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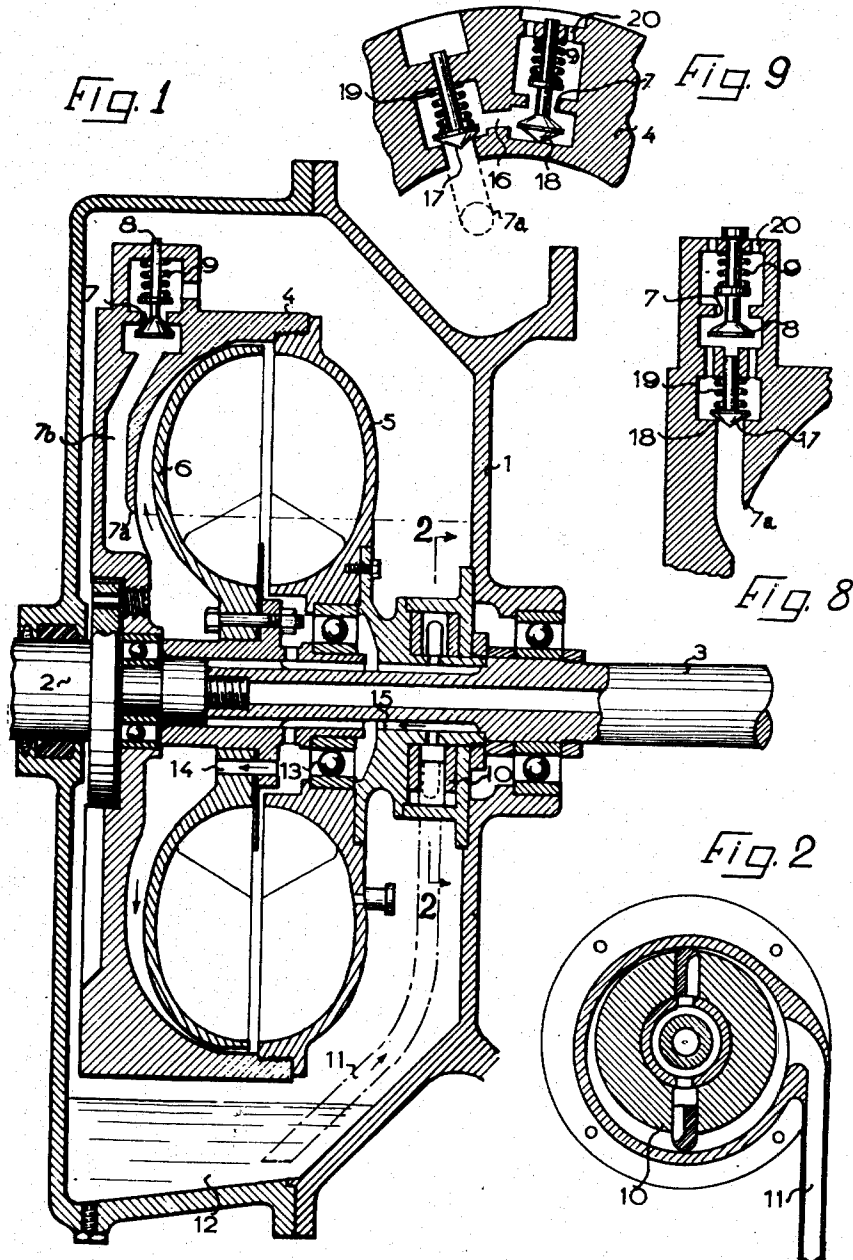
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ROTARY HYDRAULIC COUPLING

Filed May 22, 1951

2 Sheets-Sheet 1



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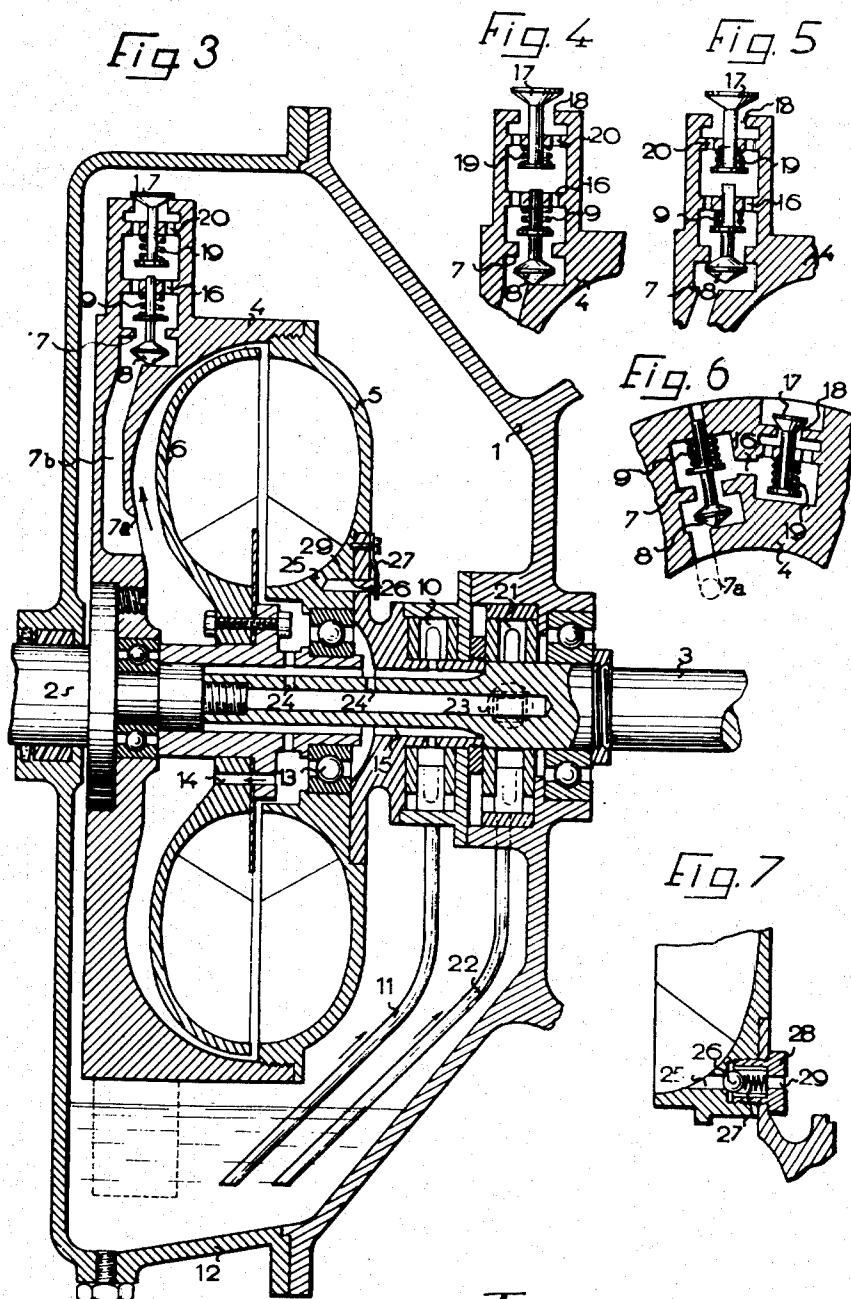
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ROTARY HYDRAULIC COUPLING

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3 Claims. (Cl. 60—54)

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The present invention relates to the Föttinger type of hydraulic power transmitter or fluid coupling in which driving and driven members are each provided with an annular series of vanes, blades or passages, and the two members are so designed and so juxtaposed that the impelling liquid while travelling in the operating circuit defined by said blades or passages effects the rotation of the driven member in response to the rotation of the driving member.

Couplings of this type offer many advantages such as a very high degree of starting smoothness, this operation being effected, without any special pedal, under the mere control of the normal acceleration of the engine, a higher degree of safety against cutting out of the engine, silent operation, absence of wear and so on. These couplings, however, have the drawback in that it is difficult to obtain at the same time the desired disengagement of the coupling, as the engine is throttled down, and the total power transmission with a minimum slippage between the driving and driven shafts as the engine is speeded up.

It is an object of the invention to provide a hydraulic coupling of the type described which is capable of ensuring both a complete declutching, as the engine is throttled down, and a substantially instantaneous re clutching, as the engine is speeded up again.

According to the present invention there is provided a hydraulic coupling comprising a pump wheel rotating with a driving shaft, a turbine wheel secured to a driven shaft, a casing fixed to said pump wheel for rotation therewith, and enclosing said turbine wheel, means driven by said driving shaft for continuously supplying liquid under pressure into said casing, at least one liquid outlet port in the peripheral wall of said casing, overflow means for conducting said liquid from the interior of said casing to said outlet port, said overflow means determining the highest level of a liquid volume constantly retained in said casing when said outlet port is open, and centrifugally operated valve means associated with said outlet port for preventing liquid overflowing said level from discharging through said outlet port when said casing rotates above a predetermined speed.

The liquid may be supplied from a sump constituted by the bottom of the housing of the device, the liquid supplying means being adapted to recycle continuously the liquid evacuated from the casing into said sump.

With the arrangement according to the inven-

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tion, when the speed of the driving shaft reaches a predetermined value, the outlet port or ports are closed and the above mentioned feeding means ensure a proper filling of the device. When the engine slows down under a given value, the strength of the spring tending to open each valve becomes higher than the combined action of the centrifugal force and liquid pressure acting in the opposite direction, and causes the opening of the outlet port controlled by said valve, so that the above mentioned predetermined amount of the liquid is expelled under the action of centrifugal force through said outlet port thus reducing the maximum torque which can be transmitted by the coupling.

In a preferred embodiment of the invention, the overflow means for conducting liquid to the outlet port, is constituted by a passage extending radially from the interior of the casing to an outlet port in the periphery of the casing, said port being controlled by said valve.

This arrangement provides three advantages, viz. the valve is located at a point where its operating conditions are optimum, the evacuation of the liquid, since it takes place through a radial passage, is rapidly effected by centrifugal force, and the casing cannot be emptied below a certain level determined by the position of the inner end of said passage so that the re-filling of the casing, when the engine is speeded up again, takes place rapidly, which ensures a substantially instantaneous re clutching.

For each outlet port there may be provided a secondary outwardly opening valve mounted in series with the above mentioned valve means (hereafter also called the primary valve) either upstream or downstream with respect to the same, said secondary valve being adapted to be brought by slight centrifugal force into its opening position, while it is urged into its closing position by an opposing spring suitably calibrated, so that said secondary valve remains closed under a relatively slow predetermined speed lower than the closing and opening speeds of the main valve.

With this arrangement, the secondary valve is always opened before the primary valve is closed. When due to deceleration, the primary valve opens, the secondary valve remains also opened until said speed falls under the above mentioned relatively low speed at which said secondary valve closes. At and below said last mentioned speed and even when the engine is completely stopped, said secondary valve always remains closed.

Additional feeding means actuated from the

driven shaft may be provided for feeding the casing when the driven shaft is rotating and the driving shaft is at rest.

Bleed means constituted by outlets may be provided in the pump wheel in the vicinity of the casing axis and controlled by spring loaded non-return valves opening outwardly under a predetermined pressure, while preventing the liquid from escaping at pressures below said pressure.

Since the casing can never be emptied, any torque generated by the driven shaft will be transmitted to the driving shaft thereby starting or assisting in starting the engine. Said engine then transmits to the driving shaft a speed sufficient to cause the opening of the secondary valve, thus re-establishing the normal running conditions.

The invention is illustrated by way of example in the accompanying drawings in which:

Fig. 1 is an axial section of a hydraulic coupling according to one embodiment of the invention.

Fig. 2 is a section along line 2—2 of Fig. 1.

Fig. 3 is a view similar to Fig. 1 of another embodiment comprising two valves for each outlet, the secondary valve being located downstream of the primary valve.

Fig. 4 is a detail view of the primary and secondary valves in the position corresponding to a speed comprised within the secondary valve opening speed and the primary valve closing speed.

Fig. 5 is a part view of said valves in the normal running condition, i. e. at a speed higher than the closing speed of the primary valve.

Fig. 6 shows an alternative arrangement in which the valves are circumferentially spaced in order to provide a reduction of the radial thickness of the casing.

Fig. 7 shows an alternative embodiment in which the bleed means are constituted by spring loaded ball valves.

Fig. 8 is a modification of Fig. 3 in which the secondary valve is located upstream of the primary valve.

Fig. 9 shows an alternative construction of the embodiment of Fig. 6 in which the valves are circumferentially spaced.

Referring to the drawings, there is shown at 1 the housing of a hydraulic coupling, at 2 the driving shaft and at 3 the driven shaft.

On driving shaft 2 is fixed for rotation with said shaft a casing 4 carrying a driving centrifugal pump 5 which projects, when shaft 2 rotates, a strong oil stream onto the blades of a receiving turbine 6 fixed on the driven shaft 3 for rotation therewith. In the example shown, there are provided, in the outer periphery of casing 4, a number of liquid outlet ports 7 controlled by valves 8 adapted to be brought by centrifugal force into their closing position, a spring 9 urging said valves towards the opening position. The outlet ports 7 communicate with the working chamber of the coupling through overflow means which, in the example shown, are constituted by a radial duct 7b opening into said working chamber at a point thereof lying nearer to the axis of the coupling than the point of said chamber remotest from said axis, so as to determine a highest level, indicated by line 7a, of constantly-retained liquid, as hereinafter more fully explained. There is shown at 10 a rotary pump, the rotor of which is rotated from shaft 2, said pump sucking through a pipe 11 oil contained in the sump 12. The oil discharged from said pump flows, as indicated by arrows in Fig. 1, through ducts 15 provided in the surface of shaft 3 and

passages 14 provided in the hub of the turbine wheel 6, and is directed towards the working space of the coupling comprised between the vanes of members 5 and 6. A portion of said oil is by-passed through bearings 13, which improves their lubrication.

It will be understood that when the driving shaft 2 rotates faster, the valves 8 are seated under the action of centrifugal force, so that casing 4 is closed. When the speed of shaft 2 slows down to less than a predetermined value, springs 9 open valves 8 by overcoming at the same time, the centrifugal force and oil pressure exerted upon the valve surface. Below said speed, the oil amount comprised within the circumferential level indicated by line 7a and corresponding to the inner openings of the radially extending overflow ducts 7b leading to the outlet ports 7 is expelled through these outlet ports and a sufficient declutching takes place instantaneously. Once this declutching is effected, the volume of oil retained within casing 4 remains substantially constant, the evacuation rate of the valve being materially higher than the discharge rate of pump 10, so that although the latter is still driven by shaft 2, the oil level is in no way affected. Thus, as soon as the engine is speeded up again, while valves 8 are closed under the action of centrifugal force, the clutching is effected almost instantaneously. It is to be understood that when the casing 4 rotates at a predetermined speed, the spring force should be sufficient to open the valves not only against centrifugal force but also against the oil pressure exerted upon the outer surface of valves 8.

In the embodiment shown in Fig. 3 a secondary valve 17 is provided downstream of the primary valve 8, these two valves being interconnected through passages 16 and 20; said secondary valve 17 is urged towards its seat by a spring 19. The two valves thus open in opposite directions. As shown in Fig. 8, the secondary valve 17 may be arranged upstream of the primary valve 8.

An additional rotary pump 21 operated from the driven shaft 3 discharges the liquid sucked through a pipe 22 from sump 12 into the axial duct 23 and radial ducts 24 to feed the working space of the coupling in the same manner as the above described pump 10. Bleed holes 25 located in the pump wheel 5 in the vicinity of the casing axis and intended to evacuate the liquid excess, are provided with non-return valves 26 maintained in closed position by springs 27. As shown in Fig. 7, these non-return valves may consist of balls 26 loaded by calibrated springs 27.

This device operates as follows:

As described with reference to Figs. 1 and 2, when the engine is normally running, the primary valve 8 urged by a centrifugal force capable of overcoming the strength of spring 9 is closed. The secondary valve or valves 17 are opened for the same reason. The casing fills up and the coupling transmits a normal torque. The excess of liquid is evacuated through bleed holes 25, since the non-return valves 26 are opened under the action of the discharge pressure of pump 10. In the case when said non-return valves are constituted by balls, the objectionable action of centrifugal force liable to prevent said balls from tightly seating is eliminated by guiding said balls in bores provided with longitudinal grooves capable of ensuring the evacuation of the liquid as shown in Fig. 7.

When the engine is throttled down, the pri-

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mary valve 8 opens when the speed falls below a predetermined value due to the resulting decrease in the centrifugal force; the liquid is expelled out through passages 16 and 20 and through the secondary valve 17 which is still open, until the inner level of the liquid, due to the decrease in the centrifugal force, reaches the level indicated by line 7a. Then, the driven shaft is no longer materially driven. If the speed further slows down, the secondary valves 17 close, due to the further decrease of the centrifugal forces, whereupon they remain closed until the device completely stops. During this time, the device is partly filled again, but since, in the meantime, the rotation speed varies between a very low value and zero, this causes no material rotation of the driven shaft. Since the pressure of liquid within casing 4 decreases with decreasing rate of supply, the non-return valves 26 are closed by their springs 27 to prevent the liquid from escaping through the passages 25.

Thus, in the case when the driving shaft is to be started from an external force applied upon the vehicle, i. e. from driven shaft 3, the amount of liquid stored in the casing may be sufficient to allow the transmission of a torque from the driven shaft to the driving shaft even with a high degree of slip between said two shafts. However, in case of comparatively high resisting torques, the additional pump 21 operated by the driven shaft will supply a sufficient quantity of liquid, the valves 17 remaining closed, to decrease the slip between the driving and driven shafts and cause the engine to be started. As soon as the engine runs, the operation phases described with reference to Figs. 1 and 2, take place normally.

In the embodiments shown in Figs. 6 and 9, the secondary valve 17 is located side by side with respect to the primary valve 8, so as to reduce the radial thickness of the casing wall. The operation of the device is unchanged, the passage 16 interconnecting the two valves extending circumferentially.

It will be understood that the invention is in no way limited to the type of valves described and that the same, instead of being of the axial type shown may be made in any other shape such as lever-controlled tilting valves. Moreover, said valves may be located in other zones of the casing. The above-mentioned non-return valves may be also located in any desired point, provided that they open outwardly of the casing.

What I claim is:

1. A hydraulic coupling comprising a pump wheel rotating with a driving shaft, a turbine

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wheel secured to a driven shaft, a casing fixed to said pump wheel for rotation therewith and enclosing said turbine wheel to thereby define a work chamber, means driven by said driving shaft for continuously supplying liquid under pressure into said casing, a liquid outlet port in the peripheral wall of said casing, overflow means within said casing establishing a communication between said outlet port and a point of said work chamber nearer to the axis of the coupling than the point of said chamber remotest from said axis, an inwardly opening check valve arranged to cooperate with said outlet port to close the same by centrifugal force when said casing rotates above a predetermined speed, and resilient means adapted to maintain said valve in opening position as long as said casing has not attained said predetermined rotational speed.

2. A hydraulic coupling comprising a pump wheel rotating with a driving shaft, a turbine wheel secured to a driven shaft, a casing fixed to said pump wheel for rotation therewith and enclosing said turbine wheel to thereby define a work chamber, a rotary pump driven by said driving shaft for continuously supplying liquid under pressure into said casing, at least one radial liquid outlet duct within said casing and extending from the peripheral wall thereof to a point of said work chamber nearer to the axis of the coupling than the point of said chamber remotest from said axis, and a centrifugally operated spring-loaded valve mounted in said duct to close the same when said casing rotates above a predetermined rotational speed.

3. A hydraulic coupling according to claim 2, in which secondary outwardly opening valve means are disposed in series with said centrifugally operated or primary valve means, said secondary valve means being urged by spring means towards the closing position and by centrifugal force towards the opening position, said spring means being so calibrated as to bring said secondary valve means into the closing position at a rotational speed lower than the opening or closing speeds of said primary valve means.

References Cited in the file of this patent

UNITED STATES PATENTS

Number	Name	Date
1,766,520	Klimek	June 24, 1930
1,873,688	Walker	Aug. 23, 1932
2,304,336	Carlson et al.	Dec. 8, 1942
2,332,593	Nutt et al.	Oct. 26, 1943
2,404,657	Roberts et al.	July 23, 1946
2,416,948	Pavlecka	Mar. 4, 1947