Oct. 9, 1951 W. P. VAN LAMMEREN 2,570,889

FLUID IMPACT COUPLING AND CLUTCH

Filed July 29, 1947

Fig. 1.

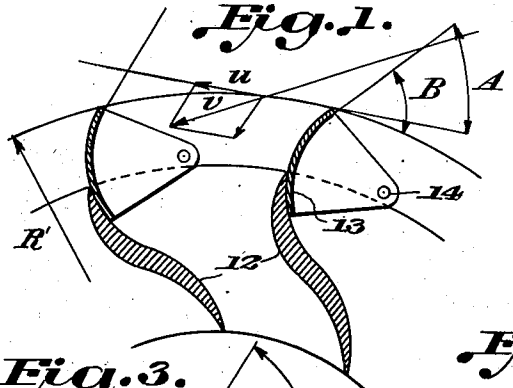


Fig. 2.

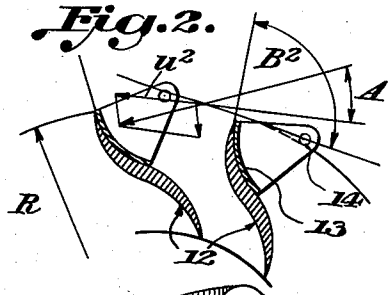


Fig. 3.

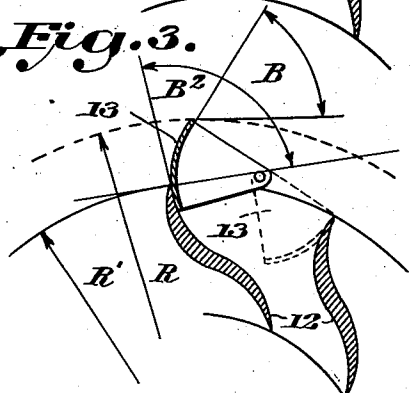


Fig. 4.

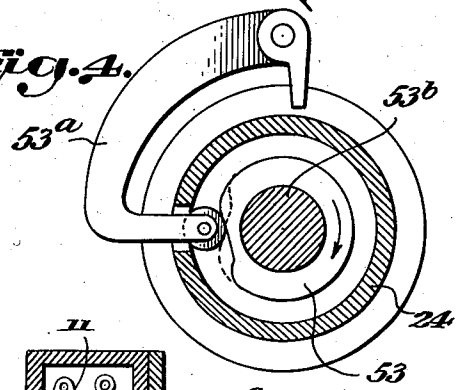
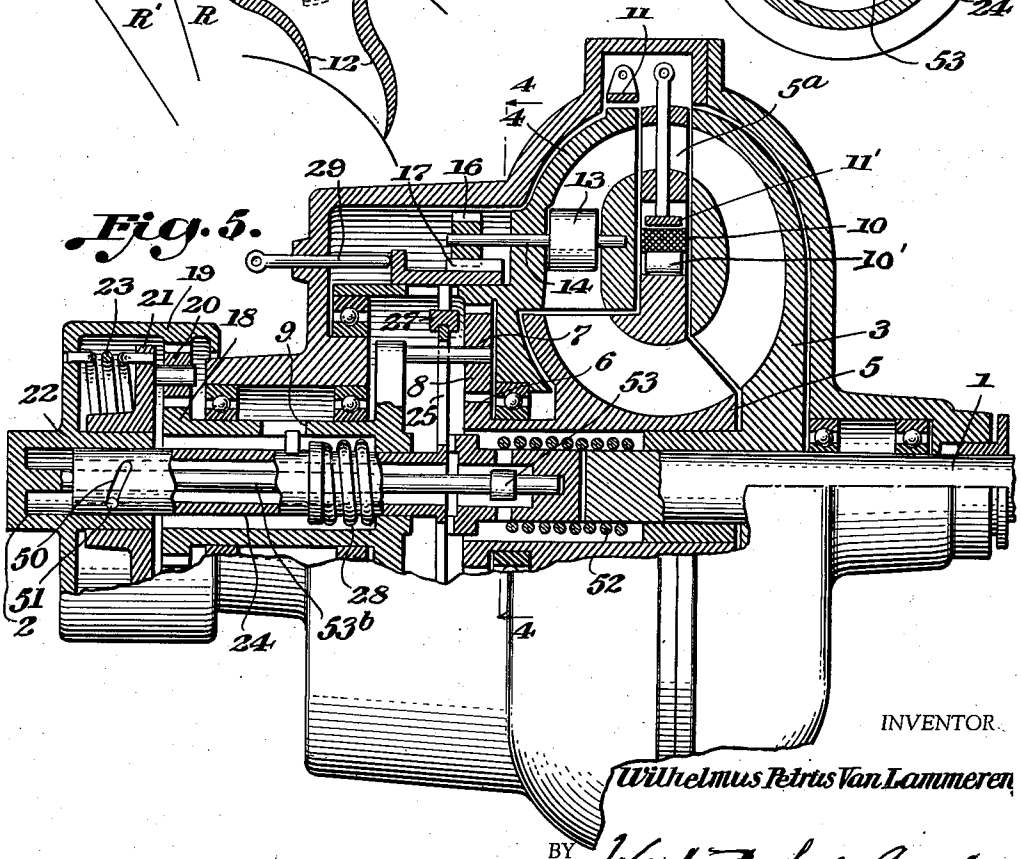


Fig. 5.



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FLUID IMPACT COUPLING AND CLUTCH

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8 Claims. (Cl. 192—3.2)

1

This invention relates to hydraulic power transmissions of the type having an impeller, a driven vane wheel rotated by the impeller and a guide vane wheel. Although the invention is capable of application to transmissions of the type stated in which the guide vane wheel is permanently stationary, it may be applied with particular advantage in cases where the guide vane wheel is capable of rotation, and particularly in those cases where the driven vane wheel and guide vane wheel can exchange functions to provide a reversal of the drive as described in the specifications of my British Letters Patent Numbers 458,143 and 485,080. The invention may be applied with especial advantage to the arrangement described in the latter specification where the drive is transmitted to the driven shaft through an epicyclic gear coupled to the driven and guide vane wheels and also to the driven shaft and operating in conjunction with brakes and a free wheel on the vane wheels to enable reversal or an automatic variation of the torque output to take place.

In hydraulic transmission of the type stated the efficiency for different ratios between the speeds of a fluid-supplying vane wheel and a fluid-receiving vane wheel depends partly upon the angle of entrance of the fluid with respect to the fluid-receiving vanes and partly upon the effective diameter of the fluid-receiving vane wheel.

With a view to maintaining a high degree of efficiency notwithstanding changes in the relative speeds of such vane wheels, the present invention provides a hydraulic power transmission device of the type referred to in which there is a fluid-receiving vane wheel having vanes which are adjustable to vary simultaneously the angle they make with the fluid stream and the effective diameter of the fluid-receiving vane wheel in a sense which is complementary, in tending to attain a position of maximum efficiency; that is to say, in adjusting the vanes to make a smaller angle with the fluid stream the effective diameter of the wheel is increased, and vice versa.

By making use of this feature of the invention, if there is any alteration in the ratio of the speeds of the two vane wheels, both the angle of entrance of the fluid-receiving vanes and the diameter of the fluid-receiving vane wheel are adjusted in a sense such that if the vanes make a relatively small angle with the fluid stream and the resulting velocity component tending to rotate the vane wheel is small, the diameter, and therefore the applied torque, is large, whereas if the vanes make a large angle with the fluid stream and the

2

velocity component is large, the diameter and the torque are small. Such a manner of adjustment tends to maintain maximum efficiency.

In carrying the invention into effect the adjustable vanes may be composed each of a fixed and a movable part, the latter being slidable on the fixed part in a manner such as to be capable of varying its angle with respect to the direction of entry of the fluid stream and also of varying the effective diameter of the wheel; for example the movable part may be mounted for pivoting about an axis extending parallel to the axis of rotation of a vane wheel having its passages extending in a generally radial direction, said movable part having an arcuate surface contacting with a corresponding surface on the fixed part of the vane so that by rotation of said movable part the angle thereof with the periphery of the vane wheel and also the effective diameter of the latter are varied.

A further feature of the invention consists in the provision of means whereby the adjustment of the vanes can be effected automatically in accordance with the load for the time being applied to the driven shaft. For this purpose the invention proposes to employ a coupling having two relatively movable driving and driven parts arranged to undergo relative movement upon variation of the relative value of the driving torque and the load torque applied thereto, the said parts having a resilient interconnection adapted to absorb and balance an increase in the difference between the load torque and the driving torque whilst permitting the said relative movement, and being included in the final drive of a transmission according to the present invention, and so connected to the adjustable vanes as to be able to effect the requisite adjustment thereof upon variation of the ratio between the load torque and the driving torque. Such a coupling per se forms the subject of my British Patent application No. 20,274 of July 12, 1939, entitled "Improvements in or Relating to Mechanical Power Transmission Devices."

In connection with the last mentioned feature of the invention another feature consists in the provision, in cases where the drive is reversible, of means for rendering inoperative the automatic control and enabling the vanes to be adjusted by hand to the most suitable position for reverse operation.

A further development of the invention consists in the provision, in a transmission having adjustable vanes in accordance with the invention, of means for obtaining a mechanical drive directly from the driving to the driven shaft. This fea-

ture of the invention is advantageously employed in conjunction with the feature already referred to according to which the adjustment of the vanes is effected automatically in accordance with the load, the combined arrangement being such that when the direct mechanical drive takes place the vanes occupy a predetermined position. In this connection, a further feature of the invention is that said predetermined position is such that the passage for fluid between the vanes is completely or partly closed, so as substantially to stop the circulation of fluid when the condition of direct drive exists.

In order that the invention may be clearly understood and readily carried into effect reference will now be made to the accompanying drawings, in which the invention is illustrated in general and also as applied to a hydraulic transmission embodying the principles disclosed in the specification of my British Letters Patent No. 485,080, that is to say a transmission embodying driven and guide vane wheels the functions of which are reversible and in which the drive is transmitted from said driven and guide vane wheels to the driven shaft by means of an epicyclic gear.

In the drawings:

Figures 1, 2 and 3 are diagrammatic side elevations of individual vanes adjustable to vary simultaneously the angle of entry and the effective wheel-diameter, the views illustrating the position of the vanes and the form thereof;

Figure 4 is a detail section taken on line 4—4 of Figure 5; and

Figure 5 is a part sectional elevation of a transmission device having the invention applied thereto and illustrating the general arrangement of the parts, and illustrating the provision of a direct mechanical drive.

Referring to the drawings, the hydraulic transmission device illustrated comprises a driving shaft 1 and driven shaft 2, the former being connected to an impeller 3 which supplies fluid to a driven vane wheel 4, guide vane wheel 5 and guide van ring 5a. The guide vane wheel 5 is connected to the sun wheel 6 of an epicyclic gear, the annulus 7 of which is provided on the driven vane wheel 4 and the planet wheels 8 of which transmit the drive to an intermediate driven shaft 9. The guide vane wheel 5 has at its periphery a ring 10 having a free wheel connection 10' with the vane wheel whilst brakes 11 and 11' can be applied as required to the periphery of the driving vane wheel 4 and the ring 10 respectively. The arrangement so far described is a constructional variation of that described in the specification of my prior British Patent No. 485,080 and its operation may be briefly stated as follows: For forward operation the brake 11' is applied to the ring 10 and upon setting the impeller in rotation it first causes the driven vane wheel 4 and therefore also the annulus 7 of the epicyclic gear to be driven, the guide vane wheel 5 remaining stationary. The drive to the intermediate driven shaft 9 thus takes place through the epicyclic gear. As the speed of rotation increases and the reverse torque decreases the reaction pressure on the guide wheel 5 gradually falls to zero and the guide wheel commences to rotate with the driven vane wheel 4 until eventually the impeller, guide wheel and driven vane wheel are all rotating at substantially the same speed and a condition approaching that of direct drive is obtained. For reversing the drive the brake 11' of the ring 10 is released and the brake 11 of the wheel 4 applied.

Referring now to the diagrams illustrating the effect of varying the angle of the vanes for a given velocity of fluid stream, in Figures 1 and 2, it can be seen that if V is the velocity of the fluid stream entering a vane wheel and A the angle of entrance thereof with the periphery of the vane-ring, then if the angle B of the vanes with the periphery of the vane-ring is relatively small, the peripheral component of V indicated by U will also be relatively small and consequently the speed of rotation of the vane wheel will be low. As is shown in Figure 2 however, if, for the same velocity of the fluid stream at the same angle of entrance, the angle B_2 of the vanes is relatively great, then the peripheral velocity component U_2 will also be great and the speed of rotation of the vane wheel will be high. If, in conjunction with the adjustment of the angle, the diameter of the vane wheel is also altered so that for a small vane angle (see Figure 1) the diameter R is large, then the torque will also be great for a given power input. Similarly, if for a large vane angle as illustrated in Figure 2, the diameter R' is reduced, the torque will also be reduced for a given power input. In both cases the efficiency may be at a maximum.

It can be seen from the drawings that each vane consists of a fixed part 12 and a movable part 13, the latter being pivotally mounted upon a pivot pin 14 extending transversely of the vane, the part 13 having an arcuate rear surface which slides upon a corresponding arcuate surface of the fixed part of the vane during pivotal movement.

It will be seen that by rotating the adjustable part 13 of the vane about its pivot pin the angle of the vane with respect to the periphery of the vane-ring can be adjusted, and at the same time the effective diameter of the vane wheel is varied automatically. As is evident from the drawing, the shaping of the adjustable part 13 of the vane is such that as the vane angle is reduced to give a lower wheel velocity, the effective diameter of the wheel is increased to give an increased torque. Preferably the leading edge of the adjustable part of the vanes is shaped with a flat surface as shown to receive the fluid stream.

In order to enable the vanes to be adjusted during rotation of the driven vane wheel the following construction is employed; the pivot pin 14 of each vane part 13 is provided with a pinion 16 which meshes with teeth provided on a sleeve 17 which rotates with the driven vane wheel 4 but is axially adjustable with respect thereto. The teeth of pinion 16 and the co-operating teeth are disposed at an angle to the axis of rotation so that by reciprocating the sleeve the pinion 16 can be caused to rotate and thus adjust the vanes. Such reciprocation and consequent adjustment is effected automatically in accordance with the ratio between the torque output and the load by means of a coupling comprising a sun wheel 18 provided on the intermediate driven shaft 9, which sun wheel is connected to an annulus 19 by means of planet wheels 20, the annulus 19 being secured to the final driven shaft 2 and the planet wheels 20 being mounted on a carrier 21 free to rotate on a hub 22 of the annulus. The said carrier 21 is connected by a helical spring 23 to the annulus 19, the connection of said spring 23 being such that if there is an increase in load which tends to hold the final driven shaft 2 stationary, the

spring will be wound up. It will be seen that there is therefore a relative movement allowed between the intermediate and final driven shafts 9, 2, whenever the ratio between the driving and the load torque varies, and such relative movement is made use of to effect adjustment of the pinion-controlling sleeve 17 by mounting in the hub of the annulus a spindle 24 having at its end outwardly extending arms 25 which pass between the intermediate driven shaft 9 and gears 6, 7, 8 and engage slidably in an annular groove at 27 in the axially adjustable sleeve 17.

The position of the vanes illustrated in Figure 3 of the drawings is that which is most suitable for forward rotation of the driven vane wheel 4 at low speed and is also the most favourable position for the reverse operation of the transmission when the brake 11 is applied to the driven vane wheel 4 and the latter is stationary. During reverse rotation the helical spring is inoperative and the drive is transmitted through the coupling by abutments (not shown) on the carrier 21 and annulus 19. However, for that position of the parts of the coupling the adjustable vanes will occupy a different position and to attain the preferable position it is necessary to be able to adjust the vanes by hand for reverse rotation. To this end the arms 25 are not rigidly connected to the threaded spindle 24 but have a spigot and socket connection therewith, the arms 25 being urged towards the spindle 24 by means of a spring 28. When it is required to adjust the vanes by hand the arms 25 are moved against the action of the spring 28 by a suitable connection 29 which may conveniently be interconnected with the control for applying the brakes 11 to the respective vane wheels.

The figures illustrate a suitable construction providing for a direct mechanical drive between the driving shaft 1 and driven shaft 2, as well as for the complementary feature of controlling the vanes so as substantially to close the fluid passages under certain conditions of operation.

Referring to Figure 5, the adjustable vane-parts 13, are movable through a substantial arc and their initial position (before the spring 23 is wound up) is that illustrated in Figure 1, the greater part of the fluid-passage between the fixed vane-parts being obscured. As the spring is wound up due to increasing load, the vane-part 13 moves successively to the positions illustrated in Figures 2 and 3. The automatic axial adjustment of the arms 25 to effect adjustment of the vanes can be initiated through a pin and slot connection 50, 51 between the spindle 24 and hub 22 and the slot can be shaped exactly as required to produce any desired rate of movement of the vanes.

For providing a direct mechanical drive a coil-spring clutch 52 is provided adapted to couple the driving shaft 1 and intermediate shaft 9, the coupling and uncoupling of said shafts being controlled by a cam 53 mounted on a rod 53*b* rigid with the driven shaft 2. Upon relative rotation of the shaft 2 (and therefore the cam 53) and the shaft 9, the cam operates an arm 53*a* (see Figure 4) to cause clamping and loosening of the coil clutch 52 in a manner similar to that described in the British application No. 20,274 of 1939.

The operation is as follows: In the idle position of the transmission the initial tension of the spring 23 is such that it will be balanced by a torque of the turbine wheel of predetermined value (say 0.6 time the normal motor torque).

The position of the adjustable vane part 13 is then as shown in Figure 1. If now the transmission begins to work, a small fluid stream passes through the opening at the leading edge of the adjustable vane parts. If now the torque increases to a predetermined value (say 0.9 time the motor torque) then the clamping spring 52 comes in action and the direct-drive is made effective. In this position of the spring 23 and shafts 2 and 9, the passage of the fluid streams is wholly or nearly closed. In consequence the friction losses are reduced to a minimum, and the device acts like a mechanical direct-drive coupling. If the load torque increases to a further predetermined value of say 1.1 times the motor torque the clamping spring 52 is released. The position of the adjustable vane part is then as shown in Figure 2 and the further operation of the device is as hereinbefore described. The shape of the groove 50 is made such that, during the clamping of the spring 52 (i. e. during the mechanical direct-drive) the vane-parts are adjusted so that the circulation of the fluid stream is substantially stopped, in order to avoid friction losses.

It will be appreciated that although the adjustable vanes have been described as applied to the driven vane wheel 4, they may also be applied to the guide vane ring 5*a* which for the purposes of this specification is regarded as constituting a permanently fixed vane "wheel," or to the guide wheel 5. In each case the vane "wheel" constitutes a fluid-receiving wheel, the vane wheel from which it receives fluid being regarded as the fluid-supplying wheel.

I claim:

1. An hydraulic power transmission of the Fottinger convertor type, comprising a driving and a driven shaft, a vaned impeller on the driving shaft, a fluid receiving vane wheel on the driven shaft and a guide vane wheel, said vaned members co-operating to form a fluid power transmitting circuit, adjustable vanes on the fluid receiving vane wheel whereby the angle of entrance of the fluid stream is varied in accordance with the difference between the driving torque and the load torque applied to the transmission, said vanes tending to close the passage of the fluid stream as said difference in the driving and driven torques tends to zero, and mechanical means for coupling and uncoupling automatically the driving and driven shafts in order to obtain a direct drive, said means acting in accordance with the variation of the rates between the driving and load torques.

2. An hydraulic power transmission of the Fottinger convertor type, comprising a driving and a driven shaft, a vaned impeller on the driving shaft, a fluid receiving vane wheel on the driven shaft and a guide vane wheel, said vaned members co-operating to form a fluid power transmitting circuit, adjustable vanes on the fluid receiving vane wheel whereby the angle of entrance of the fluid stream is varied in accordance with the difference between the driving torque and the load torque applied to the transmission, said vanes consisting of a fixed part and a movable part, the movable part being slidable on the fixed part and tending to close the passage of the fluid stream as said difference in the driving and driven torques tends to zero, and mechanical means for coupling the driving and driven shafts in order to obtain a direct drive, said means acting in accordance with the variation of the ratio between the driving and load torques.

3. An hydraulic power transmission as claimed in claim 2 wherein the leading edge of the movable part of each vane has a plane surface which receives the fluid stream.

4. An hydraulic power transmission of the Fottinger convertor type, comprising a driving and a driven shaft, a vaned impeller on the driving shaft, a fluid receiving vane wheel on the driven shaft and a guide vane wheel, said vaned members co-operating to form a fluid power transmitting circuit, adjustable vanes on the fluid receiving vane wheel whereby the angle of entrance of the fluid stream is varied in accordance with the difference between the driving torque and the load torque applied to the transmission, said vanes consisting of a fixed part and a movable part, the movable part being slidable on the fixed part and tending to close the passage of the fluid stream as said difference in the driving and driven torques tends to zero, and mechanical means for coupling automatically the driving and driven shafts in order to obtain a direct drive, said means acting in accordance with the variation of the ratio between the driving and load torques, the slidable part of each vane having a resilient extension adapted to co-operate with and follow the curve of the fixed part of the vane.

5. An hydraulic power transmission of the Fottinger convertor type, comprising a driving and a driven shaft, a vaned impeller on the driving shaft, a fluid receiving vane wheel on the driven shaft and a guide vane wheel, said vaned members co-operating to form a fluid power transmitting circuit, adjustable vanes on the fluid receiving vane wheel whereby the angle of entrance of the fluid stream is varied in accordance with the difference between the driving torque and the load torque applied to the transmission, said vanes consisting of a fixed part and a movable part, the movable part being slidable on the fixed part and tending to close the passage of the fluid stream as said difference in the driving and driven torques tends to zero, the movable part being also slidable in a manner such as to vary the effective diameter of the wheel, and mechanical means for coupling automatically the driving and driven shafts in order to obtain a direct drive, said means acting in accordance with the variation of the ratio between the driving and load torques.

6. An hydraulic transmission as claimed in claim 5 wherein the movable part of each vane is mounted on the spindle of a pinion having inclined teeth and engaging correspondingly inclined teeth on an annulus rotating with the vane wheel but axially adjustable with respect thereto, the rotation of said pinion which is effected by said axial adjustment of the annulus causing adjustment of the vanes.

7. An hydraulic power transmission of the Fottinger convertor type, comprising a driving and a driven shaft, a vaned impeller on the driving shaft, a fluid receiving vane wheel on the driven shaft and a guide vane wheel, said vaned members co-operating to form a fluid power transmitting circuit, adjustable vanes on the fluid receiving vane wheel whereby the angle of entrance of the fluid stream is varied in accordance with the difference between the driving torque and the load torque applied to the transmission, said vanes tending to close the passage of the fluid stream as said difference in the driving and driven torques tends to zero, and mechanical means for coupling automatically the driving and driven shafts in order to obtain a direct drive, said means acting in accordance with the variation of the ratio between the driving and load torques and brought into operation when the adjustable vanes occupy a predetermined position such as completely or partly to close the passage for fluid between the vanes and stop or impede the circulation of fluid.

8. An hydraulic power transmission as claimed in claim 7 wherein said mechanical coupling means comprises a coil-spring clutch controlled by a cam mounted on a rod rigid with the driven shaft.

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