March 16, 1965 K. EICKMANN 3,173,375

ROTARY FLUID MACHINES

Original Filed Feb. 9, 1959 3 Sheets-Sheet 1

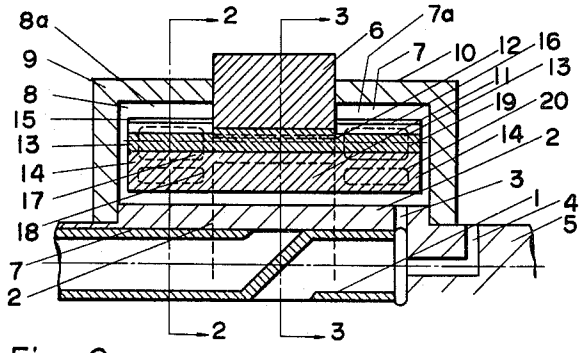



Fig. 1

Fig. 2

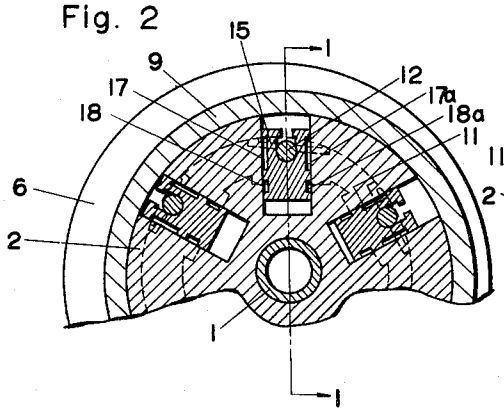



Fig. 3

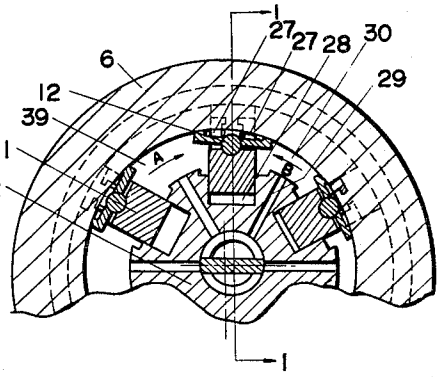



Fig. 4

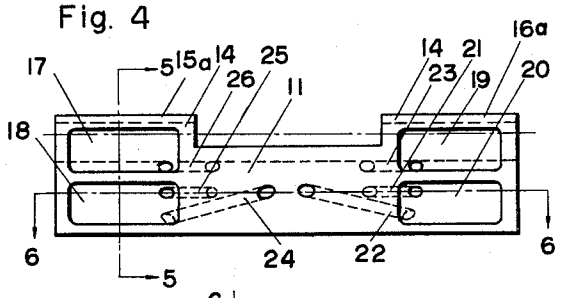



Fig. 5

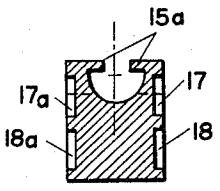



Fig. 6

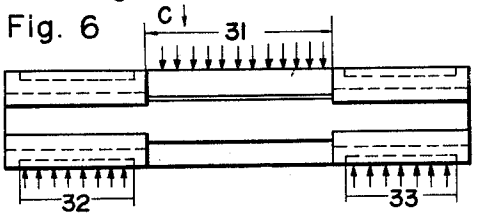
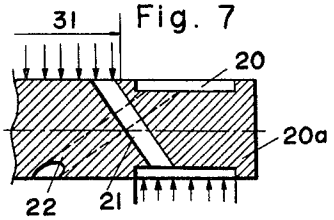



Fig. 7




INVENTOR:
KARL EICKMANN

By Burgess, Dinklage & Sprung
ATTORNEYS

March 16, 1965 K. EICKMANN 3,173,375

ROTARY FLUID MACHINES

Original Filed Feb. 9, 1959 3 Sheets-Sheet 2

Fig. 8

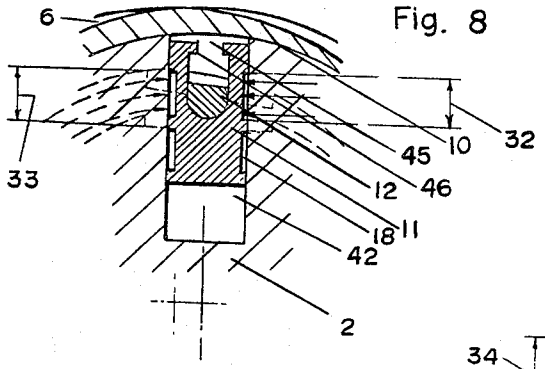



Fig. 9

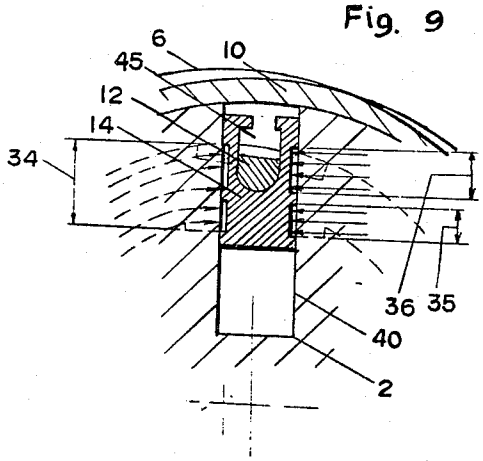
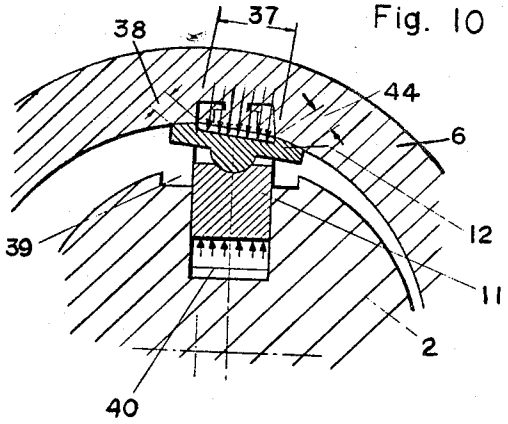



Fig. 10




INVENTOR.
KARL EICKMANN
BY
Burgess, Dinklage & Sprung
ATTORNEYS

March 16, 1965 K. EICKMANN 3,173,375

ROTARY FLUID MACHINES

Original Filed Feb. 9, 1959 3 Sheets-Sheet 3

Fig. 12

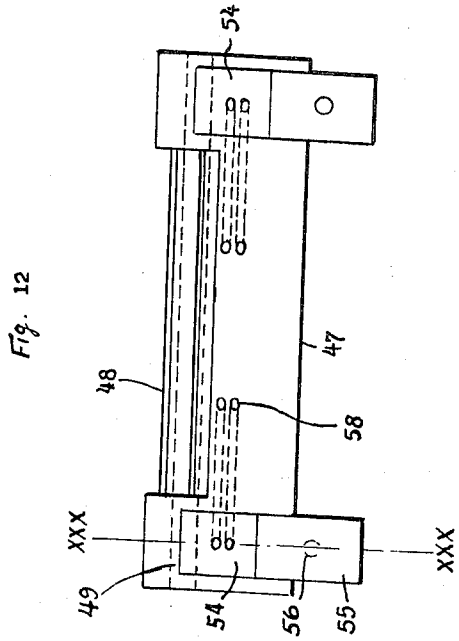
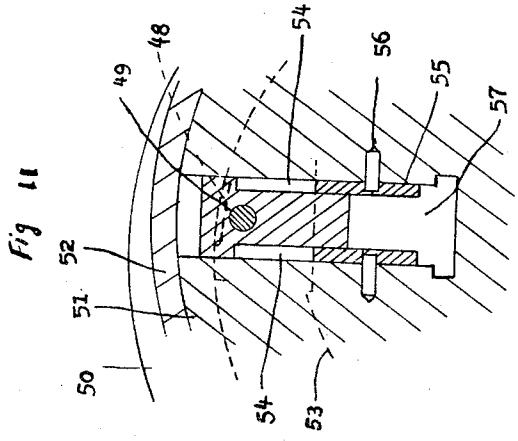



Fig 11




INVENTOR
KARL EICKMANN

United States Patent Office

3,173,375

Patented Mar. 16, 1965



3,173,375

# ROTARY FLUID MACHINES

Karl Eickmann, 2420 Isshiki, Hayama-machi, Kanagawa-ken, Japan

Original application Feb. 9, 1959, Ser. No. 792,001, now Patent No. 3,099,964, dated Aug. 6, 1963. Divided and this application Mar. 2, 1962, Ser. No. 177,088

10 Claims. (Cl. 103—136)

This invention relates to rotary machines of the vane type and more particularly to those in which the vanes are provided with axially extended slide portions in the side walls of the rotor for stabilizing the movement of the vanes and eliminating the tilting of the vanes in their respective guide slots during operation. These vanes are further provided with recesses into which fluids (liquids or gases) under pressure may be directed so as to produce counteracting forces to those exerted by the pressure fluid tangentially or almost tangentially to the rotor and normal to the longitudinal vane wall and probably to those exerted radially against the bottom vane wall. In this way, the vanes may be stably supported in their respective slots and the tangential force causing undesired friction may be completely or at least partially counterbalanced.

This application is a division of my copending application Serial No. 792,001 filed February 9, 1959, now Patent No. 3,099,964.

Heretofore, various vane constructions for rotary piston machines have been used, although in such constructions there has been undesirable tilting of the vane due to the tangential force exerted thereagainst. Such tilting causes friction and wear along the lower trailing edges of the vane and slot and thereby limits the operating pressure. Previously, little importance was attached to this tilting and the resultant friction and wear, since rotary piston machines, especially rotary vane machines operating at low efficiency or at low or middle pressure were considered adequate.

In normal rotary vane machine operation, the pressure fluid acts on the vane in many directions each of them normal to the surface or to a surface of the vane assembly or of the vane, but the resultant of the said pressure-acting is in one direction, when the cell acts as a compression cell and in the opposite direction, when the cell acts as a suction cell. The vane travels more or less radially outwardly within the slot provided in the rotor during rotation of the machine about one half of the axis of rotation and then more or less radially inwardly within the slot during rotation about the other half of the cycle. During such operation in conventional rotary vane machines, the resultant of forces in liquids or gases under pressure in the working cells acts on the vane tangentially or almost tangentially to the rotor, causing pressure against the vane during its travel within the slot. The friction produced by this force against the vane is the more considerable, the higher the acting pressure becomes, and consequently reduces the efficiency of the machine. The elimination of tilting and friction of the vane and of the slot walls has become increasingly important where the machine is required to operate at high pressure and where the machine is required to operate at high efficiency and with minimum frictional losses as it is the case with conventional unadjustable gear transmissions or gear transmission machines.

An improved rotary vane machine, capable of high pressure up to ten times higher than in vane machines without rotor side wall supported vanes, is set forth in British Patent No. 744,446 (of Feb. 8, 1956) and U.S. Patent No. 2,975,716. In that case a rotary machine or pump and motor is provided with vanes which slide in slots located in the rotor and in the rotor side walls and which are enclosed by a casing ring, thereby defining compression cells capable of high pressure. Each cell during rotation of the machine as a motor admits a fluid under pressure as the cell increases in volume and releases the fluid as the cell decreases in volume, while in acting as a pump the cell draws in the fluid by suction as the cell increases in volume and expels the fluid as the cell decreases in volume. The tilting of the vanes is eliminated in this rotary vane machine by the fact that the vanes are axially extended into and borne by slots in side walls of the rotor. The side walls of the rotor are provided on both axial ends of the rotor and are rotatable with the rotor. The walls of the slots in the side discs of the rotor support the vanes during the rotor's rotation and during the vane's travel. This improvement resulted also in the said capacity of high pressure. Since this improved rotary machine is capable of high pressure and therefore able to be used in cases of large power, the elimination of slide friction is important in order to increase the efficiency of the machine.



An object of the present invention is to improve the efficiency, the lifetime and the capacity of pressure and power of rotary vane machinery and to provide a vane in rotary vane machines having means for stabilizing the vane during rotation and for counterbalancing the force of the fluid under pressure in working cells acting on the vane in tangential or almost tangential direction, whereby to permit free floating travel of the vane in the guide slot and reduce friction and wear of the vane passing along adjacent parts.

In order to reach such improvements in rotary vane machines provided with vanes, which are extended axially into and borne by slots also in the rotor's end wall means, recesses are provided in the bearing surfaces of the vanes both axial extensions into which the pressure fluid may be transmitted by means of bore passages or channels. These vane recesses are arranged in such a manner that the force or pressure fields produced by the liquid or gas under pressure therein may be utilized to counterbalance the tangential force acting against the opposite side of the vane. Upon reaching this state of counterbalance, the vane substantially floats in the guide slot radially or almost radially inwards and outwards between the forces of liquids or gases under pressure. Said recesses are provided in order to receive or to contain fluids under pressure in the bearing surface of the vane—regardless whether simple plate-formed vanes with plane surfaces are used or highly specialized vane-slider-assemblies.

Thus, in a construction according to this invention, slide under load by contact of metal between the vane and guide slot is almost completely avoided. The tangential or almost tangential force acting on the vane is transmitted to the wall of the slot of the rotor's side disc by means of the pressure fields as per above construction. Consequently, instead of metal acting on metal, the pressure fluid acts on metal and the vane is free to float in the guide slot with a minimum of friction. Some friction, mostly caused by resistance against motion, is still present in the fluid adjacent to the vane, but this is comparatively negligible.

Besides, it has been found, in accordance with the present invention, that frictional forces of the vane as well as centrifugal forces and forces acting on the bottom side of the vane may be more or less completely balanced by corresponding counteracting forces. Thus, stabilized support of the vane during operation will reduce friction to a minimum and these counterbalancing forces will prevent any overloading that would otherwise cause undue wear.

Thus, means to reduce the tangential resulting forces acting on the vane embodiment as well as the radial forces acting on the vane embodiment can be combined according to this invention.

Specifically, in accordance with a preferred embodiment a vane assembly can be divided into a vane and a slide element. The slide element is located in an elongated recess of partial cylindrical configuration provided along the top radial edge of the vane and can pivot to a limited extend around its longitudinal axis in said recess. The top surface of the slide element is shaped to slide smoothly along the inside surface of the case during rotation. Recesses can be provided in the top surface of the slide element into which the pressure fluid of the machine may be passed, so as to act against both the slide element and the case and prevent friction. In acting against the slide element and casing ring a force is provided which completely or at least partially counterbalances the outward radial forces exerted against the bottom of the vane in the slot and in turn transmitted radially outwardly to the slide element. In order to accomplish this counterbalancing by means of the so-called pressure fields created in the slider recesses and also to stabilize vane support, the slide element is preferably constructed of greater width than the vane. Accordingly a larger area of contact with the case surface is provided which improves the supporting stability and the sealing of the slide element against leakage of liquid or gas.

The invention applies to small vane machines of only small horsepower e.g. combustion engines for cars and the like and with special effect to those with some 10,000 (ten thousand) horsepower e.g. propeller drives for ships. Said pressure fields are most suitable for vane machiner in which every mechanical part is floating, i.e. which have floating control means and floating casing rings. In case of hydraulic fluids in working cells under high pressure of some thousand p.s.i. and unit's sizes of some thousand horsepower total efficiency of the vane machine up to almost 97% can be obtained.

The invention is illustrated by the accompanying drawings in which:

FIG. 1 is a partial sectional view of a rotary vane machine taken along the line III—III of FIG. 2 and along the line IV—IV of FIG. 3.

FIG. 2 is a partial sectional view taken along the line I—I of FIG. 1 showing the vane recesses.

FIG. 3 is a partial sectional view taken along the line II—II of FIG. 1 showing the slide element.

FIG. 4 is side view showing an embodiment of the vanes of rotary vane machine according to the invention.

FIG. 5 is a sectional view taken along the line V—V of FIG. 4.

FIG. 6 is a top plan view of the vane in FIG. 4, the view illustrating the forces acting on the vanes in tangential direction to the rotor, and the balancing pressure fields counteracting said forces.

FIG. 7 shows a partial section taken along the line VI—VI of FIG. 4.

FIG. 8 is an enlarged view of a partial section taken along the line I—I of FIG. 1, showing the end portion of the vane position in the slot defined by the side walls of the rotor, the view illustrating the acting force in tangential direction and the balancing pressure fields in counter direction where the vane has moved slightly outwardly in a radial direction within the slot.

FIG. 9 is a sectional view similar to FIG. 8 showing the vane further outwardly positioned in a radial direction within the slot of the rotor and indicating the consequent existence of greater forces.

FIG. 10 is an enlarged view of a partial section taken along line II—II of FIG. 1 showing the force of the vane and of the slide element produced in a radial direction and the balancing of these forces by means of corresponding counter pressure fields.

FIG. 11 is a slide view of another embodiment of the invention showing adjustable vane recesses for correspondingly adjusting the balancing pressure fields.

FIG. 12 is an enlargement of a cross-sectional view taken along the line XXX—XXX of FIG. 11 illustrating the manner of adjustment of the vane recesses.



Referring to the drawings, in FIGS. 1, 2 and 3, 1 represents a control shaft of a rotary piston machine, 2 is the rotor, and 3 represents bores guiding radially outwardly and inwardly the pressure medium to and from the slot space below the vane through the control shaft. 4 is a channel guiding outwardly the pressure medium from the rotor during each revolution of the machine. 5 is a shaft driving the rotor. 6 is a casing ring eccentric to the rotor. 7 and 8 are extended portions of the vane slot defined by the side walls of the rotor. 7*a* and 8*a* are pressure medium chambers located radially in the vane slot above the vane. 9 and 10 are side-covers sealing axially and radially the side walls of the rotor. 11 are vanes of the rotary piston machine and 12 is a slide element thereon. 13 is an extended rotating member of the slide element adjacent to the side wall. 14 is an extended portion of the vane located along the side wall and rotating therewith. 15 and 16 are U-shaped portions located on the extended portions of the vane with projections to prevent the slide positioned therewithin from falling out of the vane. 17 and 18 are balancing pressure recesses provided in the vane. 17*a* and 18*a* are corresponding balancing pressure recesses provided separately in the vane. 27 is a balancing pressure recess in the slide. 28 is a pressure medium chamber located radially and inwardly of the vane. 29 is a passage in the rotor for guiding the pressure medium into and out of the vane cells. 30 and 39 are two adjacent vane cells. These cells are the piston chambers of the rotary piston machine.

In FIGS. 4, 5, 6 and 7, 11 is the vane, 14 is the extended side portion of the vane within the extended side wall of the rotor. 15*a* are inner bent projections along the top edge of the vane preventing the slide from falling out of its vane seat. 17, 17*a* 18, and 18*a* are balancing pressure recesses in the vane. 19 and 20 are additional balancing pressure recesses in the vane, opposite corresponding balancing pressure recesses 19*a* and 20*a*, shown in FIG. 2. 23 is a bore channel guiding the pressure medium into the balancing pressure recess 19. 21, 22, 24, 25 and 26 are bore channels for correspondingly guiding the pressure medium to balancing pressure recesses 20, 20*a*, 18 and 17. 31 shows the working medium under pressure acting on the vane in the direction of the arrow C. 32 and 33 show the balancing pressure fields in the vane recesses on the opposite side of the vane and the working direction of the counteracting pressure in the balancing pressure fields.

In FIGS. 8, 9 and 10, 2 is the rotor. 11 is the vane, 12 is the slide element, 6 is the casing, and 10 is the side wall sealing cover. 40 and 42 are radial pressure chambers inwardly to the vane, and 45 is a radial pressure chamber outwardly to the vane while 46 is a buckling of the vane. 39 is a vane cell. 33 and 34 show the working medium under pressure acting on the vanes in different radial positions. 32, 35 and 36 show the pressure medium in the balancing pressure recesses counteracting the working pressure. 44 and 38 are widened slide ends providing a stabilized support for the slide. 40 shows the pressure medium acting radially and outwardly on the vane. 37 shows the balancing pressure field created in the slide element recesses which counteracts the pressure in 40.

In FIGS. 11 and 12, 47 is the vane, 48 is the sliding member of the slide element, and 49 is the rotating member, 50 is the enclosing ring, 51 is the side wall of the rotor, and 52 is the sealing cover of the side wall of the rotor. 53 is the rotor. 54 is the balancing pressure recess provided on the side face of the vane and 55 is the sealing plate sealing from below the balancing pressure recesses. 56 is the pin fixing the sealing plate on the side face of the guide slot 57 and rotor side wall. 58 is the conduit connecting the balancing pressure recess to the vane cell on the opposite side of the vane.

The invention is more fully described by the following:

In FIGS. 1, 2 and 3, the disposition of the vanes and slide elements in the rotary piston machine and the position of the balancing pressure recesses are shown. Vanes **11** slide in radial guide slots **8** and **7** provided in rotor **2** and revolve together with the rotor. The rotary piston machine is enclosed by casing ring **6** and upon rotation vanes **11** are thrown radially outwardly by centrifugal and other forces, so that slide element **12**, positioned on the top portion of the vane abuts the inside surface of casing ring **6** and slides along the casing ring. In this way variable volume working cells are formed between casing ring **6**, vanes **11** and slide elements **12**, and side covers **9**, **10** sealing axially and radially the rotor and side walls. During each revolution of the rotor, vanes **11** and side elements **12** travel radially inwardly and outwardly, so that the capacity of working cells is increased and decreased accordingly. Two adjacent working cells are shown in positions **30** and **39** in FIG. 3. These cells take up the working medium by means of distribution shaft **1** through the passages **29** during one half of the cycle and force the medium out through another passage of the distribution shaft **1** during the second half of the cycle. The rotor is driven by driving shaft **5**, and the pressure medium leaves the rotor through bores **4**. Passing through channels **3**, the pressure medium enters slot chamber **28** radially below the vane through the distribution shaft. In these slot chambers, the pressure medium acts on the vane bottom forcing the vane radially outwardly. The vanes are shorter than the length of the slots located in the side walls or are provided with radial bores or slots, so that the pressure medium may flow radially and outwardly along the lateral ends of the vanes, thereby reacting chambers **7***a* or **8***a*, located radially within the slots and above the vanes. An important feature of the design concerns the passage of the pressure medium as from chambers **7***a*, **8***a* into the balacing pressure recesses **27** provided in slide elements **12**. In these balancing pressure recesses of slide element **12**, a pressure field is produced and the pressure medium is directed radially inwardly, toward the vanes to counteract the radial outward pressure in slot chamber **28**, and the centrifugal force acting on vane **11**. This counteracting field completely balances the pressure force on the vane bottom from chamber **28** as well as the acting pressure force and centrifugal force, so that the vane in radial direction is completely free from or is only slightly affected by the fluid pressure force, acting pressure force, and centrifugal force. Even when the pressure of the medium is very high, the vanes are pressed radially and inwardly by said sliding element pressure field, so that excessive compression between the vane and the casing ring or between the slide and the casing ring does not occur. Any overloading is therefore automatically prevented.

Where the rotary piston machine serves as a motor with fluid pressure medium, revolving in the direction of arrow B, as shown in FIG. 3, vane **11** is caused by the working pressure in the cell **30**, to rotate the rotor in the direction of arrow B. The pressure medium passing through bores **23**, **21** and **25**, **26** in the vane, enters the balancing pressure recesses **20**, **19** and **18**, **17** from the cell **30**. Consequently, the balancing pressure recesses **17**, **18** and **19**, **20** positioned in the extended side portion of the vane within the slot of the rotor are affected by similar pressure on the other side of the vane ahead of cell **30**, and the pressure in these balancing fields is directed oppositely to the pressure in the direction of arrow B in cell **30**, thus balancing completely or partially the working pressure in the direction of arrow B. In this way, pressure load of vanes **11** within the slots is reduced completely or partially, and the vane travels freely in the pressure medium between the walls of the slot. While the balancing pressure recesses have been divided into **17** and **18**, many balancing pressure recesses may be used



so as to increase or decrease in any desired manner the pressure of the balanced pressure field in accordance with the degree of radial travel of the vane.

Where the rotary piston machine serves as a pressure pump rotating in the direction of arrow B, the medium pressure is produced in cell **39**, exerting a tangential pressure on vane **11** in the direction of arrow A. This pressure is balanced by the balancing pressure fields created in recesses **17***a*, **18***a*, **19***a* and **20***a*. The pressure medium flows to the balancing pressure recesses **17***a* and **18***a* through bores **24** and **24***a* and to the balancing pressure recesses **19***a* and **20***a* through bores **22** and **22***a*. This action of force is applicable similarly to all vanes **11** and slide elements **12** in corresponding positions of the rotor.

By extending considerably the vane ends **14** into slot chambers **7**, **8** a space is produced which is sufficiently large for the balancing pressure fields created within the vane to balance the tangential pressure acting on the vane to any desired extent. By bending over edges **15***a*, **16***a* of the vane, slide elements **12** cannot become dislodged, as rotating member **13** is inserted below the bent portions of the vane.

FIGS. 4, 5, 6 and 7 show the recesses creating the balancing pressure fields counteracting the tangential force against the vanes; the disposition of bores guiding the pressure medium into the balancing pressure fields; the working pressure acting on the vanes; and the pressure in the balancing pressure recesses acting upon the vanes. Vane **11** has extended portions **14** with which it is fitted in the side walls of the rotor. An essential feature of the invention comprises providing in extended portions **14** recesses such as **17**, **18**, **19**, **20**, **17***a*, **18***a*, **19***a* and **20***a* or many further recesses suitably disposed for receiving the pressure medium therein. The pressure medium acting upon the opposite face of vane **11** is guided through corresponding bores to these recesses. **31** indicates the working pressure acting on the vane in the direction of arrow C. This pressure is transmitted to the balancing pressure recess **20***a* through bore **21** as shown in FIG. 7, and likewise to the balancing pressure recess **18***a* through bore **25**. Similarly, balancing pressure recesses **19***a* and **17***a* are affected by the pressure medium through the bores **23** and **26**. **31** in FIG. 6 shows the working pressure acting on the vane in the direction of arrow C. **32** and **33** show the pressure acting upon the vane in the balancing pressure fields. As is apparent in FIGS. 6 and 7 in the aforesaid manner, the force of the working pressure acting on the vane is completely balanced by the sum of the forces in opposite direction produced in the individual balancing pressure fields via the recesses.

Where the rotary piston machine is operated in the reverse direction, the pressure at **31** in FIG. 6 acts on the opposite side of the vane and in opposite direction to that of arrow C. Thus, the pressure is not guided to the balancing pressure recesses **17***a*, **18***a*, **19***a* and **20***a*, but instead to the balancing pressure recesses **17**, **18**, **19** and **20** through corresponding bores, and this pressure acts on the side opposite faces **32** and **33** and in the opposite direction.

The larger the outward radial movement of the vane in the slot, the more will be the working face of the medium. In FIG. 8, the working medium is effected only in plane **33**, since the vane moved slightly radially outwardly within the slot. In this case, as shown in FIG. 4, bores **21** and **25** are still covered by the slot of the rotor, and only recesses **17***a* and **19***a* create the balancing pressure fields shown at **32**. In FIG. 9 the vane has traveled further radially outwardly, and bores **21** and **25** in FIG. 4 are positioned externally to the slot of the rotor and abut cell **39** of FIG. 3. In this position, as shown at **34** in FIG. 9, the tangential working pressure acts on vane **11** while the balancing pressure fields created act in the direction indicated by the pressure arrows **35** and **36**.

Although the embodiment shown employs two separate balancing pressure fields at each vane extension, many

steps of fields may be used so that the balancing pressure fields and their action increase continuously as the vane travels radially outwardly of the slot. FIG. 10 shows a manner of balancing the radial fluid pressure acting upon the vane bottom. The pressure medium is guided into inner cell 40 beneath the vane and within the slot of the rotor. The pressure medium is also present in cell 45 positioned radially outwardly of the vane within the side walls as shown in FIGS. 8 and 9. Cell 45 is connected directly with the balancing pressure recesses provided in the slide element. If the area of pressure fields 40 and 37 are the same, the force produced at area 40 will be equal to the force produced at area 37. In this way, the forces in the pressure medium acting radially outwardly upon the vane bottom and inwardly upon the slide element are in balance and the vane, in arrangement with the slide element, freely floats between casing ring 6 and rotor 2. Where the field area 37 is greater than area 40, any friction caused by the centrifugal force is offset, under a given pressure in the pump. 44 and 38 indicate extended surface portions of the sliding member. These portions provide a stable support and a stable sliding of the slide element along casing ring 6 and further, these portions perform a sealing action between balancing pressure field 37 in FIG. 10 and the pressure in the working cells of the individual vanes. Rotor 2 has a groove 39 shown in FIG. 10 to fit therein sliding parts 44, 38, where the vane travels radially inwardly.

In the embodiment shown in FIGS. 11 and 12, balancing pressure recess 54 provided on the extended portion of vane 47 is open on its bottom side and may be enlarged downwardly toward the lower end of the vane upon radial outward travel of the vane within slot 57. In order to seal the open end of the recess in a downward direction, sealing plate 55, which is adapted to slidably and tightly close recess 54 during the radial inward travel of vane 47, is provided on the side face of the slot 57 of rotor side wall 51 in direct line with recess 54. Plate 55 may be suitably secured, as for example, with a pin 56. Thus, during operation of the rotary piston machine, all sides of the pressure recesses creating the counter balancing pressure fields are tightly sealed, allowing entry and exit of the pressure medium only through ducts 58 to similar recesses on the opposite side of the vane. The counter pressure fields created, balance the vane in the slot and permit the vane to float radially inwardly and outwardly during each machine cycle with a minimum of friction, tilting and wear.

In the last mentioned embodiment, during rotation of the rotor, the balancing pressure field created increases and decreases accordingly as the vane moves in and out of the slot, affording a more complete balancing action than the step-like action which takes place where separate recesses are provided in the extended portion of the vane.

It will be possible according to the present invention to connect vane 47 and the sliding element 48 in such a way that they swing towards each other and furthermore to provide vanes with plane or even axial extensions without sliding element.

What is claimed is:

1. Rotary fluid engine comprising a casing, a rotor mounted within said casing and having side walls extending axially and radially outwardly beyond said casing to form end closures for said casing, said casing on the one hand and said rotor and side walls on the other hand being arranged for relative rotation with respect to each other, said rotor having radial slots, vanes in said slots having extensions in said side walls provided with bearing surfaces in said slots at said side walls, one of said bearing surfaces of a vane extension in a corresponding side wall having a counterbalancing recess to receive pressure fluid, and a passage leading from said recess to the face of the corresponding vane within said casing opposite such recess, such recess being disposed radially in said vane extension such that the center of pressure of said recess is substantially within the radial limits of the center portion of the vane exposed to fluid pressure.

2. Engine according to claim 1 wherein at least one recess is provided in each vane extension at a corresponding bearing surface and a corresponding passage is provided leading from each said recess to the face of the corresponding vane within said casing opposite such recess.

3. Engine according to claim 2 wherein at least one of said bearing surfaces is provided with more than one separate recess, positioned radially one above the other, the corresponding vane having a passage leading from each such recess to the face of the vane within said casing opposite such recess.

4. Engine according to claim 3 wherein the corresponding passage for each such recess leads from the opposite vane face at a radial level within the range of the uppermost and lowermost edges of the corresponding recess.

5. Rotary fluid engine comprising a casing, a rotor mounted for rotation within said casing and having side walls extending axially and radially outwardly beyond said casing to form end closures for said casing, said rotor having radial slots and said side walls having correspondingly slot extensions axially and radially outwardly enclosed within said side walls, vanes in said slots having corresponding vane extensions in said slot extensions of said side walls, said vane extensions being provided with bearing surfaces in said slot extensions at said side walls, said bearing surfaces each having at least one counterbalancing recess to receive pressure fluid, a passage leading from each said recess to the face of the corresponding vane within said casing opposite such recess, and a member cooperating with each recess to vary the volume of such recess.

6. Engine according to claim 5 wherein said member is stationary in said slot.

7. Rotary fluid engine comprising a casing, a rotor mounted for rotation within said casing and having side walls extending axially and radially outwardly beyond said casing to form end closures for said casing, said rotor having radial slots and said side walls having corresponding slot extensions axially and radially outwardly enclosed within said side walls, vanes in said slots having corresponding vane extensions in said slot extensions of said side walls, said vane extensions being provided with bearing surfaces in said slot extensions at said side walls, at least one of said bearing surfaces of a vane extension having at least two separate counterbalancing recesses, one radially above the other, to receive pressure fluid, and a passage leading from each such recess to the face of the corresponding vane within said casing opposite such recess at a radial level substantially within the range of the uppermost and lowermost edges of the corresponding recess.

8. Rotary fluid engine comprising a casing, a rotor mounted for rotation within said casing and having side walls extending axially and radially outwardly beyond said casing to form end closures for said casing, said rotor having radial slots and said side walls having corresponding slot extensions, and vanes in said slots having corresponding vane extensions in said slot extensions of said side walls, said vane extensions being provided with bearing surfaces in said slot extensions at said side walls, each of said bearing surfaces having a recess to receive counter-balancing pressure fluid from the interior of said casing, and plunger members mounted fixedly in the side wall slot extensions and slidable in said recesses to vary the volume of said recesses.

9. Engine according to claim 8 wherein passage means are provided leading from each said recess to the face of the corresponding vane within said casing opposite such recess.

10. Rotary fluid engine comprising a casing having an inner surface defining a working chamber for receiving and discharging fluids, a rotor in said working chamber

provided with radial slots, vanes substantially radially slidable in the rotor slots and forming, with said rotor and with said working chamber, expanding and contracting intervane spaces in which said fluid is received, said rotor having end wall portions extending in said casing axially and radially outwardly beyond the axial ends of said working chamber and having corresponding slot extensions of said slots, said vanes having corresponding vane extensions axially and radially outwardly enclosed within said slot extensions, said vane extensions being provided with radially and axially extending bearing surfaces in said slot extensions, each of said bearing surfaces having a recess to receive counter-balancing pressure fluid and each said vane having passage means for communicating such recesses with corresponding opening means in the face of the vane within said working chamber opposite the particular recess, each said recess being defined by a groove in the corresponding bearing surface, said groove extending radially outwardly from the bottom of the vane extension and terminating below the top of



the vane extension, and a correspondingly shaped stationary plunger for each said recess, said plunger being stationarily provided in the corresponding slot extension and being dimensioned to be slidably received in said groove for occupying said groove to vary the volume thereof during radial travel of the corresponding vane extension in the corresponding slot extension.

**References Cited** in the file of this patent

UNITED STATES PATENTS

| | | |
|---|---|---|
| 1,692,473 | Smith | Nov. 20, 1928 |
| 1,658,524 | Gurley | Feb. 7, 1928 |
| 2,149,337 | Deming | Mar. 7, 1939 |
| 2,545,238 | MacMillin | Mar. 13, 1951 |
| 2,658,456 | Wahlmark | Nov. 10, 1953 |
| 2,755,741 | Erskine | July 24, 1956 |

FOREIGN PATENTS

| | | |
|---|---|---|
| 9,499 | Great Britain | June 29, 1915 |
| 568,518 | Great Britain | Apr. 9, 1945 |

# UNITED STATES PATENT OFFICE
# CERTIFICATE OF CORRECTION

Patent No. 3,173,375 March 16, 1965

Karl Eickmann

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

In the heading to the printed specification, after line 7, insert the following:

Claims priority, application Japan, March 13, 1958, No. 6745.

column 3, line 6, for "extend" read -- extent --; line 31, for "machiner" read -- machinery --; line 46, after "is" insert -- a --; same column 3, line 58, for "position" read -- positioned --; column 8, lines 23 and 24, for "correspondingly" read -- corresponding --.

Signed and sealed this 21st day of September 1965.

(SEAL)
Attest:

ERNEST W. SWIDER
Attesting Officer

EDWARD J. BRENNER
Commissioner of Patents