April 4, 1967 A. A. SCHMITZ 3,312,175

FLUID POWER CONVERTER

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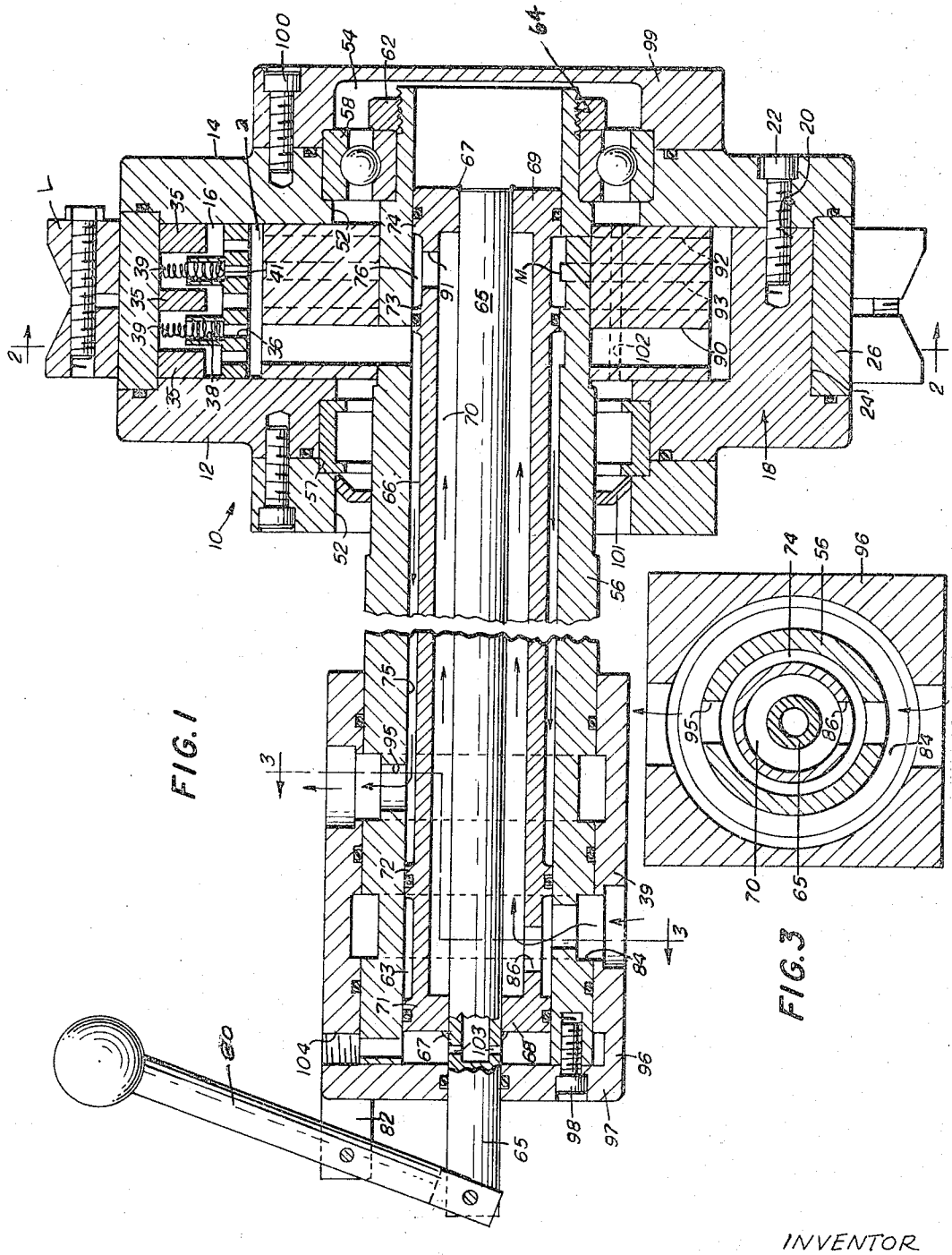


FIG. 1

FIG. 3

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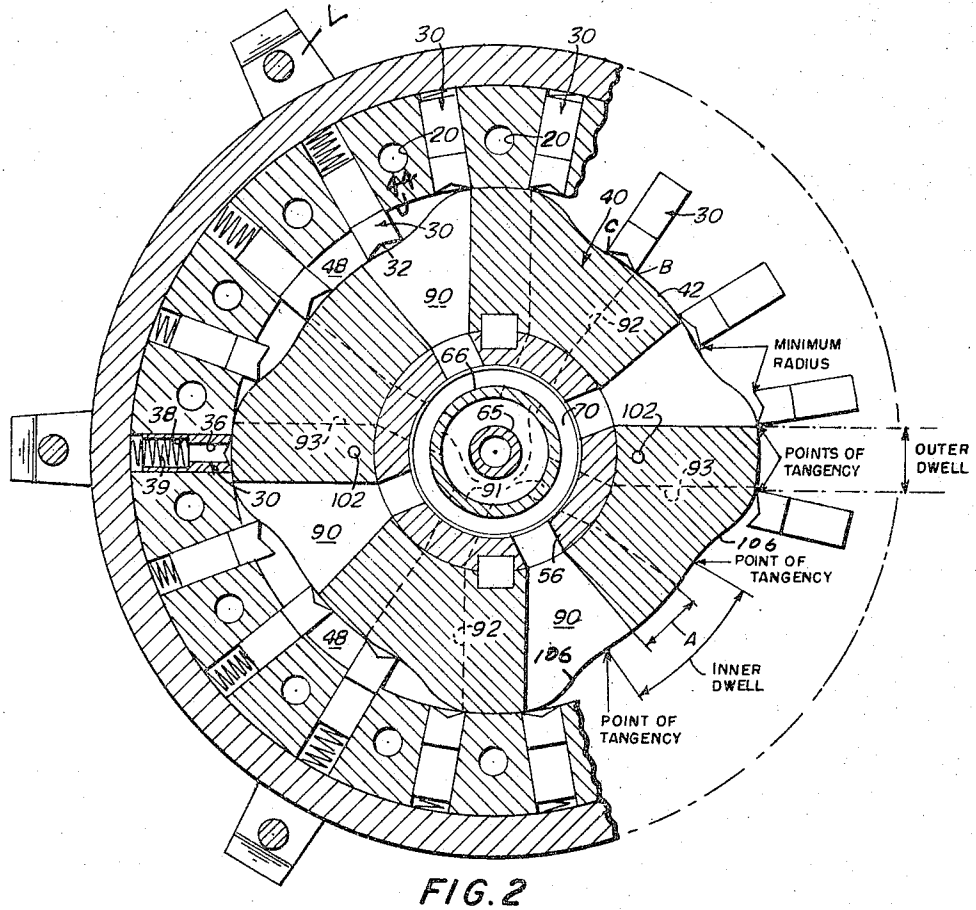


FIG. 2

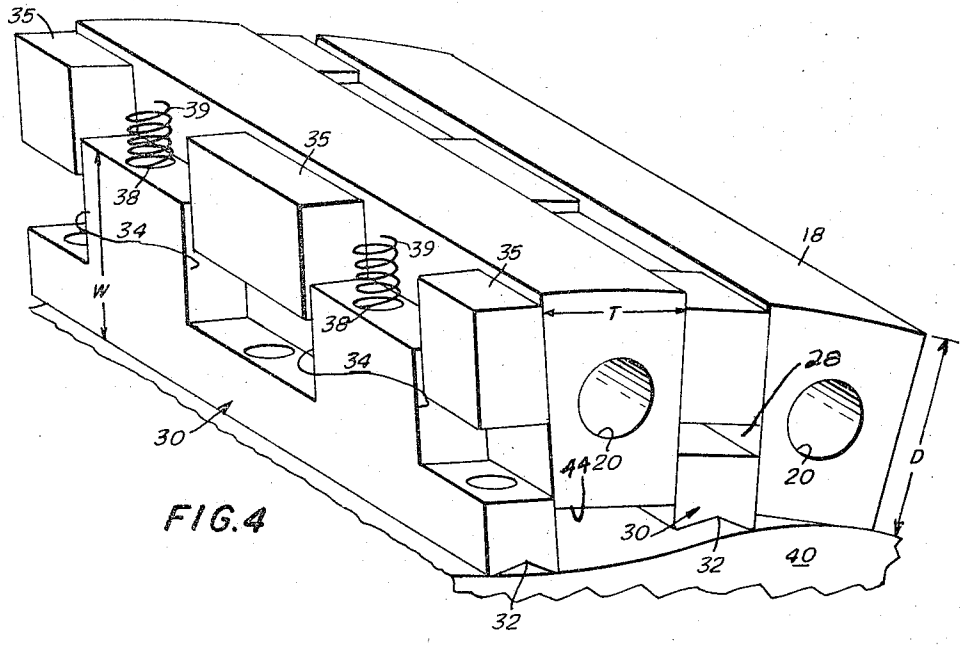


FIG. 4

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FLUID POWER CONVERTER

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This invention relates to a fluid converter and more specifically relates to a fluid converter of the radial vane variety permitting the development of high torque. An important objective of this invention is to provide an efficient fluid translating device having a high torque-to-weight ratio wherein said device can be manufactured with a compactness and economy not known to the hydraulic arts.

A still further objective of this invention is to provide a fluid translating device adaptable in manufacture for low or high power requirements while maintaining the same basic construction and avoiding the high costs inherent in machines having close tolerances normally associated with capabilities of this nature.

Another important objective of this invention is to provide a multi-speed fluid motor wherein a constant level of volume input fluid is utilized to produce more than one output speed.

A still further objective of the invention is to provide a novel annular rotor assembly comprising a plurality of loose segments held together by a tight fitting ring in a manner defining a plurality of channels for the reception of floating vanes. In furtherance of this objective the invention also provides a fluid translating device in which the above-mentioned vane channels can be alternatively machined along the inner face of an annular rotor.

Another important objective of the invention is to provide a fluid translating device wherein an annular rotor is manufactured of a rigid, low-friction material and is retained in position by an external ring of high tensile strength.

A principal and main objective of this invention is to provide a hydraulic unit of the type described wherein one or a series of separate brace members are disposed in spaced relationship along each of the vane channels and the vanes receivable in the channels are notched accordingly to receive the braces. With this arrangement the vanes may reciprocate within the channels a full stroke while the bracing members prevent the pinching of vanes in adjacent channels due to the hydraulic pressures in adjacent channels.

Primarily, this invention finds itself in the field of rotary, vane-type fluid motors and pumps. This field of fluid translators can basically be divided into two major groups; namely, (1) those translators wherein an annular stator member is contoured with an irregular inner surface and surrounds a rotating inner rotor having channels in which radially extending vanes are in sliding contact with the irregular surface, and (2) those fluid translators wherein the inner member is stationary and the outer annular member is the rotor and has vane-receiving channels formed therein. In either classification the vanes must reciprocate freely within their channels. If high pressure against one vane squeezes or distorts the adjacent channel, reciprocation of the adjacent vane becomes difficult, and an inefficient fluid translator is the result. A conventional solution is to provide ample clearance between the vane and channel so that the sides of the channel can collapse somewhat before binding. The result is a noisy operation and an increase in inefficiency, especially when a two-edged vane of the type described hereinafter is used.

Therefore, an important and principal objective of this invention is to provide a rotor wherein all the vane receiving channels are braced against collapse and distortion by braces which provide a clearance for the vanes without subtracting from the radial dimension of the vane to thereby provide a vane having adequate proportions and at the same time received in channels permitting a full deciprocating stroke.

Another important objective of the invention is to provide a rotor assembly which can be economically manufactured while providing means for maintaining control of the width of the vane slots to thereby permit a close fitting vane which operates more efficiently and quietly than loose fitting vanes which are more common in the art.

A still further objective of the invention is to provide a fluid pump or motor with a stator or rotor having a large, even number of lobes forming displacement chambers, in which each pair of diametrically opposed chambers have inlet or exhaust passages axially spaced from passages in other pairs of displacement chambers to thereby permit a selection of combinations of passages and active displacement chambers for producing various capacities and speeds from the same volume of input fluid. In this manner the entire unit remains in hydrostatic balance regardless of the combination selected.

A further objective is to provide a fluid pump or motor having a stator or rotor carrying closely spaced slidable vanes making possible higher torque by means of a greater number of reaction lobes on the opposing rotor or stator.

In summary, therefore, the invention provides a fluid converter wherein a brace or braces are positioned within vane receiving channels which braces permit the use of a large number of closely spaced vanes. With a large number of closely spaced vanes an increased number of reaction lobes can be used for obtaining additional torque. In turn the additional torque makes feasible the use of a greater number of axially spaced pairs of inlet and outlet passages for better speed selection and range of speed control.

These and other important objectives and advantages of the invention will be more fully understood upon a reading of the following specifications taken in view of the attached drawings wherein:

FIGURE 1 is a longitudinal cross-sectional view of the device;

FIGURE 2 is a cross-sectional view taken along the line 2—2 in FIGURE 1;

FIGURE 3 is a cross-sectional view taken along the line 3—3 of FIGURE 1; and

FIGURE 4 is a partial perspective of the interior of the device with portions broken away.

Referring now with more particularity to the drawings where like numerals indicate like parts, the numeral **10** indicates the housing of this invention. The housing **10** includes a pair of end plates **12** and **14** which define a chamber **16** therebetween. The plates are spaced from one another via a plurality of segments **18** which are threadedly bored at **20**. The plates define a rotor housing **21**. Each bore receives a plurality of bolts **22** for securing the end plates together. The two end plates are peripherally machined at **24** to receive an annular ring **26** which forms a cylindrical side wall for the chamber.

The segments **18** form a series of circumferentially arranged channels **28** therebetween. In the embodiment shown, the segments extend toward end plate **14** and are integral with the end plate **12**. It should be understood that the segments can be integral with the plate **14** or independent segments secured to both plates by bolts at either end. Each of the channels slidably receives a vane member **30**. The vane members extend between end plates **12** and **14** and are formed with a groove **32** at their inner ends. The vanes along their outer ends are

formed with a plurality of notches 34. Each of the vanes is bored at 36 and counterbored at 38. Each counterbore receives a spring biasing member 39. The springs 39 have one end engaging the inner surface of ring 26 and their other ends engaging ledges 41 at the bottoms of the counterbores 38. The vane is thereby normally biased to an engagement with the periphery of an interior stator member 40.

The stator member 40 is generally cylindrical and is formed with a contoured periphery 42 that has a maximum diameter substantially the same as the inner diameter of the circle defined by the inner surfaces 44 of the segments 18 and other diameters of less dimension which form, with the segments, displacement chambers 48. In the known fashion, as fluid is delivered and exhausted from these displacement chambers, the pressure exerted causes the rotor housing 21 to rotate with respect to the stationary member 40.

The end plates 12 and 14 are respectively bored at 52 and 54 to receive a fluid delivery tubular member 56. When used with a vehicle, member 56 is load bearing in addition to functioning as a fluid delivery conduit. The length of member 56 at the break shown in FIGURE 1 is of a sufficient length to provide a mounting area for securement to the vehicle or other machine. The end plates 12 and 14 and the apparatus carried thereby are rotatably mounted with respect to member 56 via the raceways 57 and 58. The member 56 is secured longitudinally to the rotor by a nut 62 which engages the threads 64 formed at the inner end of the member 56.

Slidably received in member 56 is a hollow operating shaft 65. In spaced concentric relationship with the shaft 65 between the shaft and member 56 is a sleeve 66. The sleeve is longitudinally fixed to the shaft by welds 67 at either end. The sleeve is formed with inwardly directed end flanges 68 and 69 which form with the shaft 66 and sleeve 65, a fluid transferring annulus 70. The sleeve is also formed with outwardly extending annular flanges 71, 72, 73 and 74 each in sealing and sliding engagement with the interior of the stationary tubular member 56. With the member 56, the flanges 71 and 72 form a porting annulus 63, the flanges 72 and 73 a porting annulus 75, and flanges 73 and 74 a porting annulus 76. The shaft assembly is moved longitudinally of the tubular member 56 via an operator's handle 80 and the fulcrum lug 82.

The tubular member 56 is formed with a fluid inlet 84 in communication with the porting annulus 76 and fluid transferring annulus 70 via aperture 86 in the sleeve 66. Fluid introduced under pressure, through aperture 84 is communicated to the pressure passages 92 and 93 of the stator 40 via a plurality of apertures 91 formed in the sleeve housing and the porting annulus 76. Note that although all the exhaust passages 90 are in the same plane, inlet passages 92 and 93 are longitudinally spaced along the stator axis with respect to each other.

As will be well understood by those skilled in the art, as pressure is introduced into chambers 48, the rotor 21 will rotate, causing the vanes to reciprocate within their respective channels 28. In other words, the pockets between the vanes in the chambers 48 are subjected to pressurized fluid as they pass the passages 92 and 93 and exhaust their fluid through the passages 90.

It should be noted that the flanges 73 and 74, as positioned in FIGURE 1, permit the fluid under pressure in the annulus 76 to enter all of the passages 92 and 93 of rotor 40. This fluid is exhausted through passages 90 into the porting annulus 75 and finally to the tank via outlet 95 of tubular member 56. With the handle 80 pivoted counterclockwise from the position of FIGURE 1, the sleeve or shaft is moved to the right and flange 73 assumes position M between chambers 92 and 93, cutting off passages 93 from pressurized fluid and communicating them to exhaust annulus 75. All pressurized fluid must now pass through passages 92 which are in communication with half of the displacement chambers. As also

will be understood by those skilled in the art, by moving the sleeve in this manner, the rotor is caused to rotate at twice the speed but with only half the torque that is available in the FIGURE 1 position because of the decrease in displacement.

In other words, when shifting sleeve assembly 65–66 is moved to the right (sec. 2—2) fluid passages 93 are cut off from annulus 76 and connected to annulus 75. Since passages 93 are now in communication with passages 90 through annulus 75 a free circulation of fluid can take place through the two displacement chambers 48 that are adjacent to passages 93. All working fluid is directed through the remaining two displacement chambers 48 adjacent to passages 92, thus causing the doubling of the rotational speed of annular rotor housing 21.

When pressurized fluid is introduced through annulus 75 the direction of rotation is reversed but the speed remains the same since passages 93 are still common with 90 through 75. Only two displacement chambers 48 adjacent passages 92 are subjected to differential pressure.

It should be understood at this point that for ease of description purposes, the embodiment disclosed is of a four lobe, two speed type but that due to the ability to substantially increase the number of vanes, the invention makes practicable to give the art an eight or more lobe, multi-speed (3 and above) translator. A greater number of speeds can be obtained by axially forming another set of passages along the length of the stator similar to passages 92 and 93.

The fluid translator of this invention is enclosed at the handle end by a cup-shaped member 96 formed with an end portion 97 secured to the sleeve 66 by way of bolts 98. The member 96 in its skirt portion is formed with appropriate apertures to mate with the inlet opening 84 and the outlet opening 95 of tubular member 56. At its other end, the fluid translator is enclosed by way of a cap 99 secured to the plate 14 by way of bolts 100.

FIGURE 4 of the invention discloses the novel vane configuration which is of major significance to this invention. Critical to sliding vane operation is the width dimension W and its relationship with the recess depth D in which it reciprocates. Also of importance is the dimension T of the segment 18 because the sturdiness and thickness of this member is effective as to the amount of pinching caused in adjacent channels due to a pressure transference. In order to reduce dimension T to a minimum, to permit a many-vaned pump, this invention teaches the use of spacer braces 35 within the channels to prevent any deflection of the member segments 18 which would pinch against the vanes 30 in adjacent slots. Normally, such spacers would so diminish the dimension D that the vane 30 would have an insufficient width W to operate effectively. The utilization of notches 34 to receive the braces 35 permits the full dimension W of the vane to reciprocate freely in the slots and thus all features of an efficient and compact machine are maintained.

It should be noted that the chordal distance A between the leading edge of the exhaust passage 90 and the trailing edge of the inlet passages 93 is less than the length of the inner dwell by at least the thickness of one vane. This permits the relatively heavy vanes 30 to ride the fall surface of the stator completely to the lower dwell prior to absorbing any thrust. Also, it should be noted that the leading edge of the vane (see B in upper right-hand sector of FIG. 2) receives the thrust initially as it closes the pressure passage. The edge B is relieved of load as soon as the trailing edge C passes this point. The importance of the fit of the vanes 30 in their slots 28 is that both edges of the vanes must seal. Hence the vanes must not tilt in their slots.

A fluid seepage between the stator 40 and the side plates or between the stator and the member 56 is retained in the housing by the annular rotary seal 101. The stator is formed with a seepage passage 102 to permit all such leakage to reach the inner end of member 56 and flow

through the hollow shaft **65** to discharge from the unit through holes **103** and external drain port **104**. Clamping lugs L for attaching pneumatic tire rims or other devices are provided on the housing. In the embodiment shown the lugs are a part of ring **26**. Ring **26** can be manufactured with a desired type of clamp or securing means without affecting the remaining elements of the invention.

It should be noted that one complete surface section of the four-lobed stator disclosed consists of (FIG. 2) the "outer dwell" surface, the "inner dwell" surface and the minimum acceleration surfaces **106** between the two dwell portions.

In a general manner, while there has been disclosed in the above description, what is deemed to be the most practical and efficient embodiment of the invention, it should be well understood that the invention is not limited to such an embodiment as there might be changes made in the arrangement, disposition, and form of the parts without departing from the principle of the present invention as comprehended within the scope of the accompanying claims.

I claim:

1. A fluid power converter comprising a stator having a contoured periphery, a rotor encircling said stator and defining an annular chamber therewith, means mounting said rotor for rotation about said stator, said rotor including, an annular ring, a plurality of equally and closely spaced separate segments extending inwardly of said ring and held together by said ring and defining a plurality of vane receiving channels, a vane in each of said channels, spring means in each of said channels biasing the inner ends of said vanes to engagement with said contoured surface, brace means along each of said channels forming at least one space therebetween, said brace means being of sufficient depth with respect to the depth of said channel to preclude deflection of the segments forming said channel under fluid pressure exerted thereon, and said vanes having at least one notch along the length thereof receiving said brace means.

2. A fluid power converter comprising a stator having a contoured periphery of an even number of equally spaced lobes and depressions, a rotor encircling said stator, means mounting said rotor for rotation about said stator, said rotor including an annular ring, a plurality of closely spaced separate segments extending inwardly of said ring contiguous to said lobes and held together by said ring and forming a plurality of displacement chambers with

said depressions, said segments defining a plurality of channels, a vane slidably received in each of said channels and each vane having at least one notch formed in one end thereof, brace means between adjacent segments received in the notch of each of said vanes, said brace means being of sufficient depth with respect to the depth of said channel to preclude deflection of the segments forming said channel under fluid pressure exerted thereon, the other end of said vanes being in sliding engagement with said contoured periphery, means rotatably mounting said rotor about said stator, a source of pressurized fluid, conduit means in said stator for delivering said fluid to said displacement chambers and exhausting fluid therefrom upon a rotation of said rotor with respect to said stator.

3. A fluid power converter comprising a first member having an exterior surface, a second member encircling said first member and having an interior surface adjacent said exterior surface, means mounting said members for relative rotation therebetween, one of said members having a plurality of closely spaced separate segments defining a plurality of parallel channels on its respective surface, a vane slidably received in each of said channels, at least one brace in each of said channels and said vane having a notch to slidably receive the brace, said brace being of sufficient depth with respect to the depth of said channel to preclude deflection of the segments forming said channel under fluid pressure exerted thereon.

4. In a fluid power converter of the sliding vane type, a vane carrying member comprising an annular ring, a series of alternate closely spaced separable segments and spacers held together by said ring, said segments forming side walls of vane-receiving channels and said spacers determining the width of said channels, vanes slidably received within said channels, said spacers being of sufficient depth with respect to the depth of said channels to preclude deflection of the segments forming said channels under fluid pressure exerted thereon.

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