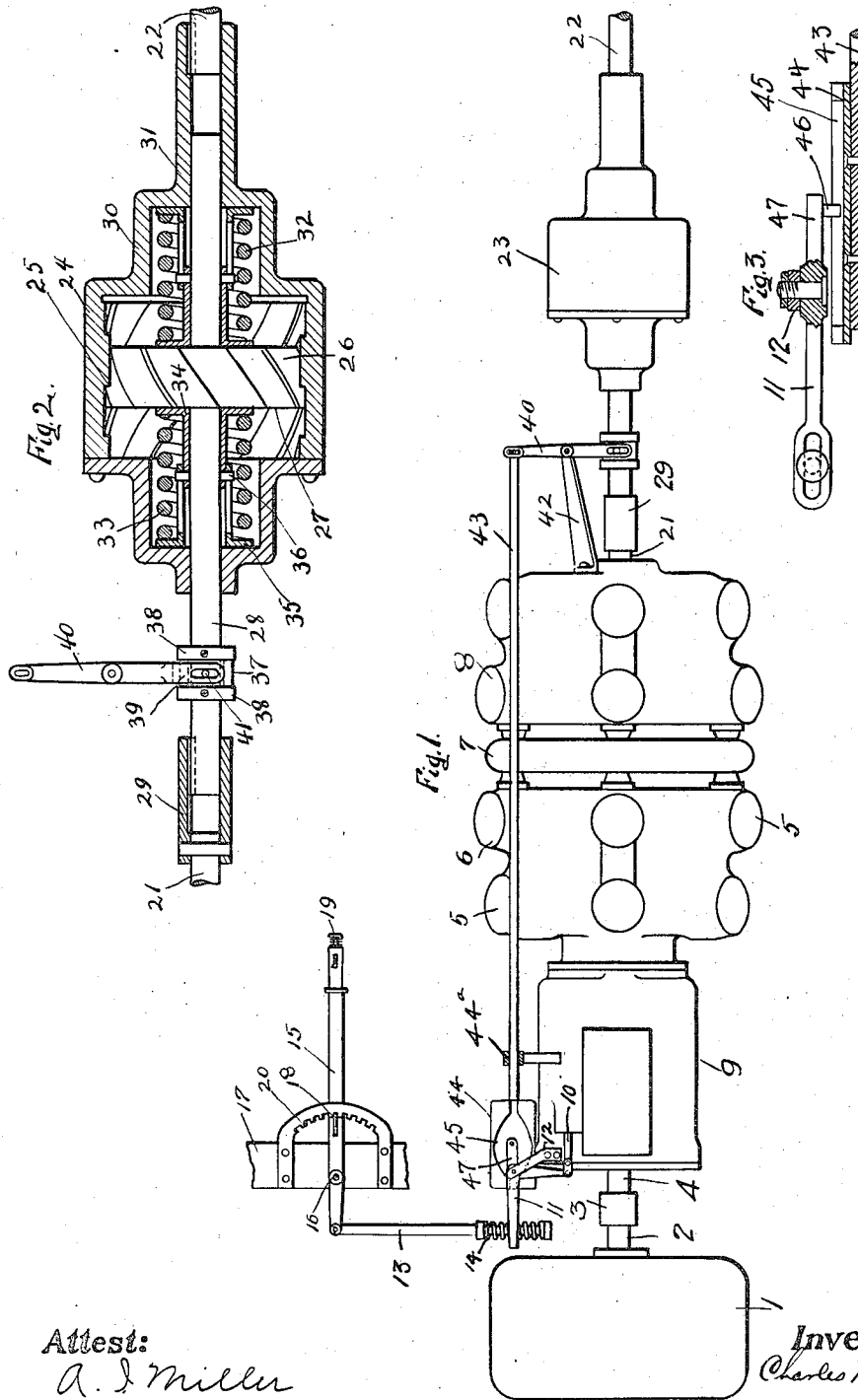


C. M. MANLY.
 CONTROL MECHANISM FOR VARIABLE SPEED GEARS.
 APPLICATION FILED OCT. 21, 1912.

1,244,878.

Patented Oct. 30, 1917.

2 SHEETS—SHEET 1.



Attest:

A. J. Miller

Notary Public

Inventor:

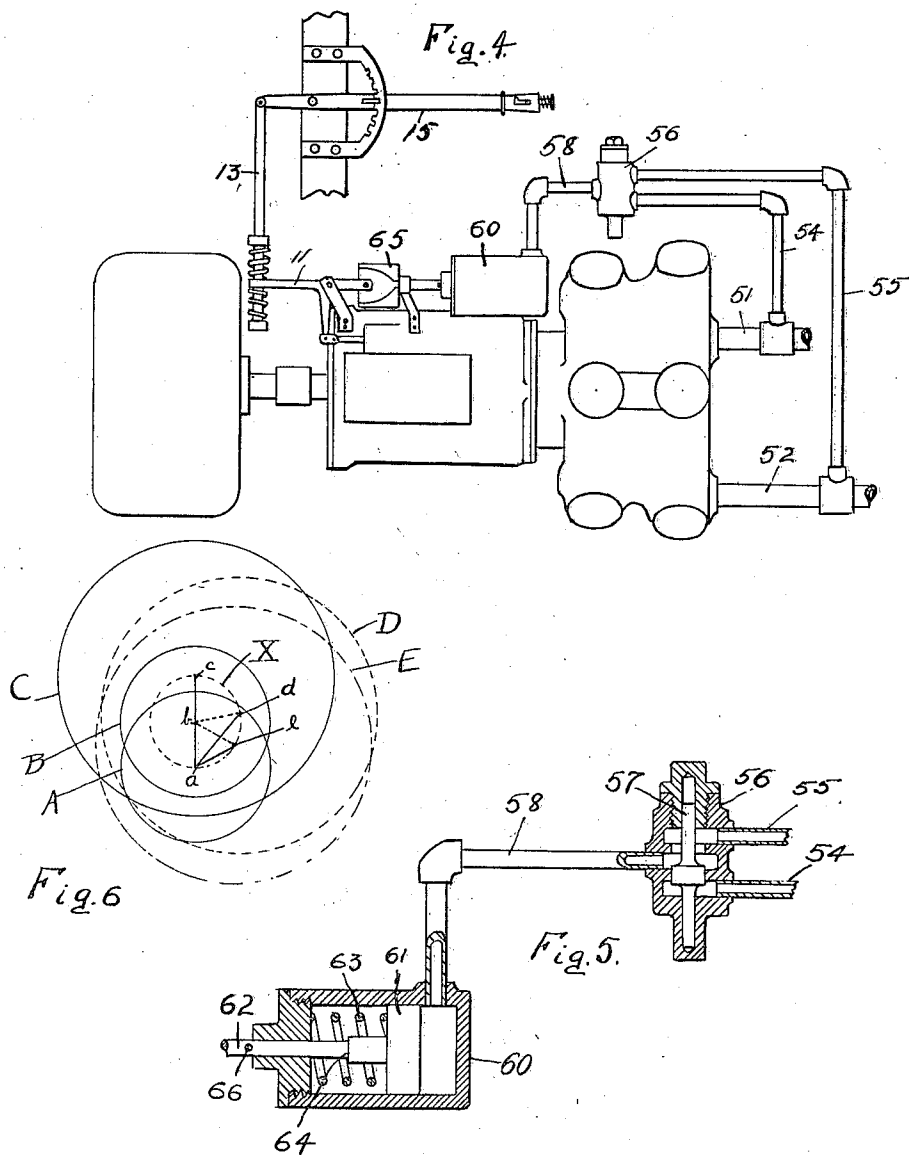
Charles M. Manly

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UNITED STATES PATENT OFFICE.

CHARLES MATTHEWS MANLY, OF FREEPORT, NEW YORK.

CONTROL MECHANISM FOR VARIABLE-SPEED GEARS.

1,244,878.

Specification of Letters Patent.

Patented Oct. 30, 1917.

Application filed October 21, 1912. Serial No. 726,994.

To all whom it may concern:

Be it known that I, CHARLES M. MANLY, a citizen of the United States, residing at Freeport, Long Island, New York, have invented a new and useful Improvement in Control Mechanism for Variable-Speed Gears, of which the following is a specification.

This invention relates to control mechanism for variable-speed gears, particularly hydraulic variable speed gears, and has for its object to provide a gear of this class with a control mechanism which will automatically maintain the speed ratio of its driving and driven parts within such limits, depending on the load of the driven member, that the gear can drive the load without requiring a greater driving power than the gear is designed to transmit.

Hydraulic variable-speed gears, because of their great flexibility, are frequently used in connection with gas engines and electric motors, and it is well known that engines and motors of this type cannot be overloaded beyond their normal rated horse power without danger of being stopped or injured. Hence when a driving mechanism of this character is employed to drive apparatus which is subjected to variable loads, such for instance as hoisting devices, ship steering gear, motor vehicles, etc., there is always danger of the apparatus being disabled because of damage to the prime mover through overloading even when ordinary care is observed in its operation.

My improved control mechanism not only removes all possibility of damage to the apparatus from overloading, but also renders the apparatus much more efficient by permitting the mechanism to be operated at all times at the highest speed that the load then on the apparatus can be moved, without exceeding the rated horse power of the prime mover. In other words, when the device is not provided with my automatic control, the operator must leave a "margin of safety" between the speed at which he runs the device and the speed possible with the rated power, whereas with my improved control mechanism the operator can set his control for full speed if he desires, because the automatic control will not permit the speed to rise beyond that possible with the rated power.

In the accompanying drawings I have shown a preferred embodiment of my invention, but it is to be understood that the

invention is in no wise limited to the details of construction there shown, but covers all such modifications thereof as fall within the scope of the appended claims.

In said drawings, Figure 1 is a plan view of a hydraulic variable-speed gear equipped with my improved automatic control mechanism.

Fig. 2 is a transverse section through the portion of the apparatus to the right of Fig. 1.

Fig. 3 is a detail view which will be hereinafter referred to.

Fig. 4 is a view generally similar to Fig. 1, but showing a modified form of the invention.

Fig. 5 is a sectional detail of a portion of Fig. 4 on an enlarged scale, and

Fig. 6 is a diagrammatic view illustrating the relation of the movement of the adjusting device to the stroke length of the pump.

Referring now to the drawings in which like reference characters indicate like parts throughout the several views; 1 represents the prime mover of the device which is here shown diagrammatically as an electric motor, which of course may be any other preferred form of engine or motor. The shaft 2 of the electric motor is connected by coupling indicated at 3, with the driving shaft 4 of the hydraulic transmission device indicated generally by the numeral 5. The hydraulic transmission device forms no part *per se* of the present invention, but may be of any preferred type in which the speed of the driven member relative to the driving member may be varied by the movement of a suitable control lever. As here shown, this device is of the general type disclosed in my prior Patent No. 801,097 of October 3, 1905, and more particularly of the type disclosed in my pending application No. 639,464 and comprises a radial cylinder pump 6 having a crank shaft of adjustable stroke connected by suitable fluid transmitting passages 7 with a radial cylinder motor 8 similar to the pump except that its crank has a fixed stroke instead of a variable stroke. It will be obvious that when the pump shaft is driven from the prime mover as described, the pump will deliver fluid to the motor in a volume depending upon the position of adjustment of the crank shaft of the pump, and as the fluid used in transmission of this character is incompressible, the speed of the motor 8 will be correspondingly varied. In

the application above referred to the variable stroke crank of the pump comprises a crank pin of fixed stroke having rotatably adjustable thereon an eccentric bushing of an eccentricity equal to that of the crank pin. In Fig. 6 I have illustrated diagrammatically the operation of the adjustable stroke crank pin. In this figure the solid line circle A represents the pump shaft with center at *a*, the circle B the crank pin with center at *b* rigid therewith, and the circle C represents the eccentric bushing at its full stroke position with the center of its outer periphery at *c* and rotatably adjustable around the center *b* of the crank pin. When the bushing is adjusted around the crank pin its center *c* moves through the circle X and the length of the crank arm varies from its maximum represented by the line *c b a* to zero, when *c* coincides with *a*, thence back to a maximum when *c* reaches its original position. The dotted circle D shows the position of the bushing when it has been adjusted through a sufficient angle for the crank arm *d a* to be three quarters the maximum crank arm, and the dot and dash line circle E shows the position of the bushing at its half stroke adjustment, as represented by the line *e a*. A fluid pressure operated adjusting mechanism for the bushing is provided, which mechanism is contained in a casing 9 and forms a part of the pump construction, and is controlled by rod 10 projecting through the wall of the casing. The stroke adjusting mechanism is so designed that when the rod is in a central neutral position as shown in Fig. 1 of the drawing, the bushing is at its zero position, or concentric with the center of rotation, and when moved from such central position in either direction, causes the eccentric to be rotated away from zero to an extent proportional to the extent of movement of the rod, and in a corresponding direction, so that the speed and direction of movement of the motor are determined by the position of the rod. For adjusting the rod it is customary to provide some lever system depending on the use to which the machine is put, and I have here shown a typical system comprising a bell-crank 11 mounted on a bracket 12 attached to the casing 9 and having one of its arms connected by an intermediate link to the rod 10. To the other arm of the bell-crank a connecting rod 13 is slidingly connected and yieldingly held against relative movement thereto by means of springs 14 surrounding the rod and engaging either side of the bell-crank arm. The other end of the connecting rod 13 is pivotally attached to a lever arm 15 fulcrumed at 16 on a fixed support indicated at 17. The springs 14 are stiff enough to transmit the motion of the lever 15 to the rod 10 without compression, and the lever 15 provided with a lock-

ing pawl 18 is controlled by button 19 on the end of the lever, the said pawl being adapted to engage the notches of a segmental rack 20 attached to the support 17 whereby the control rod 10 is locked in position to which it is adjusted by the operator.

When the apparatus is used with the customary manual control described above, the shaft 21 of the motor is connected directly to the shaft 22, representing the load to be driven, but with my improved control apparatus I separate these shafts and mount between them a double-acting spring dynamometer coupling 23 of novel construction, which will now be described. Referring particularly to Fig. 2, this coupling comprises a cylindrical casing 24 having on its inner surface a plurality of spiral threads 25 with which are engaged corresponding threads 26 on the exterior of a drum or disk 27 rigidly attached to a shaft 28 connected by a coupling 29 to the shaft 21 to be rotated therewith and to have axial movement relative thereto. The casing 24 is provided at one side of its central cylindrical portion with a reduced extension 30 terminating in a further reduced portion 31 in which the shaft 28 is rotatably mounted and in which is also fixedly mounted a shaft 22, the ends of the two shafts being spaced apart a distance sufficient to permit of a limited axial movement of the shaft 28. To the other side of the central portion of the casing 24 a cap is attached, the said cap having a reduced portion similar to the reduced portion 30 of the casing, and providing a bearing for the shaft 28. Surrounding the shaft 28 and projecting into the reduced portion 30 of the casing 24 is a stout spiral spring 32, and similarly mounted in the other side of the casing 24 is a similar spring 33, the said springs normally maintaining the drum 27 in the middle of the casing 24. As considerable load is placed upon the springs 32 and 33, it is desirable that they be placed in the casing under some initial pressure. To this end, instead of mounting the springs directly between the drum and casing, I mount on the shaft 28 at each side of the drum a flanged sleeve 34 with which a slotted flanged sleeve 35 has telescopic engagement, the two sleeves being held against separation by pins 36 working in the slots of the sleeve. In assembling the device the springs are compressed between the flanges of the sleeves and the sleeves fastened together by the pins 36 before the parts are placed in the casing 24, the length of the springs and sleeves when so assembled being exactly equal to the distance between the side face of the drum 27 and the end face of the casing 24, so that when the drum moves in one direction or the other, compressing one of the springs, it is entirely free from contact with the other spring.

Loosely mounted on the shaft 28 between the cap 33 and the coupling 29 is a sleeve 37 which is held against longitudinal movement relative to the shaft by means of collars 38 attached to the shaft. A fork 39 on the end of a lever 40 is pivotally connected with the sleeve 37 by means of a pin-and-slot connection 41, the lever 40 being fulcrumed on a bracket 42 attached to the casing of the hydraulic motor 8, as shown in Fig. 1. Pivotally connected with the other end of the lever 40 is a rod 43 slidingly mounted in a bracket 44^a attached to the casing 9 and adjacent to the bracket 12. Mounted on the end of the rod 43 is a plate 44 having a longitudinal cam groove 45 whose edges are adapted to be engaged by a pin 46 projecting downwardly from the end of a rearward extension 47 of the bell-crank lever 11. It will be noted that the groove 45 is wide enough at its middle part, which is in line with the pin 46 when neither of the springs 32, 33 is compressed to permit the control rod 10 to be adjusted to its full stroke position in either direction, and that the groove becomes narrower toward both ends so as to limit more or less the range of movement of the bell-crank.

The operation of the device will now be described. The springs 32 and 33 are placed under sufficient initial compression when the parts are assembled for them to transmit without further compression the full horse power for which the prime mover is designed at the maximum speed of the driven member. If the load on the shaft 22 does not exceed the maximum load which can be driven at full speed without exceeding the desired maximum horse power, the drum 27 will remain in the middle of the casing 24 and the widest portion of the groove 45 will be aligned with the pin 46 so that the control lever may be adjusted to its full stroke position in either direction from its central zero position. If now for any reason the load on the shaft 22 be increased beyond that for which the springs are set, one of the springs, depending on the direction of rotation of the shaft, will yield under the thrust of the drum 27 permitting the drum to move to the right or left from the position shown and with it the shaft 28 and the end of the lever 40 attached thereto. The lever 40 is connected by the rod 43 with the cam plate 44, and movement of the plate in either direction brings a narrower part of the groove into the path of movement of the pin 46, to limit the range of movement of the lever 11. It will be noted that the groove 45 does not uniformly decrease in width from the middle toward each end, but instead its sides are curved so that during the first portion of the movement of the slide from its central neutral position the range of movement of the pin 46 is but slightly

limited, the amount of limitation effected by each unit of longitudinal advance increasing as the plate reaches the limit of its movement in either direction. The reason for this will be readily understood from an inspection of Fig. 6. As stated above equal linear movements of the control rod 10 effect equal angular movements of the eccentric bushing of the variable stroke crank, but as shown in Fig. 6 equal angular movements of the bushing do not effect equal changes in the pump stroke. In this figure the angle cbd represents the extent of rotation of the bushing necessary to reduce the pump stroke from full stroke to three-quarters stroke and the angle dbe represents the adjustment necessary to reduce the stroke from three-quarters stroke to half stroke. The angle cbd is substantially twice the angle dbe and considerably greater than the angle cba which represents the adjustment necessary to reduce the stroke from half stroke to zero. The sides of the groove 45 are curved to compensate for this non-uniformity of adjustment in such manner that equal longitudinal movements of the rod 43 and plate 44 as effected by relative movement of the drum 27 and its casing cause equal changes in the length of the pump stroke. If the lever 15 is locked by its pawl 18 in its full stroke position when the overload occurs, the bell-crank 11 will be moved by the rod pin 46 which engages it against the tension of the spring 14 of the rod 13 thereby moving the control rod 10 in a direction to reduce the stroke of the pump and consequently the speed of the motor. As the speed of the motor, and consequently the speed at which the load is driven, is reduced, it will obviously require less power to drive the load and consequently by properly proportioning the strength of the springs 32, 33 to the extent of movement permitted the drum 27, the load on the driven shaft 22 always maintains the control rod 10 in a position to keep the speed of the hydraulic motor within the limits permitted by the maximum horse power of the prime mover. If the lever 15 is locked as described against movement when the overload is removed from the driven shaft 22, the compressed spring 14 will expand, moving the bell-crank 11 and consequently the control rod 10 back to the position they occupied before the overload was placed upon the shaft. When it is desired to drive the driven machine or part at the maximum speed possible without damage to the prime mover the operator can lock his lever 15 in its full speed position so that the spring 14 will keep the bell-crank lever 11 as near that position as the varying load on the shaft 22 permits.

If it is desired that the load continue to be driven at a reduced speed even after the

overload be removed, the operator can, by means of the bayonet slot lock on the button 19 lock the pawl 18 out of engagement with the rack 20 so that should the automatic control shift the rod 10 to reduce the speed of the motor, the rod will remain in that position until the speed is again adjusted by the operator.

In Figs. 4 and 5 I have illustrated a modification of the mechanism above described, which modification is especially adapted for use where the pump and motor of the hydraulic gear are separated by a distance such that the mechanical connection of the dynamometer on the motor shaft with the control lever of the pump is impractical, but it will of course be understood that the mechanism shown in these figures may if preferred be used with the close coupled type of gear shown in Fig. 1.

In Fig. 4 I have shown a pump element of the hydraulic gear, which pump may be identical with the pump of the gear above described except that instead of being designed for connection with an immediately adjacent motor it has connected to it two pipes 51, 52, adapted for connection to a remotely located hydraulic motor of any preferred construction, to form with the pump a closed fluid circuit. Connected with the pipes 51, 52 are branch pipes 54, 55 respectively which lead to opposite ends of a valve chamber 56 containing a valve 57 adapted in one position to establish communication between the pipe 54 and a pipe 58 leading from the opposite side of the cylinder, and in another position to connect the pipe 55 with the pipe 58, the position of the valve depending on which of the pipes 54, 55 is connected with the high pressure side of the fluid circuit of the hydraulic gear.

The pipe 58 from the valve chamber connects with the closed end of a cylinder 60 attached by suitable brackets to the casing of the pump. Mounted in the cylinder 60 is a piston 61 whose rod 62 carries a cam plate 65 similar to one half the plate 44 as the piston 61 operates in the same direction irrespective of the direction of rotation of the driven motor. A spring 63 is mounted on the rod 62 between the piston 61 and the cap of the cylinder, the spring having an initial tension such as to hold the piston 61 in its rearward position until the pressure in the cylinder exceeds the pressure at which the maximum horse power can be delivered with the pump at full stroke. A pin 66 is provided in the piston rod to hold the spring at its desired initial tension.

A shoulder 64 is provided on the rod of the piston 61 at a point to engage the cap of the cylinder to prevent further movement of the piston when the rod 62 has been shifted to a position to adjust the control

rod 10 to its zero position. The operation of this modification of the device will be obvious from the description given above. As stated, the motor element of the gear is of fixed stroke, and consequently the pressure of the fluid to actuate it at any given moment is an exact measure of the torque exerted to move the load on the driven shaft. Hence, the spring 63 will be affected in precisely the same manner as the springs 32 and 33 of the spring dynamometer, and the piston 61 will be shifted in the same manner as the drum 27 except that the pressure in the cylinder is through the valves 57 always delivered at the same end of the cylinder 60 irrespective of the direction of rotation of the driven shaft. For this reason the cam plate 65 is but half the length of the cam plate 44, the full stroke position of the cam plate being at the end, instead of in the middle.

Having thus described my invention, what I claim as new and desire to secure by Letters Patent is:—

1. The combination of a variable speed gear, a movable control member therefor, stops for limiting the extent of movement of said member, and means governed by the load on said speed gear for altering the position of said stops.

2. The combination of a variable speed gear, comprising a driving and a driven member, means for controlling the relative speeds of said members, means for limiting the movement of said control member, and means governed by the load on the driven member of said gear for actuating the said limiting means to prevent the speed of said driven member exceeding the speed at which said load can be driven without exceeding a predetermined power.

3. The combination of a variable speed and reverse gear, comprising a driving and a driven member, means for controlling the speed of the driven member of said gear in either direction of rotation, and means governed by the load on said driven member for varying the speed thereof inversely as the load in either direction of rotation.

4. The combination of a variable speed and reverse gear, comprising a driving and a driven member, means operatable at will for varying the speed and direction of rotation of said driven member, stops for limiting the movement of said control means, and means governed by the load on said driven member for adjusting said stops to maintain the limits to which the speed of the driven member can be adjusted in either direction, proportional to the load on said driven member.

5. The combination with a variable speed gear comprising manually operatable control mechanism for varying the speed ratio of the gear, of means operatable simultaneously

with said mechanism and governed by the load on said gear for maintaining the speed of the driven member thereof at the maximum speed at which it can drive the load without requiring more than a predetermined power.

6. The combination with a variable speed gear comprising a driving and a driven member, of means operatable at will for varying the speed ratio of said members, stops for controlling the movement of said means, means governed by the load on said driven member for adjusting said stops to maintain the limits to which said control means may be adjusted proportional to said load, and yielding means for maintaining said control means at the limit of its adjustment permitted by said stops.

7. The combination of a hydraulic variable speed gear comprising a variable capacity pump and a motor driven by fluid pressure from said pump, means operatable at will for varying the capacity of said pump, and means operatively connected with said

motor controlled by the load thereon for operating said stroke-adjusting mechanism.

8. In a fluid gear, the combination with a pump member of means for regulating the velocity and pressure of fluid pumped and reversing the direction of flow of the same, and means for maintaining a constant product of velocity and fluid pressure.

9. In a fluid gear, the combination with a pump member of means for regulating the velocity and pressure of fluid pumped and reversing the direction of flow of the same, means for maintaining a constant product of velocity and fluid pressure, and means for predetermining said constant product.

In testimony whereof, I, said CHARLES M. MANLY, have signed my name to this specification in the presence of two subscribing witnesses, this 19th day of October, 1912.

CHARLES MATTHEWS MANLY.

Witnesses:

A. I. MILLER,
W. B. MORTON.