characterized by the absence of a rubbing of the vanes on the stator interior or of a shearing by such vanes of a lubricating film of fluid on the stator interior. Third, inasmuch as the seal sleeves surround the vanes, no problem arises from radially outward throwing of the vanes by centrifugal force. Still further, the described sealing action of the sleeves is an important advantage because, with such sealing action, one doesn't need a close fit between the sleeves and casing **70** to provide a seal, and, therefore, such fit can be enlarged to reduce the viscous drag on the sleeves of fluid in the interspace between the sleeves and casing **70**. All of the advantages just enumerated are realized by the discussed seal sleeves when incorporated in a machine of a constant volumetric fluid capacity as well as when incorporated in the presently described type of machine, namely, one having a variable volumetric capacity for fluids. In a constant displacement machine, the friction between the seal sleeves and the faces **122, 126** can be eliminated by using nuts or other adjustable means to couple such sleeves to the rotor in a manner whereby the sleeves are out of contact with faces **122, 126** but can be axially adjusted relative to the rotor to control the closeness of fit between the seal sleeves and the flange **84**.

It is to be understood that the above detailed description of the preferred structure for sleeve **111** is also applicable to the sleeve **112**. It is further to be understood that the modifications shown in FIGS. 15 and 16 and discussed in connection with the sleeve **112** are modifications which likewise can be made in connection with the sleeve **111**.

Seal block modifications

Earlier, a description was given of various ways of reducing the effect on the rotor and on the casing of the sawtooth axial forces generated during angular movement of the vanes past the seal blocks **90***a* and **90***b*. FIGS. 10, 11 and 12 illustrate schematically still other ways by which such effect can be reduced. In these figures, the machine under discussion is seen from the side opposite from that from which it is viewed in FIG. 1. Hence, in FIGS. 10–12, the working and balancing channels **95** and **96** are seen as positioned on, respectively, the lefthand side and the righthand side of the rotor groove **40**.

Referring first to FIG. 10, in that figure there is shown as a replacement for each of the seal blocks **90***a*, **90***b* (FIG. 1) a modified seal block construction exemplified by a seal block **250** attached by suitable means (not shown) to the inside of flange **84** (FIG. 1), the seal block **250** having a front face **251** and a rear face **252**. The face **252** has formed therein a rectangular recess or guideway **253** within which is seated an axially slidable seal block insert **255**. As shown, the rear of insert **255** projects axially outward of the seal block face **252** into balancing channel **96** to bear against the side **41** of rotor groove **40**. By so bearing against groove side **41**, the insert **255** divides the angular interval behind block **250** of channel **96** into two fluid-receiving compartments **256** and **258** which are each of the same angular width as the insert **255**. A third fluid-receiving compartment **257** (of the same angular width as **256** and **258**) is formed in the guideway **253** forward of the insert **255**.

The three compartments **256, 257** and **258** are connected to working channel **95** by, respectively, the fluid passages **260, 261** and **262** of which each extends through seal block **250** along the center line of the compartment associated with that passage. The passage **261** is continued by another passage **263** extending through insert **255** along the center line thereof and between the front of the insert and a pocket **264** formed in the rear face of the insert. The front portion of passage **263** is enlarged by a counter-bore **265** in which is seated one end of a compression spring **266** of which the other end bears against the forward closure wall of the guideway **253**.

For a no-pressure condition of rotor groove **40**, the resilient force of spring **265** urges the rear of insert **255** against the groove side **41** to assure that the insert provide a good fluid seal between the compartments **256** and **258**. After the rotor groove **40** has become filled with pressurized fluid, the pressure thereof in chamber **257** urges the insert **255** towards groove side **41** with a force which might be unduly great if not at least partly offset. Such offsetting of the last-named force, is, however, provided by the passage **263** and the pocket **264** which serve to oppose the fluid pressure force acting from compartment **257** or insert **255** with a fluid pressure force acting from pocket **264** on that insert. In other words, the insert **255** is largely axially balanced in respect to the forces thereon in much the same way as axial balancing is (as previously described) provided for the seal sleeve **111** by the sleeve balancing structure which was discussed in connection with FIG. 4.

Turning now to the sawtooth axial forces which are generated during the angular movement of, say, vane **45***e*, past the seal block **250**, the total angular extent of that seal block can, for practical purposes, be considered as divided into three separate sections of which the angular interval of each is coextensive with a respective one of the three compartments **256, 257** and **258**, and of which each section contains a respective one of the fluid passages **260, 261** and **262**. As the vane **45***e* approaches, reaches and passes in succession the passages **260, 261** and **262**, the three seal block sections which respectively contain those three passages each acts in turn as the source of a sawtooth axial force arising from such section in the same way as the force shown in FIG. 7F arose (as previously described) from the angular movement of vane **45***e* past the seal block **90***a*. Since, however, the three sections of seal block **250** each occupy only one third of the whole angular interval occupied by seal block **90***a*, and since the amplitude of the sawtooth force varies directly as the size of the angular interval from which it arises, evidently the three sawtooth force variations produced by movement of vane **45***e* past seal block **250** will each have an amplitude reduced by a factor of about three relative to that of the sawtooth force arising from the seal block **90***a*. Moreover, the repetition frequency of the sawtooth variations arising from seal block **250** will be greater by a factor of about three than the repetition frequency of the sawtooth variation arising from seal block **90***a*. Accordingly, as compared to the seal block **90***a*, a seal block of the construction shown in FIG. 10 serves to reduce the effects of the described sawtooth axial force on the rotor and on the casing both by decreasing the amplitude of that force and by increasing the repetition frequency thereof.

In the seal block construction shown in FIG. 10, the insert **255** is subjected to a turning moment at that time when the rear end of the insert is acted upon by oppositely directed but unequal forces generated by, respectively, high pressure fluid in compartments **258** and low pressure fluid in compartment **256**. Such turning moment on the insert is eliminated by the seal block **270** which is shown in FIG. 11, and which is generally similar to the seal block **250** excepting as follows.

In the seal block **270**, the angular width of the insert **271** is twice that of each of the compartments **272, 273** which are sub-divisions of the angular interval behind block **270** of the balancing channel **96**, and which lie to either side of the insert **271**. Further, the front face **274** of seal block **270** contains only two openings **275**

and 276 for receiving fluid from the working channel 95. As in the case of FIG. 10, one of those openings (namely, 276) connects the working channel to a fluid-receiving compartment 277 formed in the guideway 278 for insert 271 forward of the front end of the insert. The other opening (namely, 275) is, however, connected by fluid passages 279, 280 and 281 in seal block 270 to each of the compartments 272 and 273 on either side of the insert 271. Those last-named passages may be provided in the seal block 270 by chasings on the lower or on the upper surface thereof or in some other appropriate manner. Finally (and as a minor consideration), the insert 271 is biased rearwardly by two compression springs.

As the vane 45*e* moves past the opening 275 both of the compartments 272 and 273 switch simultaneously from a high pressure to a low pressure condition, whereby a change in the fluid pressure condition of those compartments does not exert any turning moment on the insert 271. When further movement of the vane 45*e* carries it past opening 276, the resulting switch of compartment 278 form a high pressure to a low pressure condition does not generate any turning moment on the insert. Thus, at no time during the movement of vane 45*e* past seal block 270 is the insert 271 subjected to a turning moment.

Aside from the consideration just mentioned that the insert 271 does not experience a turning moment, the seal block 270 acts to reduce the amplitude of and to increase the repetition frequency of the discussed sawtooth axial force in much the same manner as does the seal block 250 of FIG. 10. In other words, since, as concerns block 270, one sawtooth variation of such force arises from the compartments 272 and 273 which together span only one half of the total angular extend of block 270, and since another sawtooth variation of such force is next generated over that remaining half of block 270 which is spanned by the compartment 277, the seal block 270 performs relative to the seal blocks 90*a* and 90*b* (FIG. 1) to reduce the amplitude of such sawtooth force by a factor of about 2, and to increase the repetition frequency of such force by the same factor of about 2.

FIG. 12 illustrates a seal block which extends the principles involved in the construction of the FIG. 11 block so as to eliminate any turning moment on the seal block insert and so as, at the same time, to reduce by a factor of about three the amplitude of the sawtooth axial force and to increase by a factor of about three the repetition frequency of that force. From the descriptions previously given in connection with FIGS. 10 and 11, the structure and operation of the FIG. 12 seal block should be self-evident.

FIG. 13 schematically illustrates still another way by which the mentioned sawtooth axial forces on the rotor 35 and the casing 70 can be reduced in amplitude and increased in repetition frequency. When, as shown by FIG. 2, the fluid passage 97*a* and 97*b* of, respectively, the seal blocks 90*a* and 90*b* are on the center lines of those seal blocks, the traversal of vane 45*e* past block 90*a* and the traversal of the diametrically opposite vane 45*a* past the block 90*b* are actions generating two separate sawtooth axial forces which are in time phase because each vane passes at the same moment the fluid passage in the associated seal block. Inasmuch as those two forces are in time phase, the resultant force on the rotor and on the casing is twice the amplitude of the individual sawtooth forces.

When, however, as shown by FIG. 13, the fluid passages 97*a* and 97*b* are angularly positioned in the seal blocks 90*a* and 90*b* so that those passages are, respectively, lagging and leading the center lines 290*a*, 290*b* of those blocks with reference to the direction of rotation of the vanes 45, then, as the vanes 45*e* and 45*a* traverse, respectively, the blocks 90*a* and 90*b*, the sawtooth forces generated by such traversals will be characterized by a difference in time phase. As a consequence of such difference, the resultant force on the rotor and on the casing will have an amplitude less than twice that of the individual sawtooth forces, and such resultant force will have as a major frequency component a repetition frequency which is twice that of the individual sawtooth forces.

As an alternative to angularly shifting the passages 97*a*, 97*b*, away from the center lines of their seal blocks, a difference in the time phase of the sawtooth forces generated by the traversals of individual vanes over those blocks can be produced by maintaining passages 97*a*, 97*b* on the block center lines, but by using a non-integral number of vanes per seal block. As an example, when the eight vanes 45 (FIG. 2) are replaced by nine equiangularly spaced vanes so that there are four and one half vanes per seal block, such a difference in time phase will be created because no two vanes will register at the same moment with, respectively, the fluid passage 97*a* and the fluid passage 97*b*.

Whether the difference in time phase of the sawtooth axial forces is produced by using off-center fluid passages or by using a non-integral number of vanes per seal block, if only two seal blocks are used, the sawtooth forces which arise therefrom will not be fully time-balanced in their effect on the rotor and on the casing. This is so because, although such forces are spatially balanced against each other in that they act on the rotor and casing at diametrically opposite zones, the two forces do not act on the rotor and casing in the same way at exactly the same time.

While the described lack of a time balance is not usually troublesome, if such a time balance is desired, it may be realized by employing a plurality of pairs of diametrically opposite seal blocks individually spaced in an equiangular manner around the rotor, e.g., four seal blocks spaced at 90° intervals. With such arrangement, when the fluid passages respective to the blocks in each pair are disposed diametrically opposite each other, but, considering the seal blocks individually, such passages vary in their individual angular positions relative to the center lines of the associated blocks, then, while the sawtooth forces arising from each seal block pair will be synchronous, a difference in time phase will exist between the sawtooth forces generated by different pairs of seal blocks. When, instead of moving the fluid passages away from the center lines of their associated blocks, the alternative is adopted of employing a non-integral number of vanes per seal block (e.g., 10 vanes used with 4 seal blocks to yield 2½ vanes per block), again it will be true that, while the sawtooth forces arising from each pair of diametrically opposite seal blocks will be forces which are synchronous in time, a difference in time phase will exist between the sets of sawtooth forces generated by different seal block pairs because of the asynchronous manner in which the rotor-carried vanes will register with the fluid passages of different seal block pairs.

For either of the described ways in which such difference in time phase is provided by a plurality of pairs of seal blocks, because of the synchronism of the forces individual to the two oppositely disposed seal blocks in each pair thereof the two forces arising out of each such pair will act in both a space balanced and a time balanced manner on the rotor and on the casing. Accordingly, such rotor and such casing will be both spaced balanced and time balanced in respect to the sawtooth forces generated by all the pairs of diametrically opposite seal blocks.

The above described embodiments are being exemplary only, it will be understood that additions thereto, further modifications thereof and omissions therefrom can be made without departing from the spirit of the invention hereof, and that such invention comprehends embodiments differing in form and/or detail from those specifical-

ly disclosed. For example, as previously indicated, the invention hereof in many of its aspects is applicable to a radial machine as well as the axial machine specifically described. Moreover, each of the modifications described herein is usable with any other or others of such modifications which does not have a manually exclusive relation with such one modification. Accordingly, the invention hereof is not to be considered as limited save as is consonant with the recitals of the following claims.

I claim:

1. Apparatus comprising, a member which is a rotor in the relative rotational sense, said rotor having formed in it a two-sided annular groove, a member associated with said rotor and which is a stator in said sense, abutment means received in said groove, said abutment means fitting around at least part of the bottom of said groove and being disposed in said groove to divide it into working and balancing channels on transversely opposite sides of said means, at least one reaction block fixed in the direction around said groove relative to said stator and disposed in said groove to obstruct angular flow of fluid therein, inlet and outlet fluid ports communicating with said working channel on angularly opposite sides of the centerline of said block, a plurality of transversely movable vanes carried by said rotor in angularly spaced slots transverse to and opening into said working channel, and means operable by moving each vane in its slot during relative rotation of said rotor and block to permit such vane to pass said block while at intervening times obstructing angular flow of fluid in said working channel.

2. Apparatus comprising, a member which is a rotor in the relative rotational sense, said rotor having formed in it a two-sided annular groove of which the bottom and both sides are provided by said rotor, a member associated with said rotor and which is a stator in said sense, abutment means which is stationary in said sense and which is in transversely spaced relation from both said sides, said abutment means cooperating with said groove to divide it into transversely opposite, working and balancing annular fluid-receiving channels each bounded by said rotor-provided bottom and by a respective side of said groove and by a respective side wall of said means, and the continuity around said channels of said respectively associated walls being broken by at least one gap extending transversely from one of said channels to the other between angularly spaced portions of said means, at least one reaction block unit disposed in said gap and having a forward fluid-obstructing portion in said working channel and a rear portion in said balancing channel so as to obstruct angular flow of fluid in both channels, inlet and outlet fluid ports communicating with said working channel on angularly opposite sides of the centerline of said block, a plurality of transversely movable vanes carried by said rotor in angularly spaced slots transverse to and opening into said working channel, and means operable by moving each vane in its slot during relative rotation of said rotor and block to permit such vane to pass said block while at intervening times obstructing angular flow of fluid in said working channel.

3. Apparatus as in claim 2 in which said unit has a greater transverse extent in said gap than in said working channel to thereby avoid canting of said unit around a radial axis in the presence of any looseness of fit in said gap of the rear portion of said unit.

4. Apparatus as in claim 2 in which the front expanse in said working channel of said unit is subjected on opposite sides of the centerline thereof to a differential in fluid pressure transversely exerted on said expanse, and in which said apparatus includes means to apply to the rear expanse of said unit a transversely directed fluid pressure likewise characterized by a differential on opposite sides of said centerline and opposing the moment exerted on said unit by said first-named pressure differential.

5. Apparatus as in claim 2 in which said block unit is transversely floating in relation to said abutment means, said apparatus further comprising means to maintain said unit in transversely fixed position relative to said groove.

6. Apparatus as in claim 2 in which the forward end and the rear portion of said block unit are disposed in, respectively, said working channel and balancing channel and occupy relatively narrow and wide angular intervals in the direction around said groove, and in which the angular interval occupied by said rear portion is at least coextensive with the angular interval between the extremities of said inlet and outlet ports which are farthest removed from the centerline of said unit.

7. Apparatus comprising, a member which is a rotor in the relative rotational sense, said rotor having formed in it a two-sided annular groove of which the bottom and both sides are provided by said rotor, a member associated with said rotor and which is a stator in said sense, arcuate abutment means which is stationary in said relative rotational sense and which is transversely adjustable relative to said rotor, said arcuate means fitting around at least part of said bottom of said groove in transversely spaced relation from both said sides of said groove divide it into transversely opposite, working and balancing annular fluid-receiving channels each bounded by said rotor-provided bottom and by a respective side of said groove and by a respective side wall of said means, and the continuity around said channels of said respectively associated walls being broken by at least one gap extending transversely through said means from one to the other of said channels between angularly spaced portions of said means, a reaction block unit disposed in said gap and having a forward fluid-obstructing portion in said working channel and a rear portion in said balancing channel, said unit being in transversely slidable relation with said angularly spaced portions, inlet and outlet fluid ports communicating with said working channel on angularly opposite sides of the centerline of said block, a plurality of transversely movable vanes carried by said rotor in angularly spaced slots transverse to and opening into said working channel, means operable by moving each vane in its slot during relative rotation of said rotor and block to permit such vane to pass said block while at intervening times obstructing angular flow of fluid in said working channel, means to adjust the transverse relative positioning of said abutment means and the combination of said rotor and block unit so as to vary the volumetric capacity of said working channel, and means adapted during such adjustment to maintain said block unit in transversely fixed position relative to said groove.

8. Apparatus comprising, a member which is a rotor in the relative rotational sense, said rotor having formed in it an annular groove of which the bottom and at least one side is provided by said rotor, a casing having a radially projecting annular flange encircling said groove, arcuate abutment means extending said flange radially and fitting around at least part of said groove bottom to define an annular fluid-receiving channel bounded by said rotor-provided side and bottom and by a side wall of said means, at least one reaction block fixed in the direction around said groove relative to said means and disposed in said channel to obstruct angular flow of fluid therein, inlet and outlet fluid ports communicating with said channel on angularly opposite sides of the centerline of said block, a plurality of transversely movable vanes carried by said rotor in angularly spaced slots transverse to and opening into said channel, and means operable by moving each vane in its slot during relative rotation of said rotor and block to permit such vane to pass said block while at intervening times obstructing angular flow of fluid in said channel.

9. Apparatus as in claim 8 in which said arcuate abutment means is set in under said flange away from said rotor-provided groove side to render said channel outwardly bounded at least in part by said flange.

10. Apparatus comprising, a member which is a rotor in the relative rotational sense, said rotor having formed in it an annular groove of which the bottom and at least one side is provided by said rotor, a casing having a radially projecting annular flange encircling said groove, arcuated abutment means extending said flange radially and fitting around at least part of said groove bottom to define an annular fluid-receiving channel bounded by said rotor-provided side and bottom and by a side wall of said means, at least one reaction block fixed in the direction around said groove relative to said means and disposed in said channel to obstruct angular flow of fluid therein, inlet and outlet fluid ports communicating with said channel on angularly opposite sides of the centerline of said block, a plurality of transversely movable vanes carried by said rotor in angularly spaced slots transverse to and opening into said channel, means operable by moving each vane in its slot during relative rotation of said rotor and block to permit such vane to pass said block while at intervening times obstructing angular flow of fluid in said channel, and a sleeve disposed around said rotor and having an annular front face registering with the side of said flange to be in fluid sealing relation therewith.

11. Apparatus comprising, a member which is a rotor in the relative rotational sense, said rotor having formed in it an annular groove of which the bottom and at least one side is provided by said rotor, a casing having a radially projecting annular flange encircling said groove, said rotor and casing being relatively adjustable in the transverse direction, arcuate abutment means extending said flange radially and fitting around at least part of said groove bottom to define an annular fluid-receiving channel bounded by said rotor-provided side and bottom and by a side wall of said means, at least one reaction block fixed in the direction around said groove relative to said means and disposed in said channel to obstruct annular flow of fluid therein, inlet and outlet fluid ports communicating with said channel on angularly opposite sides of the centerline of said block, a plurality of transversely movable vanes carried by said rotor in angularly spaced slots transverse to and opening into said channel, means operable by moving each vane in its slot during relative rotation of said rotor and block to permit such vane to pass said block while at intervening times obstructing angular flow of fluid in said channel, a sleeve disposed around said rotor and vanes and having an annular front face registering with the side of said flange to be in fluid sealing relation therewith, means to adjust transversely the relative positioning of the combination of said rotor and sleeve and the combination of said casing, flange and abutment means to thereby vary the volumetric capacity of said channel, and means adapted during said adjustment to maintain said annular face of said sleeve in registering fluid sealing relation with the side of said flange.

12. A structure adapted for use in vane impulsion apparatus and comprising, a sleeve having a front annular sealing face and a rear face and having formed therein a plurality of fluid passages extending from the front end of said sleeve to the rear face thereof, said passages being enlarged at the rear end of said sleeve, a bushing seated in slidable relation in each such enlarged portion, and means to urge each such bushing rearwardly of the enlarged portion in which the bushing is seated.

13. A structure as in claim 12 in which the front end of said sleeve contains a tubular liner of harder material than the surrounding sleeve body and providing at least a portion of said annular sealing face.

14. A structure as in claim 12 in which said passages communicate with the interior of said sleeve at the front end thereof.

15. A structure adapted for use in vane impulsion apparatus and comprising, a sleeve having a front annular sealing face and a rear face, said sleeve having formed therein a plurality of fluid passages each extending from the front end of said sleeve to an enlarged counterbore formed in the rear face of said sleeve to extend axially inward thereof, a bushing seated in axially slidable relation in each counterbore and having a rearwardly disposed enlarged head in which is formed a rearwardly facing pocket in fluid communication with the front of said bushing through a fluid passage formed in said bushing, and means to urge the head of each bushing to project rearwardly of the counterbore in which the bushing is seated.

16. A structure adapted for use in vane impulsion apparatus and comprising, a sleeve having a front annular sealing face and a rear face, said sleeve having a plurality of angularly spaced pockets formed in radially spaced relation from a transversely extending surface of said sleeve, and said sleeve providing a plurality of angularly spaced fluid passages of which each extends radially in said sleeve from said surface thereof to one of said pockets.

17. A structure as in claim 16 in which said passages are provided by holes passing through said sleeve and having axially distributed locations therein.

18. Apparatus comprising, a member which is a rotor in the relative rotational sense, said rotor having formed in it an annular two-sided groove, a member associated with said rotor and which is a stator in said sense, abutment means disposed in said groove to divide it into working and balancing annular channels on transversely opposite sides of said means, at least one reaction block unit fixed in the direction around said groove relative to said stator disposed in said working channel to obstruct angular flow of fluid therein, inlet and outlet fluid ports communicating with said working channel on angularly opposite sides of the centerline of said block unit, a plurality of transversely movable vanes carried by said rotor in angularly spaced slots transverse to and opening into said working channel, means operable by moving each vane in its slot during relative rotation of said rotor and block unit to permit such vane to pass said block unit while at intervening times obstructing angular flow of fluid in said working channel, means providing communication of pressurized fluid between said working and balancing channels, and means in said balancing channel to obstruct the passage therethrough of fluid from said inlet port to said outlet port.

19. Apparatus as in claim 18 in which said means providing communication of pressurized fluid between said working and balancing channels comprises at least one fluid passage extending axially in said abutment means from said working channel to said balancing channel.

20. Apparatus as in claim 18 in which said means providing communication of pressurized fluid between said working and balancing channels comprises at least one fluid passage extending transversely in said block unit from said working channel to said balancing channel.

21. Apparatus as in claim 18 in which said reaction block unit extends from side to side of said two-sided groove by passing transversely through a gap formed in said abutment means, and in which said obstructing means in said balancing channel is comprised at least in part of said block unit.

22. Apparatus as in claim 18 in which said rotor slots and the vanes contained thereby extend transversely on both sides of said groove, each vane has formed therein a notch within which said block unit fits when such vane is angularly positioned at the centerline of said block unit and is passing said unit, and each vane obstructs angular flow of fluid in said working channel.

23. Apparatus comprising, a member which is a rotor in the relative rotational sense, said rotor having formed in it and around its axis an annular two-sided groove, a member associated with said rotor and which is a stator in said sense, abutment means which is stationary in said relative rotational sense but is axially adjustable relative to said rotor, said means being disposed in said groove axially intermediate the two sides thereof to divide it into working and balancing annular channels on axially op-

posite sides of said means, at least one reaction block unit fixed in the direction around said axis relative to said means and disposed in said working channel to obstruct angular flow of fluid therein, inlet and outlet fluid ports communicating with said working channel on angularly opposite sides of the centerline of said block unit, a plurality of transversely movable vanes carried by said rotor in angularly spaced slots transverse to and opening into said working channel, means operable by moving each vane in its slot during relative rotation of said rotor and block unit to permit such vane to pass said block unit while at intervening times obstructing angular flow of fluid in said working channel, means to adjust the relative axial position of said rotor and abutment means so as to vary the volumetric capacity of said working channel, means providing communication of pressurized fluid between said working and balancing channels, and means in said balancing channel to obstruct the passage therethrough of fluid from said inlet port to said outlet port.

24. Apparatus comprising, a rotor having formed in it and around its axis an annular two-sided groove, a casing around said rotor on both sides of said groove and axially adjustable relative to said rotor, said casing having an instanding annular flange encircling said groove, arcuate abutment means extending said flange inwardly and fitting around at least part of the bottom of said groove, said means being disposed in said groove axially intermediate the two sides thereof to divide it into working and balancing annular channels on axially opposite sides of said means, at least one reaction block unit fixed in the direction around said axis relative to said means and disposed in said working channel to obstruct angular flow of fluid therein, inlet and outlet fluid ports communicating with said working channel on angularly opposite sides of the centerline of said block unit, a plurality of transversely movable vanes carried by said rotor in angularly spaced slots transverse to and opening into said working channel, means operable by moving each vane in its slot during relative rotation of said rotor and block unit to permit such vane to pass said block unit while at intervening times obstructing angular flow of fluid in said working channel, means to axially adjust the relative positioning of said rotor and of the combination of said casing, flange and abutment means so as to vary the volumetric capacity of said working channel, a pair of sleeves disposed inside said casing on opposite sides of said flange to each surround part of said rotor, said sleeves each having an annular front face registering in fluid sealing relation with the side near thereto of said flange, means coupling each sleeve to said rotor for rotation together while permitting relative axial movement between such sleeve and said rotor, means to maintain each sleeve registering in fluid sealing relation with said flange during said axial adjustment, means providing communication of pressurized fluid between said working and balancing channels, and means in said balancing channel to obstruct the passage therethrough of fluid from said inlet port to said outlet port.

25. Apparatus comprising, a member which is a rotor in the relative rotational sense, said rotor having formed in it an annular two-sided groove, a casing having an radially projecting annular flange encircling said groove, a plurality of angularly spaced seal blocks extending said flange radially and each fitting around a portion of said groove bottom, said seal blocks being disposed in said groove to divide it into working and balancing annular channels on transversely opposite sides of said seal blocks, a corresponding plurality of angularly spaced reaction block units of which each extends transversely through a gap between two adjacent seal blocks and from side to side of said groove to obstruct angular flow of fluid in both said working and balancing channels, inlet and outlet fluid ports communicating with said working channel on angularly opposite sides of the centerline of each unit, a plurality of transversely movable vanes carried by said rotor in angularly spaced slots transverse to and opening into said working channel, cam means operable by moving each vane in its slot during relative rotation between said rotor and units to permit each such vane to pass each such unit while at intervening times obstructing angular flow of fluid in said working channel, and means to transmit pressurized fluid between said working channel and said balancing channel both in the angular intervals occupied by said seal blocks and in the angular intervals occupied by said reaction block units.

26. Apparatus as in claim **25** in which the combination of said casings, flange, seal blocks, vanes and cam means is axially adjustable relative to said rotor and reaction block units so as to vary the volumetric capacity of said working channel.

27. Apparatus as in claim **25** in which there are a fractional number of vanes per seal block.

28. Apparatus as in claim **25** in which said pressure transmitting means is comprised for at least one angular interval occupied by a seal block of two fluid passages extending axially in such block and angularly spaced by about the thickness of one of said vanes.

29. Apparatus as in claim **25** in which said pressure transmitting means is comprised of a plurality of fluid passages of which at least one axially extends through each of said seal blocks, and which are distributed in their angular locations relative to the centerlines of the seal blocks through which they extend.

30. Apparatus comprising, a pair of members which are, respectively, a rotor and a stator in a relative rotational sense, said rotor having formed in it an annular fluid-receiving channel, said stator having an annular radially projecting flange surrounding said channel at least one angularly stationary reaction block unit disposed in said channel to obstruct angular flow of fluid therein, inlet and outlet fluid ports communicating with said channel on angularly opposite sides of the centerline of said block unit, a plurality of transversely movable vanes carried by said rotor in angularly spaced slots transverse to and opening into said channel, means operable by moving each vane in its slot during relative rotation of said rotor and block unit to permit such vane to pass said unit while at intervening times obstructing angular flow of fluid in said channel, at least one sleeve disposed around said rotor between said rotor and stator, said sleeve having an annular front face registering with a side of said flange and having a closer fit therewith than the fit between said sleeve and said stator.

31. Apparatus comprising, an axially stationary rotor formed to provide at least part of an annular fluid-receiving channel, an axially adjustable casing around said rotor, at least one reaction block unit fixed in the direction around said axis relative to said casing and disposed in said channel to obstruct angular flow of fluid therein, a plurality of transversely movable vanes carried by said rotor in angularly spaced slots transverse to and opening into said channel, cam means operable by moving each vane in its slot during relative rotation of said rotor and block to permit such vane to pass said block while at intervening times obstructing angular flow of fluid in said channel, an axially stationary housing surrounding said casing and of an interior shape cooperating with that of the exterior of said casing to form fluid inlet and outlet chambers between said housing and casing, inlet and outlet fluid ports respectively passing from the consonantly named chambers through said casing to openings into said channel on opposite sides of the centerline of said block unit, means to adjust the axial position of said casing, means axially fixed relative to said casing to be axially adjusted therewith and to thereby vary the volumetric capacity of said channel, and seal means disposed between said housing and casing to render said chambers fluid-tight while permitting relative axial movement between said casing and housing during said axial adjustment of said casing.

32. Apparatus comprising, a member which is a rotor in the relative rotational sense, said rotor being formed to provide at least part of an annular fluid-receiving channel, a member associated with said rotor and which is a stator in said sense, at least one reaction block unit which is stationary in said sense and which is disposed in said channel to obstruct angular flow of fluid therein, inlet and outlet fluid ports communicating with said channel on angularly opposite sides of the centerline of said block unit, a plurality of transversely movable vanes carried by said rotor in angularly spaced slots transverse to and opening into said channel, means operable by moving each vane in its slot during relative rotation of said rotor and block to permit such vane to pass said block while at intervening times obstructing angular flow of fluid in said channel, and hydraulic damping means coupled with said rotor to absorb transient variations in

axially directed force exerted on said rotor by pressurized fluid flowing between said ports.

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