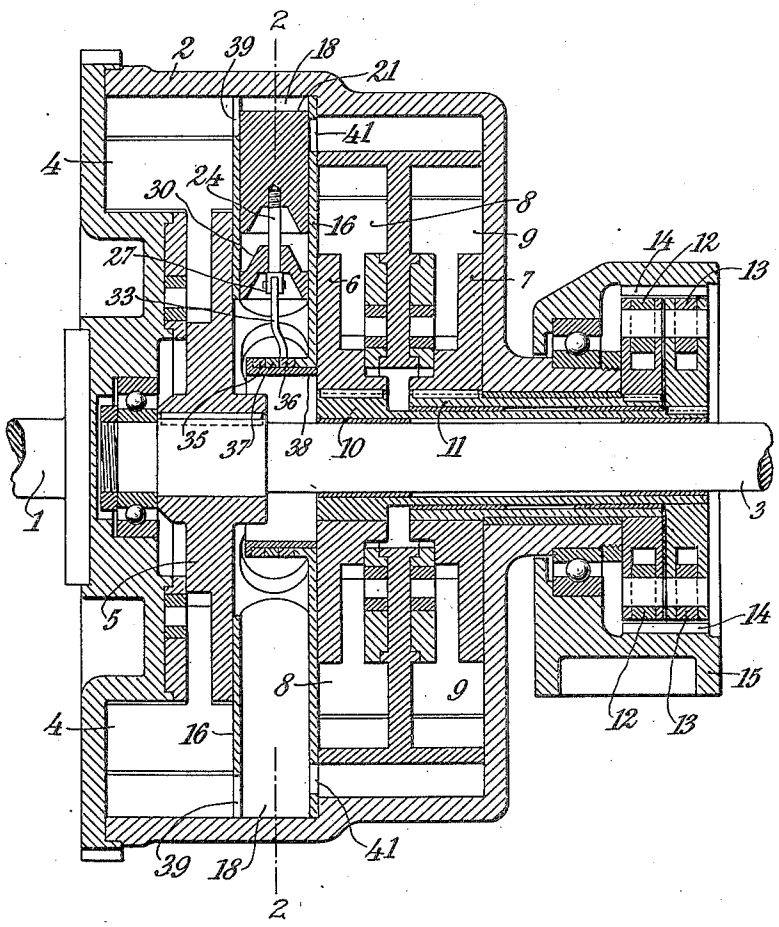
Feb. 12, 1929. 1,701,903

S. G. WINGQUIST

AUTOMATIC SPEED RATIO REGULATOR FOR HYDRAULIC COUPLINGS AND CHANGE SPEED GEARS

Filed July 17, 1922 8 Sheets-Sheet 1

Fig. 1.



INVENTOR :

Sven Gustaf Wingquist

By Attorneys,

Fraser, Dunk & Myers

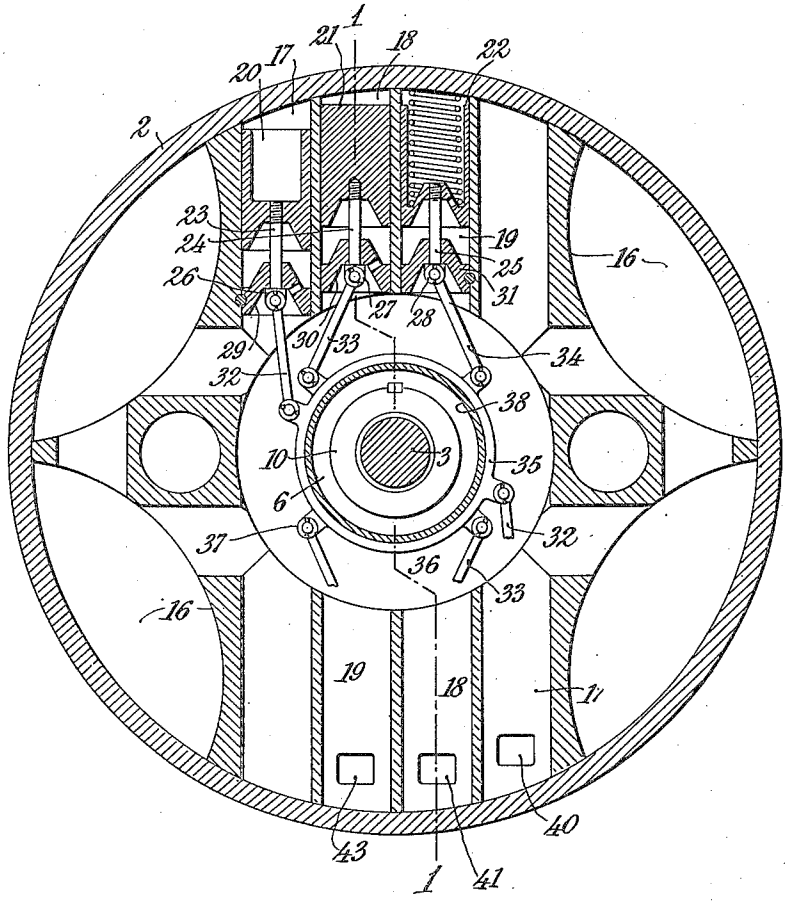
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S. G. WINGQUIST

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Filed July 17, 1922 8 Sheets-Sheet 2

Fig. 2.



INVENTOR :

Sven Gustaf Wingquist

By Attorneys,

Fraser, Dunk & Myers

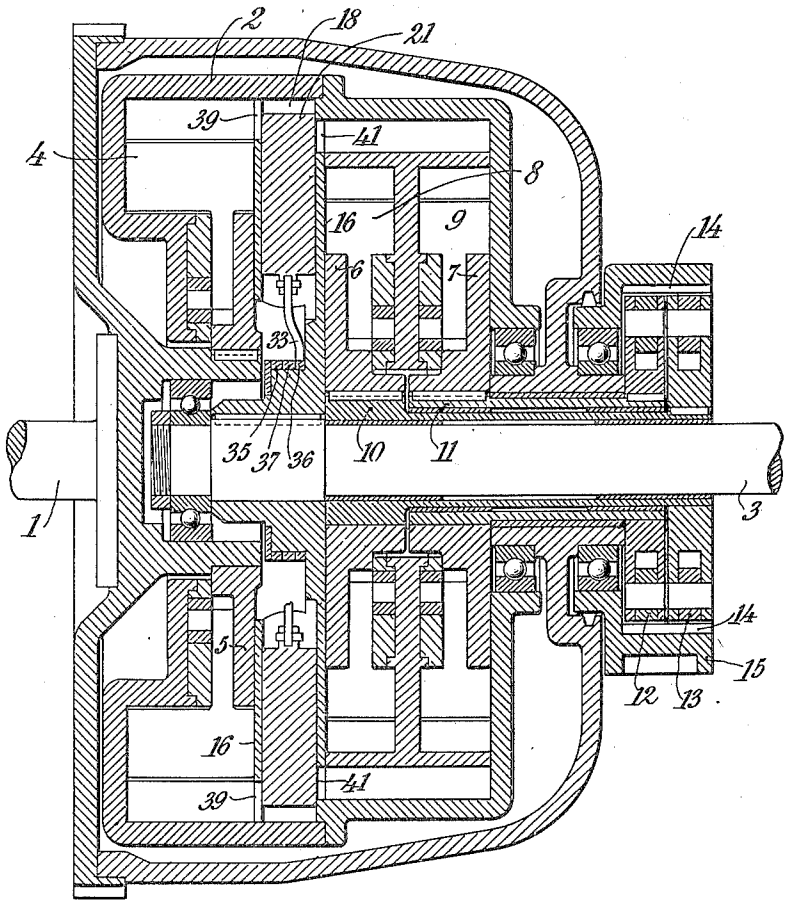
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S. G. WINGQUIST

AUTOMATIC SPEED RATIO REGULATOR FOR HYDRAULIC COUPLINGS AND CHANGE SPEED GEARS

Filed July 17, 1922 8 Sheets-Sheet 3

Fig. 3.



INVENTOR :

Sven Gustaf Wingquist

By Attorneys,

Fraser, Dunk & Myers

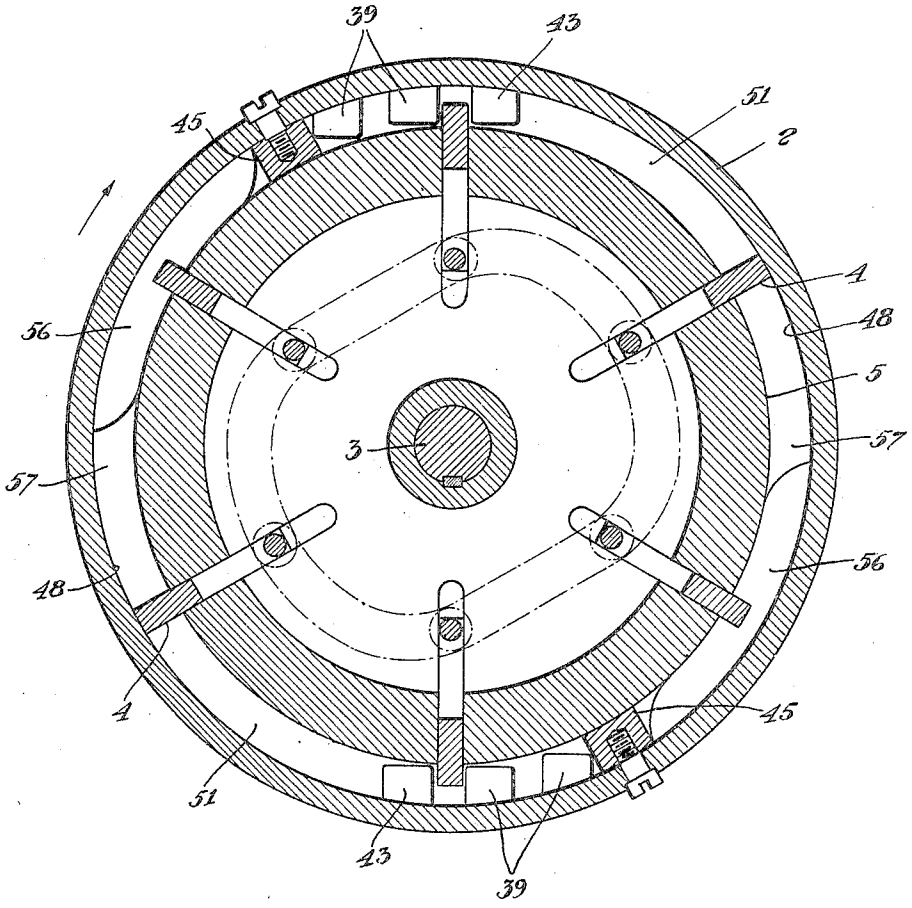
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S. G. WINGQUIST

AUTOMATIC SPEED RATIO REGULATOR FOR HYDRAULIC COUPLINGS AND CHANGE SPEED GEARS

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Fig. 4



Inventor:
Sven Gustaf Wingquist
By his Attorneys,
Fraser, Turk & Myers.

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S. G. WINGQUIST

AUTOMATIC SPEED RATIO REGULATOR FOR HYDRAULIC COUPLINGS AND CHANGE SPEED GEARS

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Fig. 5

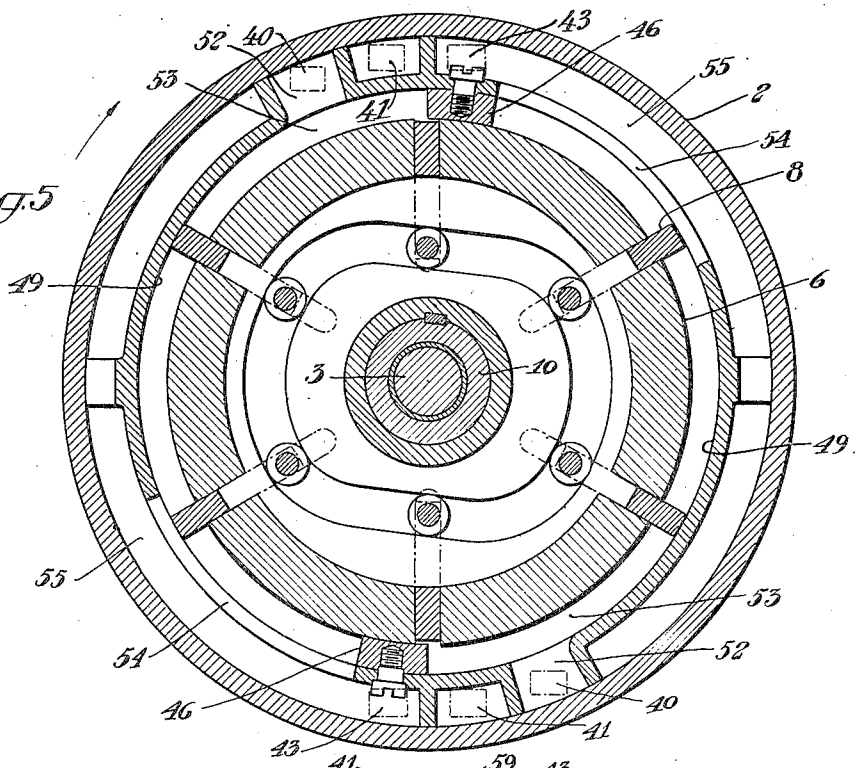
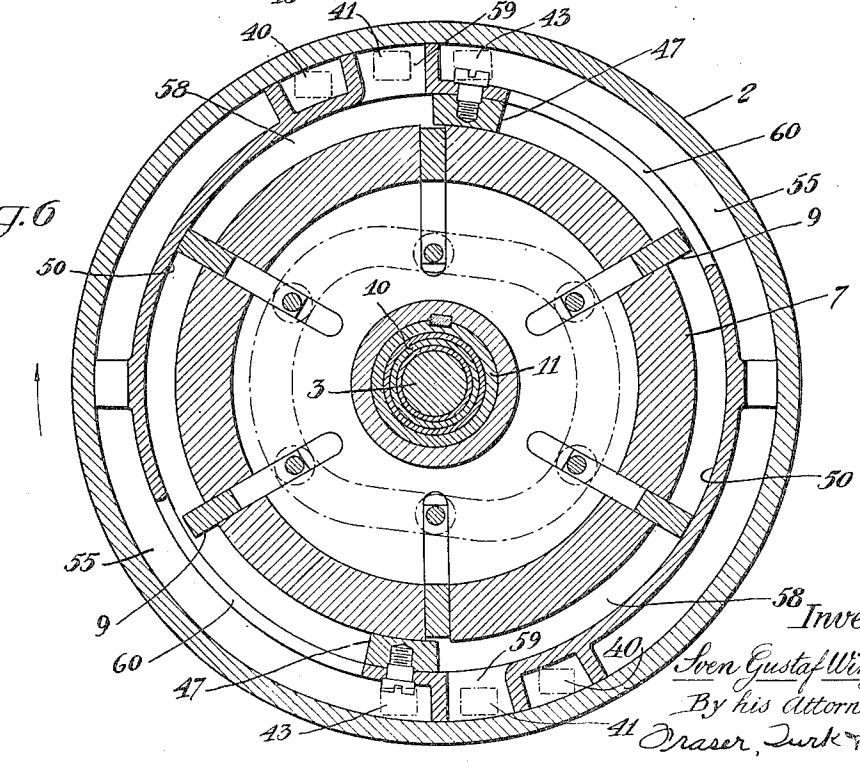


Fig. 6



Inventor:
Sven Gustaf Wingquist
By his Attorneys,
Fraser, Clerk & Myers

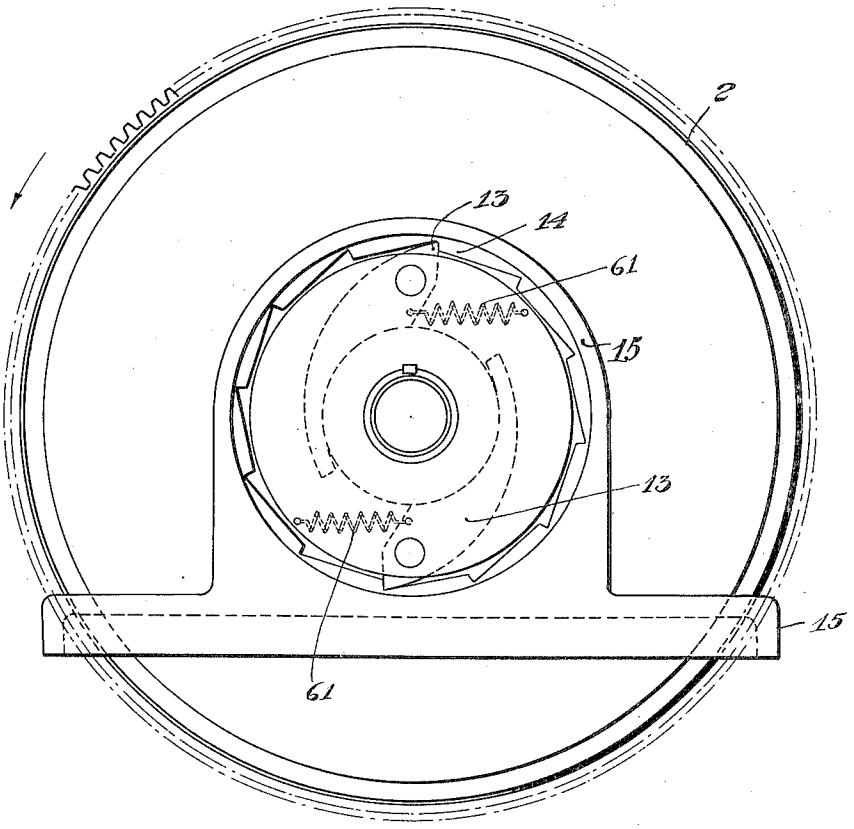
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S. G. WINGQUIST

AUTOMATIC SPEED RATIO REGULATOR FOR HYDRAULIC COUPLINGS AND CHANGE SPEED GEARS

Filed July 17, 1922 8 Sheets-Sheet 6

Fig. 7



Inventor:
Sven Gustaf Wingquist,
By his Attorneys,
Fraser, Turk & Myers.

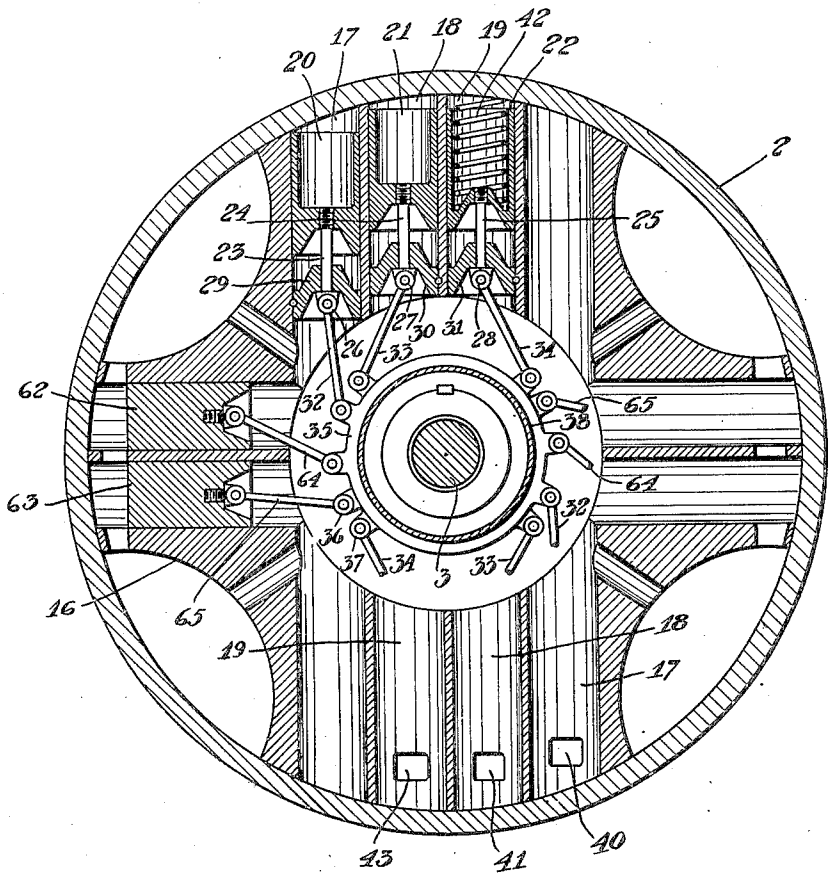
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S. G. WINGQUIST

AUTOMATIC SPEED RATIO REGULATOR FOR HYDRAULIC COUPLINGS AND CHANGE SPEED GEARS

Filed July 17, 1922 8 Sheets-Sheet 7

Fig. 8



Inventor:
Sven Gustaf Wingquist,
By his Attorneys,
Fraser, Turk & Myers.

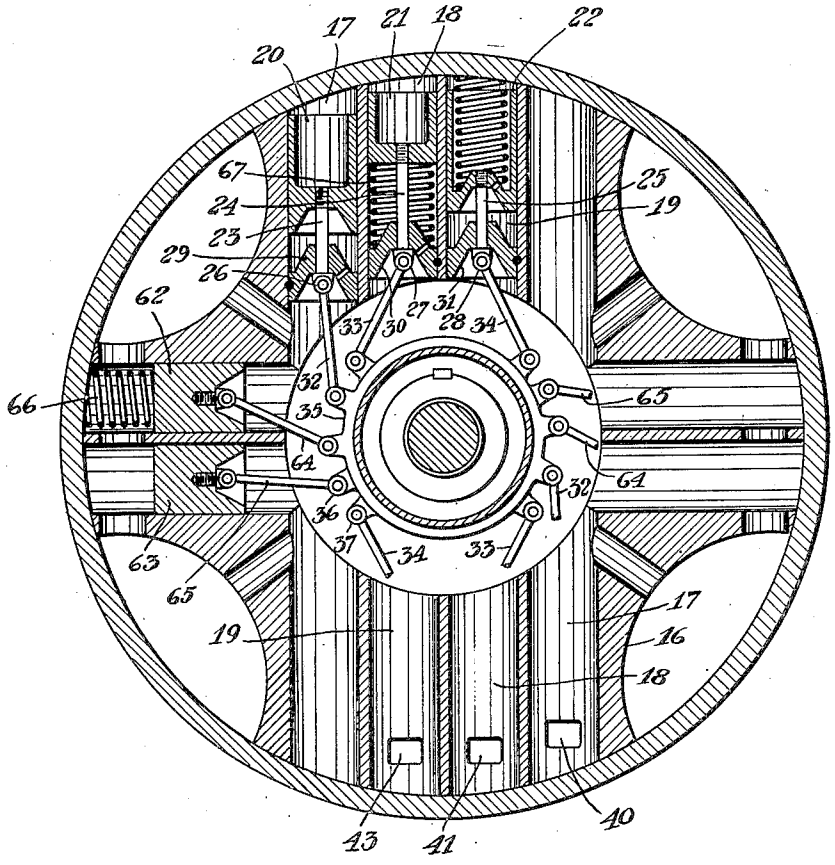
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S. G. WINGQUIST

AUTOMATIC SPEED RATIO REGULATOR FOR HYDRAULIC COUPLINGS AND CHANGE SPEED GEARS

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Fig. 9



Inventor:
Sven Gustaf Wingquist
By his Attorneys,
Fraser, Durk & Myers

Patented Feb. 12, 1929. 1,701,903

UNITED STATES PATENT OFFICE.

SVEN GUSTAF WINGQUIST, OF GOTTENBORG, SWEDEN.

AUTOMATIC SPEED-RATIO REGULATOR FOR HYDRAULIC COUPLINGS AND CHANGE-SPEED GEARS.

Application filed July 17, 1922, Serial No. 575,673, and in Sweden November 16, 1921.

This invention relates to automatic means for controlling the speed ratio of hydraulic couplings and change speed gears.

The chief object of the invention is to automatically effect a change of transmission ratio in such change speed gears so as to always maintain the most suitable coupling.

The invention consists, primarily, in the speed ratio regulating means for hydraulic change speed gears being constructed and arranged in such manner that its controlling movements will be effected under the combined influence both of the fluid pressure in the pressure chamber of the device and of the centrifugal force acting upon a rotating member, so that the transmission ratios will be dependent on and changed with both the turning moment of the driven member of the device (which moment is measured by the fluid pressure in the system) and the number of revolutions per minute of the driving member, or the driven member, or both said members.

A further object of the invention is to provide in a hydraulic coupling device one or more valves sensitive to the opposing actions of centrifugal force and fluid pressure, disposed so as to control the flow of fluid through a pump which is connected between driving and driven elements, the position of said valve or valves thereby determining the relative motion between said elements.

Further objects of the invention will be apparent from the detailed description of my invention with reference to the accompanying drawings.

In the drawings, several forms of a hydraulic coupling and change speed gear constructed according to the differential principle and specially adapted for automobiles or the like are illustrated as examples embodying the invention. Fig. 1 is a section of the one form of embodiment on the broken line 1—1 of Fig. 2. Fig. 2 is a section on line 2—2 of Fig. 1. Fig. 3 is an axial section of another form of embodiment. Fig. 4 is a transverse sectional view of the delivery pump viewed from the left-hand side of Fig. 1; Fig. 5 is a similar view of the first receiving pump to the left of Fig. 1, and Fig. 6 a similar view of the second receiving pump to the right of the first receiving pump. Fig. 7 is an end view of the change speed gear viewed from the right hand side of Fig. 1 and showing the ratchet device for checking the stator elements; Fig. 8 is a view similar to that of Fig. 2, showing a modification of the change speed gear wherein the control valves are actuated indirectly by centrifugal force; and Fig. 9 is a similar view showing a further modification wherein one of the control valves is actuated indirectly by centrifugal force and a spring and another of the control valves is actuated directly by a spring and indirectly by centrifugal force.

In the embodiment shown in Figs. 1 and 2, in which the pump elements are formed as sliding vane pumps, 1 is a driving shaft, such as the motor shaft of an automobile. Firmly connected to the said shaft is an outer casing 2 forming the driving rotor which is provided with abutments 45, 46 and 47 and working surfaces 48, 49 and 50, respectively, for the different pump elements. Journaled in the said casing 2 is a driven shaft 3, such as the shaft of an automobile, and keyed to the said shaft is a rotor 5 having radially movable vanes 4, said rotor 5 forming the driven rotor constituting, in conjunction with the driving rotor 2, the pump system normally operating as a delivery pump. Placed in the casing 2 are, further, two mutually independent stator pump elements 6 and 7 having radially movable vanes 8, 9, respectively, said stator elements being keyed to sleeves 10, 11, respectively and adapted to be turned relatively to each other and to the driven shaft 3. The stator elements 6, 7 with their vanes 8, 9 form, together with the driving rotor 2, the pump system normally operating as receiving pumps to receive the output of the delivery pump 2, 5. All the pump systems 2, 5, 2, 6 and 2, 7 are considered to be double acting, i. e. provided each with two diametrically disposed abutments and two working surfaces so that the device as a whole will be mechanically balanced. Attached to the free ends of the sleeves 10, 11 are pawls 12 and 13 cooperating with ratchet teeth 14 in a frame 15 so that the stator elements 6 and 7 are free to rotate with and relative to the driving rotor 2 when rotated in the direction of the arrow but are prevented from turning in the opposite direction. The pawls are actuated by springs 61 tending to retain the pawls in the position shown in Fig. 7 and counteracted by centrifugal force exerted by the pawls when rotating with the stators 6 and 7 as is the case in direct drive, when both receiving pumps are cut out or when one of said pumps is cut

out. By such means the pawls 12 and 13 will, while rotating and due to centrifugal force, be held out of contact with the teeth 14 of the frame 15.

Placed between the pump system 2, 5 at one side and the pump systems 2, 6 and 2, 7 at the other side is a partition wall 16 firmly connected to the casing 2, said wall carrying regulating devices arranged and operating as hereinafter described. On account of the arrangement of each pump element with two abutments, the regulating devices are arranged in pairs at diametrically opposite points, one regulating device of each pair only being shown in section in the figures, for the sake of simplicity.

Lying in a plane perpendicular to the axis of the transmission shafts and from substantially diametrically opposite points on the periphery of the partition 16, are parallel bores 17, 18, 19 forming cylindrical guides for the reciprocating piston valves 20, 21, 22. All the valves are shown in their outer positions in which the valve stem heads 26, 27, 28 placed at the inner ends of the valve stems 23, 24, 25 bear on the inner side of sleeves 29, 30, 31 which sleeves are rigidly attached to the bores 17, 18, 19 so as to prevent further outward movement of the valves. Pivotally connected to the valve stem heads 26, 27, 28 are connecting rods 32, 33, 34 pivoted to rings 35, 36, 37 respectively, which rings are adapted to turn independently on a drum 38 carried by the partition 16. By such common mounting of corresponding diametrically disposed valves a like movement of the valves belonging to each pair is secured. The valve chambers or bores 17, 18 and 19 are permanently connected at their outer ends, through the openings 39 in the partition 16 (Fig. 1), to the pressure chambers of the pump system 2, 5, so that the valves 20, 21 are directly actuated by the fluid pressure in the said chamber. In a certain predetermined position of the valves 20, the valve chambers 17 will be connected through openings 40 (shown in Figs. 2, 5 and 6) to the pressure chamber of the pump system 2, 6 so that a connection will be established between the pressure chambers of the pump systems 2, 5 and 2, 6. In a similar manner, through the valve chambers 18 and openings 41 in the partition 16, the pressure chambers of the pump systems 2, 5 and 2, 7 can be connected. The valves 20 and 21 actuated, during the operation of the change speed gear, by the fluid pressure in the pump system 2, 5 will, further, be thrown outward by the centrifugal force, and the connections between the pressure chambers of the pump systems hereinbefore described will, obviously, be established when the fluid pressure in the pump system 2, 5, and correspondingly the turning moment on the driven shaft 3, is greater than the centrifugal force exerted by the valve pistons 20, 21, said centrifugal force being dependent on the weight of the valve pistons and the number of revolutions of the rotor 2 and the shaft 1. As shown in Fig. 2, the valve piston 20 is hollow while the valve piston 21 is solid, and, on account thereof, the lighter valve 20 will connect the pressure chambers of the pump systems 2, 5 and 2, 6 at a less pressure than that at which the heavier valve 21 connects the pressure chambers of the pump systems 2, 5 and 2, 7. The by-pass valves 22 actuated by centrifugal force and by springs 42 in opposite directions establish, when the change speed gear is at rest and up to a certain low number of revolutions, a direct connection through openings 39, 43 between the pressure chamber of the pump system 2, 5 and the suction chamber common to all the pump systems so as to provide a condition of free engine in which the fluid idly short-circuits from the pressure to the suction side of the pumps.

As hereinbefore stated, the described coupling and change speed gear is specially adapted for automobiles, or the like, and in the following description of the mode of operation of the device, it will be assumed that the valves 22 commence to move outwardly and thereby to gradually shut off the connection between the pressure chambers of the pump system 2, 5 and the suction chambers, when the number of revolutions approach 300 per minute, at which speed the engine commences to develop useful power. The valves 21 are weighted and dimensioned so that, at a number of revolutions of, say, 900 per minute, or more, and at a maximum turning moment of the driving motor, their centrifugal action exceeds the fluid pressure then existing in the pump system 2, 5 so that the said valves begin to gradually shut off the connection between the pressure chambers of the pump systems 2, 5 and 2, 7. These valves will close gradually because of the fact that as they are progressively closed, the ports 41 which they control are gradually restricted, this resulting in an increase in fluid pressure, which in turn increases the resistance to further closing of the valve. Such increased resistance can only be overcome by a further increase in speed of the driving rotor. After valves 21 are fully closed and the speed of the driving element further increased, valves 20 commence to close, the outward movement of these valves in like manner gradually restricting ports 40 until when such ports are fully closed, a direct drive is established, the fluid being then locked in the pressure chambers of the delivery pump 2, 5. As hereinbefore set forth, the relative timing of the valves is determined by suitably proportioning their masses with respect to the effective pressure area of the valves. If the pressure areas are equal, as shown in Fig. 2, the valves being of the same diameter, the action of the

valves is timed by providing a different mass for each, the valves 21 which are heavier being the first to close.

Supposing the direction of running of the driving rotor 2 to be that indicated by the arrow in Figs. 4, 5 and 6 and supposing further the openings 40 and 41, Fig. 2 (also indicated for the sake of clearness by dashed and dotted lines in Figs. 5 and 6) to be shut by the valves 20 and 21, a fluid pressure will arise in the chamber 51, Fig. 4, between the abutments 45 and the vanes 4, and the rotor 5 will be caused to rotate with the same speed as does the driving rotor 2, it being assumed that the valves 22 shut off the connections 43 between the pressure chambers and the suction chambers of the delivery pump 2, 5. The pressure fluid in the chambers 51 enters through the openings 39 to the valve chambers 17 and 18, Fig. 2, at the outer ends of the control valves 20 and 21, which will, thus, be exposed to the said fluid pressure.

If then the load to be driven increases, also the fluid pressure will increase and at a certain relationship between said pressure and the centrifugal force exerted for instance by the valves 20, said valves will be moved inward and open the ports 40, Figs. 2 and 5, leading to channels 52, Fig. 5, which are open to the chambers 53 between the one side of the abutments 46 and the vanes 8 of the first receiving pump 2, 6. Due to the pawl and ratchet device 13, 14, Figs. 1 and 7, the stator 6 with its vanes 8 cannot turn in a direction opposite to that of the driving rotor 2 and, consequently, a fluid pressure will arise in the said chambers 53 causing a reaction force on the abutments 46, which will be transmitted through the rotor body 2 to the abutments 45 of the delivery pump 2, 5 and from there through the fluid to the driven rotor 5. Thus, the torque of said latter rotor 5 will increase correspondingly and at the same time the rotor 5 will slip with relation to the driving rotor 2 or obtain a decreased speed of rotation, which is dependent on the relation between the volumetric capacities of the delivery pump 2, 5 and the receiving pump 2, 6. The fluid delivered to the receiving pump 2, 6 is withdrawn through the openings 54 at the opposite sides of the abutments 46 and led to the annular suction chambers 55, which communicate through openings 56, Fig. 4 with the suction chambers 57 of the delivery pump 2, 5. Thus, a circulation of fluid takes place between the delivery pump 2, 5 and the receiving pump 2, 6.

Further, if the load to be driven be still increased, the speed of the driving rotor 2 remaining constant, also the fluid pressure increases and the valves 21 are caused to open the ports 41, Figs. 1, 2 and 6, and establish a communication between the pressure chambers 51 of the delivery pump 2, 5 and the pressure chambers 58 of the second receiving pump 2, 7, Fig. 6, through the openings 39, Fig. 4, the valve chambers 18 and the ports 41, Fig. 2, and the channels 59, Fig. 6, open to said latter pressure chambers 58. As described with relation to the coupling in of the pump 2, 6, a fluid pressure will now arise also in the chambers 58 and cause a reaction force on the abutments 47, which is, as before, transmitted to the driven rotor 5, the slip of which relatively to the driving rotor 2 will be further increased at the same time as its torque will increase correspondingly. The fluid delivered to the receiving pump 2, 7 is returned through the openings 60 at the opposite sides of the abutments 47, the annular chambers 55 and the openings 56, Fig. 4, to the suction chambers 57 of the delivery pump 2, 5.

It will be understood that the transition between successive stages of coupling, as hereinbefore set forth, occurs without shock inasmuch as the movement of the valves is smooth and gradual. During the periods that the valves are in motion a slipping clutch effect is an inherent characteristic of the transmission, and this is true of each of the three sets of valves. In shifting from neutral through the several stages to direct drive, it will be seen that the gradual closure of valves 22 finally results in a positive fluid drive at the ratio determined by the relative volumetric capacities of the cooperating pumps. Thereafter the gradual closing of valves 21 results in a smooth transition from the first speed ratio to the second speed ratio which is determined by the relative capacities of pumps 2, 5,—2, 7. The same action will occur during the closure of valves 20 during which the transmission ratio is shifted from second speed to direct drive. During the aforesaid transition periods, until the particular pair of valves which may be operative under given conditions of fluid pressure and centrifugal force, has opened sufficiently to pass fluid substantially without any obstruction in quantities sufficient to meet the demands of the receiving pump or pumps which at that time may be cooperating with the delivery pump 2, 5, a slipping clutch action is taking place. The resistance which the valves in any partially closed position offer to the flow of fluid, results in a direct application of torque between the driving rotor 2 and the driven rotor 5, because of the fluid pressure developed from the partial closure of such valves. This slipping clutch action will decrease until when a particular set of valves is closed, a positive coupling is provided at the next higher transmission ratio.

It will be understood that the action of valve springs 42 is so proportioned with respect to the centrifugal mass of by-pass valves 22 that said valves will occupy a wide open position (a radially inward position) whenever the speed of the driving rotor is re-

duced below 300 R. P. M., for example, or such other speed as may be found satisfactory for the idling speed of the engine when the transmission is in neutral. With the said valves fully open, no appreciable pressure will be developed in the chambers of the delivery pump 2, 5, and the transmission will be set in neutral. This automatic declutching action positively insures against stalling the engine in a motor vehicle for example, the clutch always disengaging before the engine is slowed down to a stalling speed.

In accelerating, the operation is just the reverse of that above set forth. The coupling starts in neutral, valves 22 gradually closing and thus progressively reducing the slip between the driving and driven elements. For a certain period after the closure of valve 22, valves 17 and 18 remain open, permitting both the receiving pumps to coact with the delivery pump to establish the first speed driving ratio determined by the relative capacity of the pumps. Upon a further increase in speed, or decrease in fluid pressure, the valves 21 will gradually close, thus cutting out the receiving pump 2, 7, and thereby gradually establishing the second speed driving ratio. Upon a still further increase in speed or decrease in load the valves 20 will commence to close, thereby by gradual stages shifting from second speed to direct drive. From the foregoing it will be clear that each pair of valves during successive transition periods under conditions resulting in changes in driving ratio toward direct drive virtually provides a smooth and gradually increasing clutching effect for the transmission stage of next higher ratio, the reverse operation occurring in shifting from direct drive through the several stages to neutral.

It will be clear from the aforesaid that the transmission ratio will be entirely automatically regulated in accordance with the actual circumstances, being controlled both by the load on the driven shaft and the R. P. M. of the driving shaft. The operator is thus required merely to control the fuel supply to the motor according to the speed at which he desires to travel.

The form of embodiment shown in Fig. 3 differs from that previously described in that the driving shaft 1 is connected directly to the rotor 5, which thus becomes the driving instead of the driven rotor, and the driven shaft 3 is firmly connected, through the partition wall 16, to the casing 2 which will thus form the driven rotor instead of the driving one. The valves (only 21 visible in Fig. 3) also here are placed in the partition wall 16 of the casing 2 and actuated, as before, by the pressure in the pump system 2, 5, but their action will now be dependent on the number of revolutions of the driven member (casing 2 with shaft 3) instead of by the number of revolutions of the driving member (casing 2 with shaft 1).

The modification shown in Fig. 8 differs from that shown in Fig. 2 in that the valves 20 and 21 are actuated indirectly by centrifugal force by means of radially slidable weights 62 and 63 connected by means of rods 64, 65 with the turnable rings 35 and 36, respectively, to which the valves 20 and 21 are connected in a similar manner, two such weights being provided for each pair of valves 20 and 21, respectively (only one for each pair of valves being shown in Fig. 8).

The modification shown in Fig. 9 differs from that shown in Fig. 8 in that the valves 20 are, in addition to the other actuating means, actuated indirectly by springs 66 counteracting the centrifugal force of the weights 62 and, thus, assisting the fluid pressure indirectly, while the valves 21 are actuated directly by springs 67 counteracting the fluid pressure and assisting the centrifugal force of the weights 63 indirectly.

It is obvious that the conditions may be varied by suitably changing the valve systems which may be actuated, if desired, by springs or the like supporting or counteracting the centrifugal force, or the fluid pressure, respectively. Moreover, the valve systems need not be directly operated by the centrifugal force and the fluid pressure, as shown, but, if desired, the said powers may act on the valve systems through suitable translations.

Also in other respects, the invention is not limited to the embodiments of hydraulic coupling and change speed gears shown in the drawings but may be employed even in cases where there are two pump systems only, or more than three.

I claim:

1. A hydraulic coupling and change speed gear, comprising a delivery pump and at least one receiving pump, and valve controlled passages between said delivery pump and said receiving pump, means operated by the fluid pressure in said delivery pump to open said passages, and means operated by the centrifugal force to close said passages.

2. A hydraulic coupling and change speed gear comprising a delivery pump and at least one receiving pump, and valve controlled passages between said delivery pump and said receiving pump, means adapted to control said passages, said means adapted to be actuated by a force resulting from the combination of fluid pressure and centrifugal force, so as to open and close said passages respectively, in accordance with the fluid pressure set up in the delivery pump in overcoming the load to be driven and in accordance with the speed of one of the rotating members of the change speed gear.

3. A hydraulic coupling and change speed

gear, comprising a delivery pump and at least one receiving pump, and valve controlled passages between said delivery pump and said receiving pump, means adapted to control said passages, said means being movably carried by a rotating member of the change speed gear and adapted to be moved in a direction to open said passages by the fluid pressure set up in said delivery pump in overcoming the load to be driven and being adapted to be moved in a direction to close said passages by the centrifugal force due to the rotation of said rotating member, whereby the position of said control means will be determined by the resultant force of the aforesaid opposed forces.

4. A hydraulic coupling and change speed gear, comprising a delivery pump and a plurality of receiving pumps, and valve controlled passages between said delivery pump and said receiving pumps, a plurality of means adapted to control said passages, said means being movably carried by a rotating member of the change speed gear and adapted to be moved in a direction to close said passages by centrifugal force due to the rotation of said member and to be moved in a direction to open said passages by fluid pressure in the delivery pump due to overcoming the transmission load and each of said means having a different ratio between its susceptibility to centrifugal actuating force and its active area exposed to fluid pressure whereby at increasing engine speed, the torque load remaining constant, the several control means will act in a predetermined series to vary the quantity of fluid delivered by said delivery pump.

5. A hydraulic coupling and change speed gear, comprising a delivery pump and a plurality of receiving pumps, and valve controlled passages between said delivery pump and said receiving pumps, a plurality of means adapted to control said passages, said means being movably carried by a rotating member of the change speed gear and adopted to be moved in a direction to close said passages by centrifugal force due to the rotation of said member and to be moved in a direction to open said passages by fluid pressure in the delivery pump due to overcoming the transmission load and each of said means having a different ratio between its centrifugally active mass and its active fluid pressure surface whereby at increasing torque loads, the engine speed remaining constant, the several control means will act in a predetermined series to permit an increasing quantity of fluid to be delivered by said delivery pump.

6. A hydraulic coupling and change speed gear according to claim 1, including an auxiliary spring fluid control means adapted to be actuated by centrifugal force in accordance with engine speed whereby until the engine has reached a certain predetermined speed, free egress of the fluid from said pump will be afforded regardless of any load on the transmission.

7. A hydraulic coupling and change speed gear according to claim 1, in which said delivery pump and said receiving pump or pumps are formed by a rotary driving member, a rotary driven member and at least one normally stationary member, one of said rotary members being common to both the delivery pump and the receiving pump or pumps, said control means being carried by said common rotary member.

8. A hydraulic coupling and change speed gear according to claim 1, in which said delivery pump and said receiving pump or pumps are formed by a rotary driving member, a rotary driven member and at least one normally stationary member, said rotary driven member being common to both the delivery pump and the receiving pump or pumps, the means controlling the connections between said delivery and receiving pumps being carried by said common driven member.

9. A hydraulic coupling and change speed gear, comprising a delivery pump and one or more receiving pumps and communications between said delivery pump and said receiving pump or pumps, means adapted to automatically control said communications, said means comprising substantially radially movable piston valve members mounted in a partition between said delivery pump and said receiving pump or pumps and exposed at their outer ends to fluid pressure set up in the delivery pump to open said communications and having its centrifugally active mass proportioned with respect to its active fluid pressure area so as to close said communications at a predetermined speed of rotation of the member carrying said partition and at a predetermined fluid pressure set up in said delivery pump.

10. A hydraulic coupling mechanism with which to connect a driving element with a driven element, said mechanism comprising a fluid pump connected between and actuated in accordance with the difference in speed between said elements, a passage connecting the intake and exhaust sides of said pump, a valve in said passage adapted to control and dominating the circulation of fluid through said pump and thereby to control the relative movement between said elements, and an automatic valve control comprising centrifugal means sensitive to the speed of the driving element, said means tending to close said valve, and pressure sensitive means tending to open said valve, said two means acting oppositely upon the valve to cause a substantial reduction in the relative speed between the driving and driven elements upon a ma-

terial increase in the speed of the driving element and to cause a substantial increase in the relative speed between said elements upon a material increase in the fluid pressure.

11. A hydraulic coupling mechanism with which to connect a driving element with a driven element, said mechanism comprising a fluid pump connected to and actuated in accordance with the difference in speed between said elements, a passage connecting the intake and exhaust sides of said pump, and automatic valve means to control the flow of fluid through said passage solely in accordance with the resultant force of the opposing actions of centrifugal means and fluid pressure, the said valve means being sensitive in the one direction to centrifugal force due to the speed of the driving element, and in the opposite direction to the fluid pressure generated by the pump, increments in fluid pressure when the speed is constant adjusting the valve means to increase the flow of fluid through said passage, and increments in the speed of the driving element when the pressure is constant adjusting the valve means to decrease the flow of fluid through said passage.

12. A hydraulic coupling mechanism with which to connect a driving element with a driven element, said mechanism comprising a pump connected between and actuated in accordance with the difference in speed of said elements, a valve adapted to control and dominating the flow of fluid from said pump, centrifugal means controlled by the speed of the driving element tending to close said valve and pressure-sensitive means tending to open said valve, the position of the valve and consequently the flow of fluid from said pump being substantially determined by the resultant of the said opposing forces of centrifugal action and fluid pressure.

13. A hydraulic coupling mechanism with which to connect a driving element with a driven element, said mechanism comprising a fluid pump connected between and actuated in accordance with the difference in speed between said elements, and means sensitive to the opposing actions of centrifugal force and fluid pressure adapted to control and dominating the flow of fluid through said pump in accordance with the resultant of said opposing actions, the relative motion between the driving and driven elements being substantially inversely proportional to the centrifugal force under conditions of uniform pressure and being substantially proportional to the fluid pressure under uniform conditions of centrifugal action.

14. A hydraulic coupling mechanism with which to connect a driving shaft with a driven shaft, the slip of said mechanism being controlled by the flow of a fluid, said mechanism comprising a valve arranged to receive pressure from the fluid tending to open the valve, together with centrifugally-operated means tending to close the valve, said valve being adapted to control and dominating the flow of fluid.

15. A hydraulic coupling mechanism with which to connect a driving shaft and a driven shaft, comprising a casing enclosing a body of fluid, the flow of which corresponds to and controls relative rotation of said shafts, and the pressure of which varies with the load on the driven shaft, a valve adapted to control and dominating such flow of said fluid, said dominating valve being arranged and adapted to be acted upon oppositely by centrifugal force tending to close it, and by the pressure of said fluid tending to open it, the slip of the coupling varying directly with the load and inversely with the speed.

In testimony whereof I have signed my name.

SVEN GUSTAF WINGQUIST.