Feb. 10, 1959

D. B. GARDINER

2,872,873

POWER TRANSMISSION

Filed Nov. 1, 1954

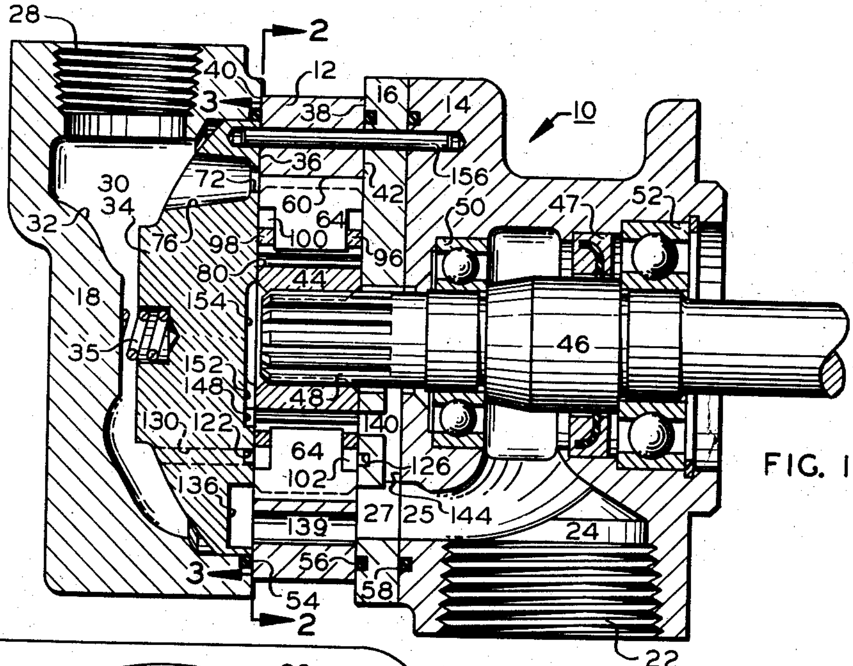

FIG. 1

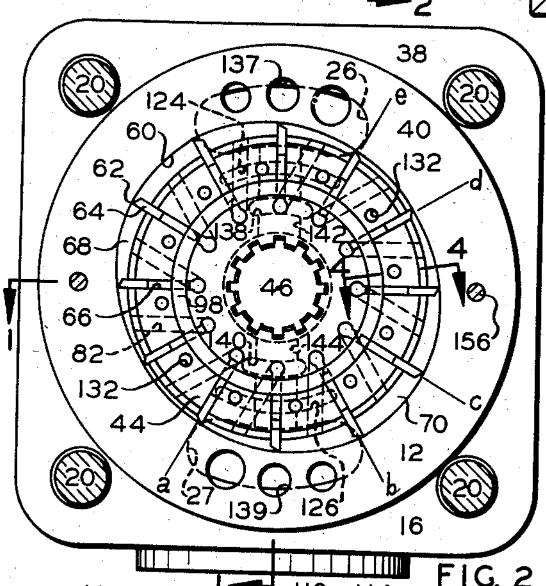

FIG. 2

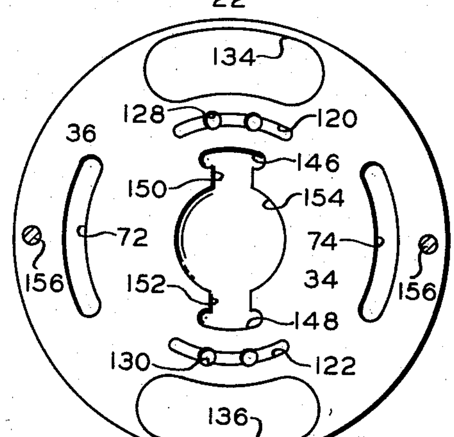

FIG. 3

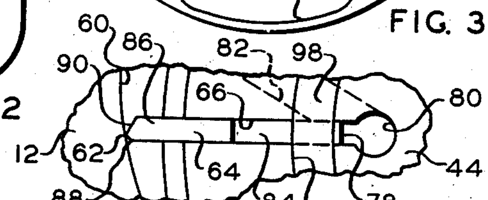

FIG. 5

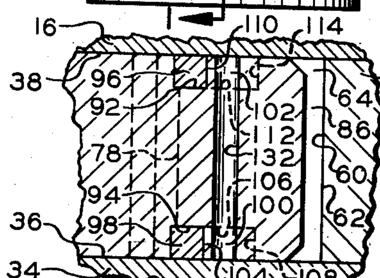

FIG. 4

INVENTOR.
DUNCAN B. GARDINER

BY Ralph L. Tweedale

ATTORNEY

United States Patent Office

2,872,873

Patented Feb. 10, 1959

1

2,872,873

POWER TRANSMISSION

Duncan B. Gardiner, Detroit, Mich., assignor to Vickers Incorporated, Detroit, Mich., a corporation of Michigan

Application November 1, 1954, Serial No. 465,940

3 Claims. (Cl. 103—136)

This invention relates to power transmissions, and is particularly applicable to those of the type comprising two or more fluid pressure energy translating devices, one of which may function as a pump and another as a fluid motor.

The invention is concerned generally with rotary fluid energy translating devices adapted for use in hydraulic power transmission systems and in particular to those of the sliding vane type.

A form of pump in the hydraulic power transmission field utilizes a rotor having a plurality of spaced radial vanes rotatable therewith and slidable relative thereto in slots provided in the rotor. The rotor and vanes cooperate with a vane track in the stator member which defines one or more working chambers between the outer periphery of the rotor and the vane track and through which the vanes pass carrying fluid from the inlet port to the outlet port.

The outer edges of the vanes are adapted to be kept in contact with the track and the spacing between adjacent vanes varies as the rotor turns producing a pumping action. As the rotor turns the vanes are extended outwardly by centrifugal force and are retracted inwardly by cam contour action of the track during the discharge phase of the device. On the discharge phase pressure in the working chamber is exerted over the outer ends of the vanes and tends to collapse the vanes. It has therefore been conventional practice to prevent vane collapse by transmitting pressure to the underside of the vanes.

In one type of device which has found wide commercial use in its field, discharge pressure is continuously transmitted through selected porting to the undersides of the vanes. The success of such pump is due to automatic wear takeup. A large amount of wear on the ends of the vanes and to some extent on the cam track can take place with only a very slight effect on the efficiency of the pump. This is due to the fact that the vanes are maintained in contact with the cam track by centrifugal force and fluid pressure and that the pump action is independent of how much the tips of the vanes have been worn off. Although a certain amount of wear is permissible without destroying the efficiency of the pump, when certain limiting speeds and pressures and volumetric capacities are exceeded the wear rate is so great as to be economically unfeasible. Thus, one limiting factor in this design of pump has been found to be the rate of wear between the outer edges of the vanes and the surface of the vane track under excessive operating conditions.

Because of the limitations of devices wherein outlet pressure is continuously conducted to the inner ends of the vanes, other constructions have been designed which connect the high pressure side of the device to the inner ends of the vanes only on the high pressure or discharge phase thereof and which connect the low pressure or inlet side of the device to the inner ends of the vanes on the suction phase. Although it would appear that this would provide proper balancing of the inward and outward pressures on the vanes as they passed through the low and high pressure working chambers of the device, this is not the case. At higher speeds the vanes become improperly balanced and not only is the volumetric efficiency of the device decreased but likewise its pumping capacity. As the speed of the device increases the centrifugal force urging the vanes against the track increases but this advantage is overcome by an increased pressure differential tending to separate the vanes from the track. As the speed of the rotor increases the pressure differential required to cause the vanes to pump the fluid out of the intervane spaces into the delivery passage also increases and at certain higher speeds becomes so great that the vanes are unbalanced inwardly and move away from the track.

Still other types of devices have been designed to provide a proper unbalanced condition to meet the requirements which utilize intricate valving and porting and even auxiliary booster units.

It is therefore a general object of this invention to provide an improved low cost, efficient and long wearing fluid energy translating device of the sliding vane type.

It is another object of this invention to provide a fluid energy translating device having an improved vane biasing arrangement.

It is still another object of this invention to provide a fluid energy translating device having an improved vane structure and vane biasing arrangement which reduces wear and increases efficiency.

It is a further object of this invention to provide a rotary fluid translating device incorporating sliding vanes which are T-shape in cross section and a pressure transmitting arrangement to both ends and to the adjacently located stepped portions of the vanes which assures proper control of the vanes during the complete cycle at high pressure and high speeds and which reduces wear, and increases volumetric efficiency.

It is also an object of this invention to provide a fluid energy translating device incorporating radially sliding T-shaped vanes with a pressure transmitting system associated therewith to produce a controlled, safe and efficient pressure differential thereon for urging the vanes in contact with the vane track.

Further objects and advantages of the present invention will be apparent from the following description, reference being had to the accompanying drawing wherein a preferred form of the present invention is clearly shown.

In the drawing:

Figure 1 is a longitudinal sectional view of a fluid energy translating device embodying a preferred form of the present invention and taken on line **1—1** of Figure 2.

Figure 2 is a sectional view taken on line **2—2** of Figure 1.

Figure 3 is a sectional view taken on line **3—3** of Figure 1.

Figure 4 is a partial section on an enlarged scale taken on line **4—4** of Figure 2.

Figure 5 is an enlarged partial sectional view of the vanes as illustrated in Figure 2.

Referring now to Figure 1 there is shown a rotary fluid pump indicated generally by the numeral **10**, the housing of which comprises a cam track section **12** sandwiched between a body section **14** having a wear plate **16** and en end cover **18**, all of which are suitably connected to each other by bolts **20**. The body section **14** is provided with an inlet supply connection port **22** having an inlet passage **24** leading therefrom which is branched and terminates in a pair of fluid openings, one of which is shown in Figure 1, and indicated by the numeral **25**, registering with duplicate opposed fluid openings **26** and **27** extending through the wear plate **16** and which are shown in hidden lines in Figure 2.

An outlet connection port **28** is located in the end cover **18** which leads directly from a pressure delivery chamber **30** formed in an enlarged bore **32** of the end cover **18** when a pressure plate **34** is floatably mounted in the bore **32**. The pressure plate **34** and wear plate **16** are each formed with a flat surface indicated, respectively, by the numerals **36** and **38** which abut opposing flat faces **40** and **42** of the cam ring **12** and provide fluid sealing engagement for the immediately adjacent faces of a rotor **44** mounted within the cam ring **12**. The pressure plate is adapted to be urged against a portion of the flat face **40** of the cam track section **12** and in fluid sealing engagement with the rotor by pressure in the pressure chamber. A spring **35** initially biases the pressure plate toward the rotor until pressure builds up in the pressure chamber.

The rotor **44** is driven by a shaft **46** provided with a seal **47**, and extends from the body for connection to a prime mover, not shown. The shaft is spline connected to the rotor at **48** and is rotatably mounted in bearings **50** and **52**, mounted in the body **14**. O ring seals **54** and **56** prevent leakage at the juncture of the end cover **18** and the wear plate **16** with the cam track section **12** while an O ring seal **58** prevents leakage at the juncture of the body **14** and the wear plate **16**.

The inner surface of the cam section **12** forms a track substantially elliptical in shape, indicated by the numeral **60**, against which the outer tips **62** of vanes **64**, the latter of which are slidably mounted in slots **66** of the rotor **44**, are adapted to be maintained in contact. The vane track and the outer periphery of the rotor define two opposed working chambers, indicated by the numerals **68** and **70**, which for the purposes of convenience, may be divided into fluid inlet zones or fluid delivery zones. The fluid inlet zones are those portions of the working chambers **68** and **70** registering with the opposed fluid inlet openings **26** and **27** in the wear plate **16**. The fluid delivery zones are those portions of the working chambers **68** and **70** registering respectively with opposed arcuate fluid delivery ports **72** and **74** in the pressure plate **34**, which are connected to the pressure chamber **30** by means of duplicate passages **76** leading therefrom, one of which is shown in Figure 1.

The vane track **60** includes an inlet zone ramp extending from "*a*" to "*b*," a true arc portion extending from "*b*" to "*c*," a delivery zone ramp extending from "*c*" to "*d*," and another true arc portion extending from "*d*" to "*e*." The track is symmetrical about both its major and minor axes, thus each of the ramp and true arc portions from "*a*" to "*e*" are duplicated in the upper portion of the track. As the ends of the vanes traverse the inlet ramps, the vanes move radially outward with respect to the rotor, and while the vane ends traverse the delivery ramps the vanes move radially inward. In the true arc portions, the vanes partake of no radial movement.

Referring now to Figures 2, 4 and 5, the inner ends of the vane slots **66** are enlarged to form with the inner ends **78** of the vanes **64**, small undervane pressure chambers **80** which undergo cyclic contraction and expansion during rotation of the rotor. Means for transmitting the cyclically changing intervane pressure in the fluid inlet and outlet zones to the undervane pressure chambers is provided by constructing a plurality of angular ports or passages **82**, one for each vane slot, extending from the periphery of the rotor to the undervane pressure chambers. The inner ends **78** of the stem portion of the vanes indicated by the numeral **84** are exposed to pressure in the undervane chambers which tends to urge the vanes outwardly against the track.

The outer end surface of each head portion **86** of the T-shaped vanes comprises a leading tapered surface **88** and a trailing tapered surface **90** forming the tip **62** of the vane which engages the track. The tapered outer end surfaces of the vanes are exposed to the cyclically changing pressure in the intervane spaces as the vanes pass through the fluid inlet and outlet zones which tends to separate the vanes from the track.

The stems **84** of the vanes **64** are slidable through portions of the vane slots **66** which are reduced to the width of the vane stems by constructing annular grooves **92** and **94** in opposite faces of the rotor, which intersect the vane slots slightly above the undervane pressure chambers **80**, and suitably brazing, or press or shrink fitting therein annular insert members **96** and **98**. Two intermediate pressure chambers **100** and **102** are thus formed in each vane slot which are located on opposite sides of each vane stem **84** beneath the vane head **86**. The flat surface **36** of the pressure plate **34**, the outer periphery **104** of the annular insert member **98**, the side of each vane stem facing the pressure plate indicated by the numeral **106**, and a shoulder **108** of the underside of the vane head **86** formed at the juncture of the head and stem of the vane forms the chamber **100**. The flat surface **38** of the wear plate **16**, the outer periphery **110** of the annular insert member **96**, the side of each vane stem facing the wear plate indicated by the numeral **112**, and a shoulder **114** of the vane head **86** formed at the juncture of the vane head and stem on the opposite side of the stem to the shoulder **108** forms the intermediate pressure chamber **102**. The shoulders **108** and **114** of the vane head form movable walls of the intermediate chambers **100** and **102** which causes expansion and contraction of the chambers as the vanes reciprocate in the vane slots. The shoulders **108** and **115** are exposed to pressure in the intermediate chambers **100** and **102** respectively and are effective to urge the vanes outwardly in engagement with the track.

For the purpose of connecting the pressure chamber **30** to the intermediate chambers **100** and **102** on the intervane space expansion phases of the device, diametrically opposed pressure supply ports **120** and **122** are provided in the pressure plate adjacent the rotor and duplicate pressure supply ports **124** and **126** are provided in the wear plate adjacent the opposite side of the rotor. The ports **120** and **122** of the pressure plate **34** and the duplicate ports **124** and **126** in the wear plate are located adjacent the inlet zone ramp "*a*—*b*" and its duplicate ramp of the vane track. Duplicate pairs of drilled passages **128** and **130** in the pressure plate open to the pressure chamber and respectively lead to the ports **120** and **122** for transmitting pressure from the pressure chamber **30** to the ports **120** and **122** and thence to the intermediate chambers **100**. A plurality of drilled passages **132** extending completely through the rotor, one between each vane slot, transmit pressure from the ports **120** and **122** on the pressure plate side of the device through the rotor to the wear plate ports **124** and **126** from whence it is transmitted to the intermediate chambers **102** of the vanes as the chambers **102** pass across said ports during each rotor revolution.

On the intervane space contraction phases of the device means are provided to create a pressure in the intermediate vane chambers **100** and **102** above that existent on the outlet passage side of the device. As the vanes traverse the delivery zone ramps of the track they slide inwardly and fluid is displaced from the contracting undervane chambers **80** and also from the contracting intermediate vane chambers **100** and **102**. The means for creating increased pressure is a restricted flow path through which fluid displaced from the intermediate vane chambers must flow. Specially constructed restricted flow paths may be provided in the pressure and wear plates through which fluid must flow from the intermediate chambers on the displacement phases of the device. However, the formation of specially constructed restricted flow paths may be avoided, as disclosed in the present construction, by utilizing the integral leakage flow paths of the device between the rotor and the adjacent pressure plate and wear plate. As the vanes move inwardly the slight amount of fluid in each of their intermediate chambers is extruded therefrom to restricted flow paths formed between the opposite flat faces of the rotor and the immediately adjoining flat faces **36** and **38** respectively of the pressure plate and the wear plate from whence it is returned to the inlet side of the device through the passageway created between the rotor and shaft splining.

The width of each vane stem is designed to form shoulders having an effective area which when acted upon by outlet pressure on the inlet cycles of the device and by pressure greater than outlet pressure on the fluid delivery cycles of the device will create an outward unbalancing force sufficient to maintain the outer ends of the vanes in contact with the track and below that which would cause excessive wear.

In order to facilitate filling of the fluid inlet zones and the intervane spaces which are expanding on the inlet phase of the device and to produce proper balance on opposite sides of the rotor the fluid inlet ports **26** and **27** in the wear plate are duplicated respectively by fluid inlet ports **134** and **136** in the pressure plate. The opposed duplicate inlet ports **26** and **134** in the wear and pressure plates are connected to each other by a plurality of drilled auxiliary passages **137** in the cam track section **12** while the remaining pair of opposed duplicate inlet ports **27** and **136** in the wear and pressure plates are connected to each other by a duplicate plurality of drilled auxiliary inlet passages **139** also in the cam track section **12**. In addition, auxiliary undervane inlet ports **138** and **140** are provided in the wear plate **16** which are connected to the inlet passages of the body by slots **142** and **144** in the wear plate. Duplicate auxiliary undervane inlet ports **146** and **148** are provided in the face of the pressure plate adjacent the rotor which are connected respectively by slots **150** and **152** to a recess **154** in the pressure plate in alignment with the rotor shaft. Thus the duplicate, opposed auxiliary undervane inlet ports in the wear and pressure plates are not only connected to each other across the rotor through the medium of the undervane pressure chambers on the intervane spaces expansion phases of the device but are interconnected to each other by the passageway formed between the splining of the shaft and rotor which is connected at opposite ends thereof to the slots provided in the wear and pressure plates which lead to their associated auxiliary undervane inlet ports. Dowel pins **156** are utilized to provide proper alignment of the oppositely located porting in the wear and pressure plates with the working chambers of the devices.

In operation, as the rotor turns fluid entering the inlet port **22** is conducted to the expanding spaces between the vanes traversing the inlet ramp "*a*—*b*" and its duplicate inlet ramp by means of the branched inlet passage **24** and its openings thereof and the arcuate inlet ports **26** and **27** of the wear plate.

The duplicate inlet ports **134** and **136** in the pressure plate are respectively connected to the wear plate inlet ports **26** and **27** across the rotor through the medium of the expanding fluid inlet zones of the working chambers of the device and proper filling is assured by the interconnection of the opposed inlet porting of the wear and pressure plates through the drilled passages **137** and **139** in the cam section **12**.

Fluid is conducted from the fluid inlet zones of the device to the expanding undervane chambers **80** by means of the angular rotor passages **82**. Proper filling of the undervane chambers is assured by providing the auxiliary undervane ports **138** and **140** in the wear plate and the duplicate auxiliary undervane ports **146** and **148** in the pressure plate which are interconnected to each other through the undervane chambers of the vanes. The duplicate wear and pressure plate undervane ports are also interconnected to each other and to the inlet side of the device through the slots **150** and **152** in the pressure plate, the passageway formed between the splining of the rotor and the shaft, and the slots **142** and **144** in the wear plate.

The displacement of the device is conducted from the fluid delivery zones of the working chambers through the medium of the delivery ports **72** and **74** of the pressure plate to the pressure chamber **30** and thence by means of outlet connection port **28** to a hydraulic system, not shown.

As the rotor turns the vanes traversing the delivery zone ramp "*c*—*d*" and its duplicate ramp are retracted causing contraction of their associated undervane and intermediate chambers while simultaneously the vanes traversing the inlet zones of the ramp "*a*—*b*" and its duplicate ramp are extended causing an expansion of their associated undervane and intermediate chambers.

The pressures on the undervane surface **78** and the outer tapered surfaces **88** and **90** of each vane are substantially equalized because of the angular drilled passages **82** in the rotor which connect the cyclically changing intervane space pressure to the undervane chambers. The resultant force on the opposite ends of the vanes, however, is unbalanced because of the differential end areas of the vanes. As the vanes pass through the fluid inlet zones across the inlet ramp "*a*—*b*" and the duplicate inlet ramp they are extended outwardly by centrifugal force, but engagement of the tips **62** of the vanes with the vane track is insured by transmitting outlet pressure to the intermediate chambers in which the shoulders **108** and **114** of the vanes are exposed. This is accomplished by transmitting pressure from the pressure chamber **30** through the paired drilled passages **128** and **130** and the intermediate chamber pressure ports **120** and **122** of the pressure plate to the intermediate chamber **100** of each vane traversing the inlet zone ramp and also transmitting pressure chamber pressure to each complementary vane intermediate chamber **102** through the medium of the drilled rotor passages **132** and the intermediate chamber pressure ports **124** and **126** in the wear plate.

Simultaneously, as the complementary vanes pass through the fluid outlet zones of the device across the fluid delivery ramp "*c*—*d*" and its duplicate ramp, they are retracted and fluid in the intermediate chambers **100** and **102** is displaced from the contracting chambers. Fluid displaced from the intermediate chambers **100** must flow to the inlet side of the device through the restricted flow passage formed between the flat face **36** of the pressure plate and the side of the rotor adjacent said plate to the pressure plate recess **154** and thence by the passageway formed between the splining of the shaft and rotor to the wear plate slots **142** and **144** which lead to the wear plate inlet ports **26** and **27**. Fluid displaced from the intermediate chambers **102** must flow through the restricted passageway formed between the flat face **38** of the wear plate and the side of the rotor adjacent said plate to the wear plate slots **142** and **144** leading to the fluid inlet ports **26** and **27**.

The rotor is made thinner than the cam track section **12** to provide running clearance between the rotor and adjacent plate surfaces and thus a restricted passageway is formed on opposite sides of the rotor between the rotor faces and the opposed flat faces of the pressure and wear plates. As fluid displaced from the contracting intermediate chambers must be forced through the restricted flow path the pressure in the intermediate chambers is raised above that existent at the outlet side of the device in the pressure chamber. With pressure greater than outlet pressure existent in the intermediate chambers **100** and **102** acting on the shoulders **108** and **114** and with substantially equalized pressure on the differential end areas of the vanes the vanes are unbalanced outwardly to insure engagement of the tips **62** of the vanes with the vane track. The pressure in the intermediate chambers is adapted to be raised sufficiently to overcome the slight unbalanced condition existing on the ends of the vanes in favor of separating the vanes from the vane track to a favorable unbalanced condition insuring contact of the tips of the vanes with the track but low enough to provide a favorable wear rate condition.

While the form of embodiment of the invention as herein disclosed constitutes a preferred form, it is to be understood that other forms might be adopted, all coming within the scope of the claims which follow.

What is claimed is as follows:

1. A fluid energy translating device comprising a stator including a vane track, a rotor rotatably mounted within the track, a plurality of vanes, each being substantially T-shape in cross section, slidably mounted in the rotor, each vane having a track engaging surface and the intervane spaces adjacent the track undergoing during a cycle of the device alternate expansion and contraction phases as the rotor turns, fluid passages in the stator for conducting fluid to and from the intervane spaces, one of which is a low pressure passage and the other a high pressure passage, means forming a plurality of pressure chambers for each vane, each vane having a surface in each of its associated pressure chambers effective under pressure to urge the vanes in contact with the track, means for transmitting the cyclically changing intervane pressure to one of the chambers of each vane, passage means for connecting the high pressure passage to the other chambers of each vane on the intervane space expansion phase of the device, and means forming a restricted flow path from the other chambers of each vane on the intervane space contraction phase of the device.

2. A fluid energy translating device comprising a stator including a vane track, a rotor rotatably mounted in the track carrying a plurality of sliding vanes the outer ends of which are adapted to engage the track and the intervane spaces undergoing during a cycle of the device alternate expansion and contraction phases as the rotor turns, each vane having a stem portion forming steps on opposite sides of the stem portion effective under pressure to urge the vane against the track, each vane also having an inner surface effective under pressure to urge the vane in contact with the track, means forming separate pressure chambers for each inner surface and step of each vane, means for transmitting the cyclically changing intervane pressure to the inner surface pressure chamber of each vane, means for transmitting pressure substantially equal to the operating pressure of the device to the vane step pressure chambers on one of the intervane space phases of the device, and means for creating a pressure greater than the operating pressure of the device in the vane step pressure chambers on the other intervane space phase of the device.

3. A fluid energy translating device comprising a stator including a vane track, a rotor rotatably mounted in the track carrying a plurality of sliding vanes the outer ends of which are adapted to engage the track and the intervane spaces undergoing during a cycle of the device alternate expansion and contraction phases as the rotor turns, each vane having a stem portion forming steps on opposite sides of the stem portion effective under pressure to urge the vane against the track, each vane also having an inner surface effective under pressure to urge the vane in contact with the track, means forming separate pressure chambers for each inner surface and step of each vane, means for transmitting the cyclically changing intervane pressure to the inner surface pressure chamber of each vane, means for transmitting pressure substantially equal to the operating pressure of the device to the vane step pressure chambers on the intervane space expansion phase of the device, and means forming a restricted flow path from the vane step pressure chambers on the intervane space contraction phase of the device.

References Cited in the file of this patent

UNITED STATES PATENTS

559,324	Dyer	Apr. 28, 1896
1,087,181	Pitman	Feb. 17, 1914
1,805,063	Wrona	May 12, 1931
2,473,309	Stephens	June 14, 1949
2,545,238	MacMillin et al.	Mar. 13, 1951
2,714,858	Markeij	Aug. 9, 1955
2,743,090	Malan	Apr. 24, 1956

FOREIGN PATENTS

433,488	Great Britain	Aug. 15, 1935