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POWER TRANSMISSION

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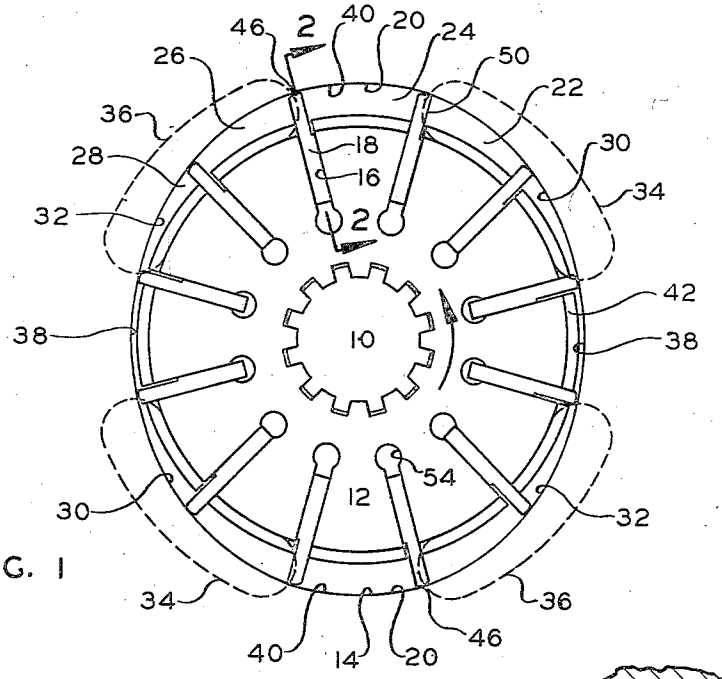


FIG. 1

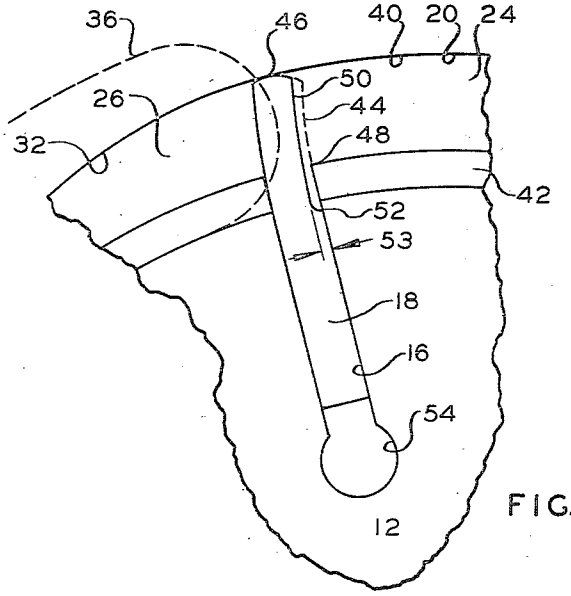


FIG. 3

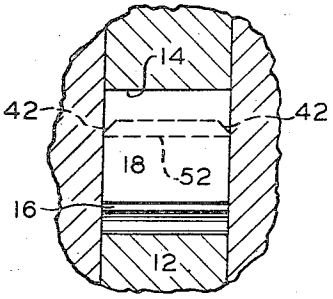


FIG. 2

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POWER TRANSMISSION

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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.

More particularly the invention relates to an improvement in sliding vane type pumps and motors for use in such a transmission.

The invention is well adapted for use with a radially sliding vane pump such as illustrated in the patent to Harry F. Vickers, No. 1,989,900, although it is pointed out that the present invention is applicable to both pumps and motors and further, is not limited to those having radial vanes.

One of the limiting factors in the service life of such units is the development of localized wear points on the cam track that controls the in and out movement of the vanes. These surface disturbances interrupt the smooth contour of the track to cause noisy and inefficient operation.

It is an object of this invention to provide an improved vane type pump or motor in which wear between the track and vanes is greatly reduced.

Another critical wear area in such units is the area of contact between the vanes and the walls of the slots in which they are carried. Wear at this point increases the clearance space through which leakage can take place and thus reduces the efficiency of the unit.

It is also an object of this invention to provide a vane type unit in which wear between the vanes and the guiding slots is minimized.

Another object is to provide these advantages without appreciably increasing the cost of units currently in production and, further, to enable the low cost modification of existing units to incorporate the present invention.

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 transverse partial section showing pumping mechanism of the type described in the above-mentioned patent to Vickers, and corresponds to Figure 8 of that patent.

Figure 2 is a partial section taken on line 2—2 of Figure 1.

Figure 3 is an enlarged scale view of a segment of the mechanism of Figure 1.

Referring now to Figure 1, there is shown a shaft 10, rotor 12 and a cam ring 14. Rotor 12 has a plurality of slots 16 therein each containing a radially slidable vane 18. The space between rotor 12 and cam ring 14 forms a pair of crescent-shaped working spaces 20. Vanes 18 extend across the working spaces 20 to abut the cam track and thus form between each pair of adjacent vanes a working chamber. The numerical designations 22, 24, 26 and 28 have been given several of these working

chambers. As the working chambers pass through the crescent-shaped working spaces those chambers bounded by a vane which is riding along either of the ramps 30 will be increasing in size and hence those areas of the working spaces adjacent the ramps 30 may be termed inlet zones. Conversely those working chambers bounded by vanes which are abutting the ramps 32 are decreasing in size and those areas of the working spaces adjacent ramps 32 may be termed outlet zones.

The dotted outlines of the kidney-shaped ports 34 indicate the positional relationship in which the actual ports overlie the inlet zones of the pumping mechanism. Similarly the dotted outlines 36 indicate the positional relationship in which the outlet ports overlie the outlet zones. Fluid enters and leaves each working chamber axially through the open sides of the working chambers which cooperate with the kidney-shaped ports. The various track ramps 30 and 32 are separated by true arc track portions 38 and 40 which are concentric with the axis of rotor 12. Rotor 12 has its sides chamfered at 42 to facilitate the entry of fluid to the working chambers and the discharge of fluid therefrom.

The construction thus far described is conventional and the description can be amplified by reference to the heretofore noted patent to Vickers. During the time the tip of any particular vane is traveling on one of the ramps 30 or 32 the pressure on opposite sides thereof is equalized, since the working chambers on each side of the vane communicate with a common port. While the tip of any vane moves across the true arc portions 40 of the cam track 14 there is a pressure differential thereacross. This results from communication of the leading working chamber with a high pressure port while the trailing working chamber communicates with one of the low pressure ports. Similarly, as the tip of any vane traverses one of the true arc segments 38, the leading working chamber communicates with one of the low pressure ports while the trailing working chamber communicates with a high pressure port. It can be seen that the extended tips of the vanes are thus subjected to a bending force in the region of the true arc sections of the cam track. Since the vanes crossing the minor diameter true arc portions 38 are almost fully retracted, the bending moment thereon is minor. However, for those vanes traversing the major diameter true arc portions 40 the bending force thereon is quite severe.

The conventional construction utilized in the past has provided a vane of substantially uniform thickness to engage the sides of the vane receiving slot and extend therefrom to engage the cam track. The heretofore noted patent to Vickers is typical of this type of structure. In the drawing accompanying this specification, the conventional vane structure is indicated by the dotted outline 44 in Figure 3. In pumps utilizing the conventional construction, there has been noted excessive wear of the vane track in the area indicated by the numeral 46, and excessive wear of the rotor and the individual vanes at the point indicated by the numeral 48. Further, unless the conventional vanes are very carefully fitted to the vane slots, internal leakage in such units may exceed acceptable limits.

The present invention has virtually eliminated all these disadvantages from pumping units to which it has been applied. The invention provides a relief 50 on the trailing side of each of the vanes 18. The relief 50 extends inwardly from the tip of each vane to a point within the rotor and below the chamfer 42. It can be seen that the depth of relief 50 is such that it never emerges from rotor 12, thus the trailing side of vane 18 continuously contacts the slot 16 on the line of termination 52 of relief 50.

Deflection of a vane 18 due to the pressure differential between working chambers 26 and 24 is clearly shown in

Figure 3. The degree of deflection has been somewhat exaggerated from that which occurs in an actual unit and the tangential depth **53** of relief **50** has been correspondently exaggerated. In an actual unit, the depth of the relief **50** might be of the order of 5 to 10 thousandths of an inch.

During operation of the unit, as a particular vane reaches the position illustrated in Figure 3 it will be flexed due to the pressure differential thereacross. A conventional vane will bear back heavily against the rotor at the point **48**, and the extending tip will be bent backward. While the vane is in the position shown, the tip of the vane contacts the discharge ramp **32** which urges the vane inward. It will be seen that if the tip of the conventional vane is to be moved radially inward, one of two things must occur. Either the radius of curvature of the extending vane tip must be increased, or the tip of the vane must move forward on the vane track, relative to the forward movement of rotor **12**. In either event it will be appreciated that greatly increased force is transmitted between the tip of the vane and the vane track at the point **46** and between the side of the vane and the rotor at the point **48**. It has been found that in actual units the tip of the vane will dig into the vane track in the region **46** and that the vane slot will be enlarged in the region **48**, while the trailing side of the vanes wear excessively at **48**. Further, since the vane tends to cock about the point **48**, the clearance space between the vane and the slot at the trailing side of the vane will provide a leakage path for the high pressure fluid in the undervane spaces **54**. This leakage path includes the space between the vane and the slot and a triangular opening formed by the chamfer **42** below the point of contact **48** between the vane and rotor.

In units built according to the present invention, the above noted disadvantages are eliminated. By relieving the trailing side of the vane, binding at the point **48** is eliminated. Even though the vane is flexed at the moment it contacts the discharge ramps **32** it can be moved inward by the track without the necessity for increase in the radius of curvature of the extending tip, and also without forward movement of the tip of the vane on the track relative to the movement of the rotor **12**. Further, since the point **52** about which the vane will tend to cock is below the chamfer **42**, any clearance space between the trailing side of the vane and the slot is closed by a full width seal at **52**. Volumetric efficiency of the unit is thus greatly improved.

There has thus been provided an improved, low cost vane type unit having higher efficiency and longer life than prior units.

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. In a fluid pressure energy translating device of the sliding vane type: a pair of relatively rotatable members; means forming vane receiving slots in one of said members; a vane track on the other of said members, said track having portions at varying distances from said one member; a vane in each of the slots; a slot engaging, plane-faced inner portion on each of said vanes; and an outer portion on each vane extending from the slot to engage said track to control in and out movement of the vane in the slot, the entire width of said outer portion being reduced in thickness relative to said inner portion, inward for a distance greater than the maximum extension of said vane from said slot, providing substantial clearance between the slot and at least one side of said outer portion in all vane positions and thereby avoiding binding of the vane in the slot due to flexure of the extended portion induced by pressure differential thereacross.

2. In a fluid pressure energy translating device of the sliding vane type: a pair of relatively rotatable members; means forming vane receiving slots in one of said members; a vane track on the other of said members, said track having portions at varying distances from said one member; and a vane extending from each of said slots to engage said track to control in and out movement of the vane in the slot, said vanes each having a flat plane side and a stepped side to provide a reduced thickness outer portion across its entire width, said reduced portion extending inward for a distance greater than the maximum extension of said vane from said slot, providing substantial clearance between the slot and at least one side of said outer portion in all vane positions and thereby avoiding binding of the vane in the slot due to flexure of the extension induced by pressure differential thereacross.

3. In a fluid pump of the sliding vane type: a pair of relatively rotatable members; means forming vane receiving slots in one of said members; a vane track on the other of said members, said track having portions at varying distances from said one member; and a vane extending from each of said slots to engage said track to control in and out movement of the vane in the slot, said vanes each having a leading flat plane side and a trailing stepped side to provide a reduced thickness outer portion across its entire width, said reduced portion extending inward for a distance greater than the maximum extension of said vane from said slot, providing substantial clearance between the slot and at least one side of said outer portion in all vane positions and thereby avoiding binding of the vane in the slot due to flexure of the extension induced by pressure differential thereacross.

4. In a fluid pressure energy translating device of the sliding vane type: a rotor; means forming a vane track encircling said rotor and having portions at varying distances from said rotor; means forming substantially radial vane receiving slots each providing an outward opening in the periphery of said rotor; a vane slidable in each of said slots to extend radially therefrom to engage said track, said vanes having a plane-faced inner portion of substantially the same width as said outward openings and having a substantially reduced thickness outer portion extending to a depth greater than the maximum extension of said outer portion from said slots, whereby at least one side of said outer portion makes no contact with said rotor at any time.

5. In a fluid pump of the sliding vane type: a rotor; means forming a vane track encircling said rotor and having portions at varying distances from said rotor; means forming substantially radial vane receiving slots each providing an outward opening in the periphery of said rotor; a vane slidable in each of said slots to extend radially therefrom to engage said track, said vanes having a plane-faced inner portion of substantially the same width as said outward openings and having a substantially reduced thickness outer portion extending to a depth greater than the maximum extension of said outer portion from said slots, whereby the trailing side of said outer portion makes no contact with said rotor at any time.

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