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MULTISTAGE DUAL SHAFT HYDRAULIC PUMP OR MOTOR

Filed July 10, 1935

4 Sheets-Sheet 1

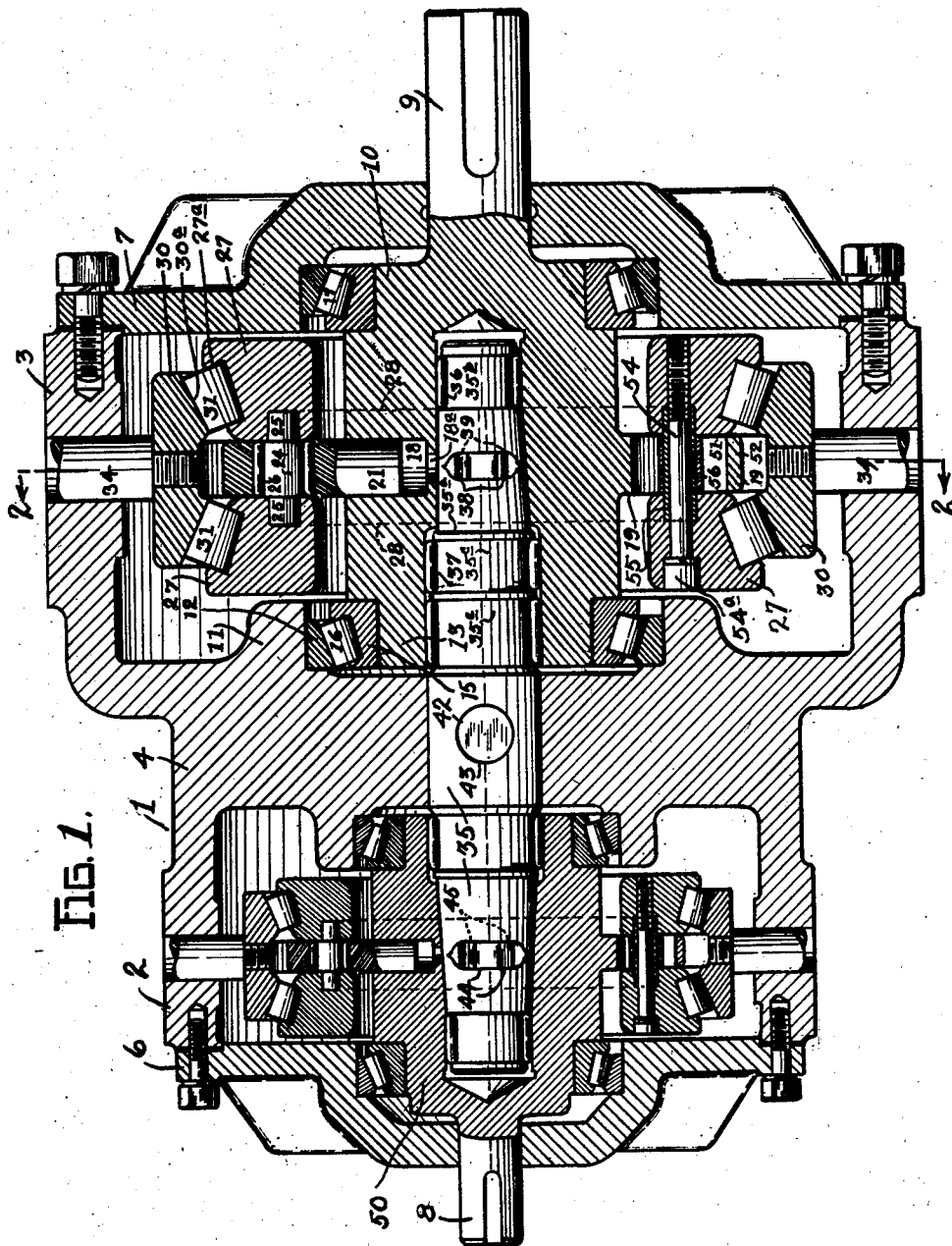


FIG. 1.

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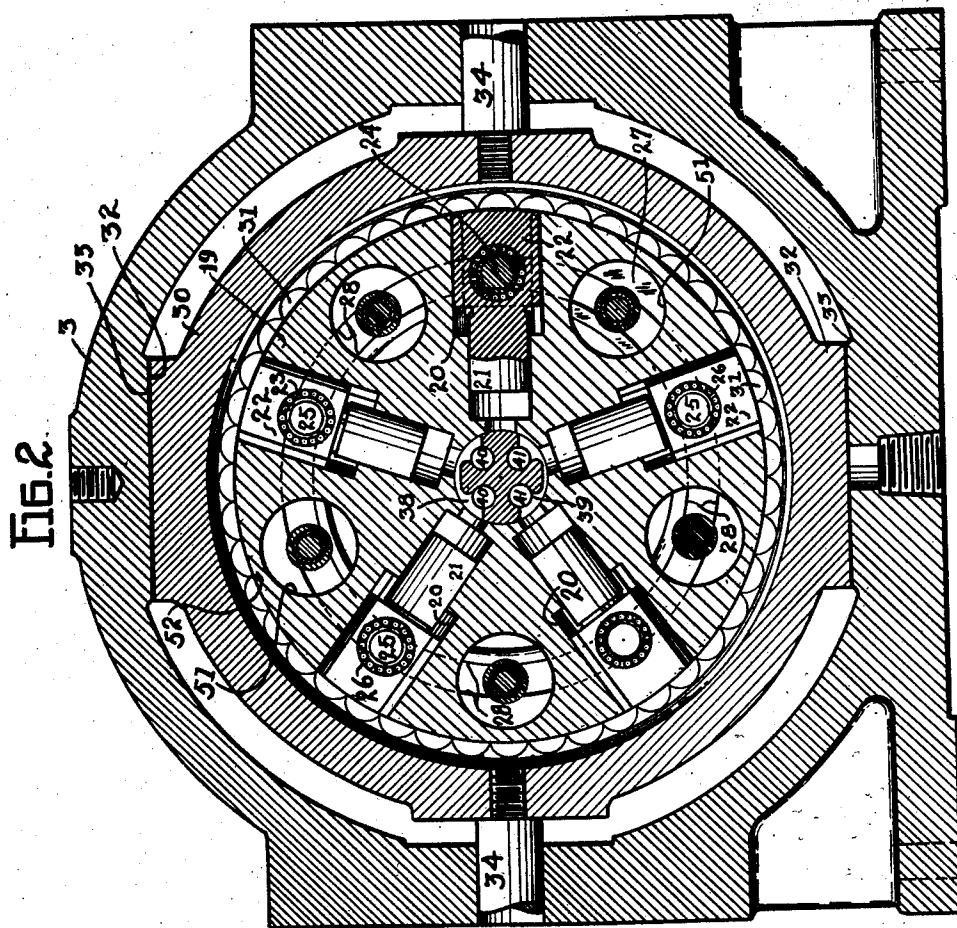
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MULTISTAGE DUAL SHAFT HYDRAULIC PUMP OR MOTOR

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4 Sheets-Sheet 2.



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MULTISTAGE DUAL SHAFT HYDRAULIC PUMP OR MOTOR

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4 Sheets-Sheet 4

FIG. 4.

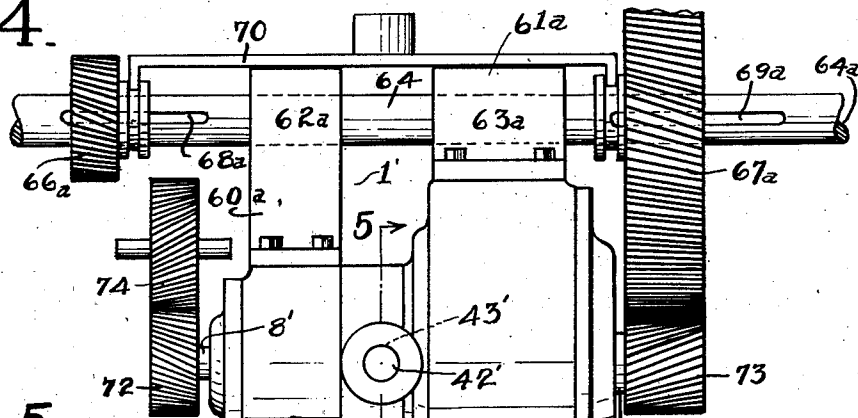


FIG. 5.

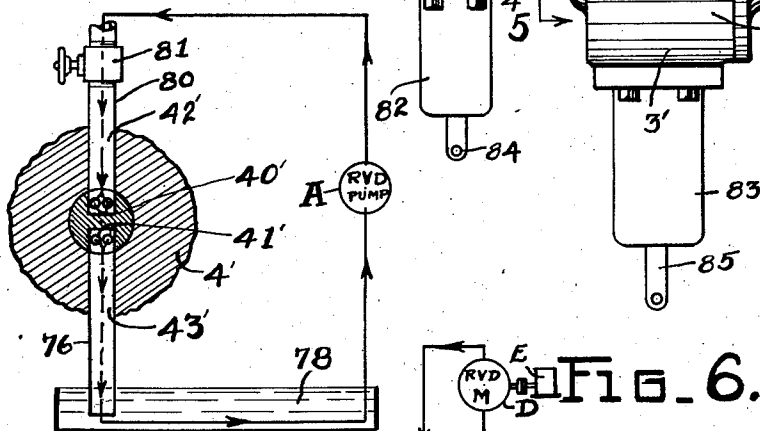
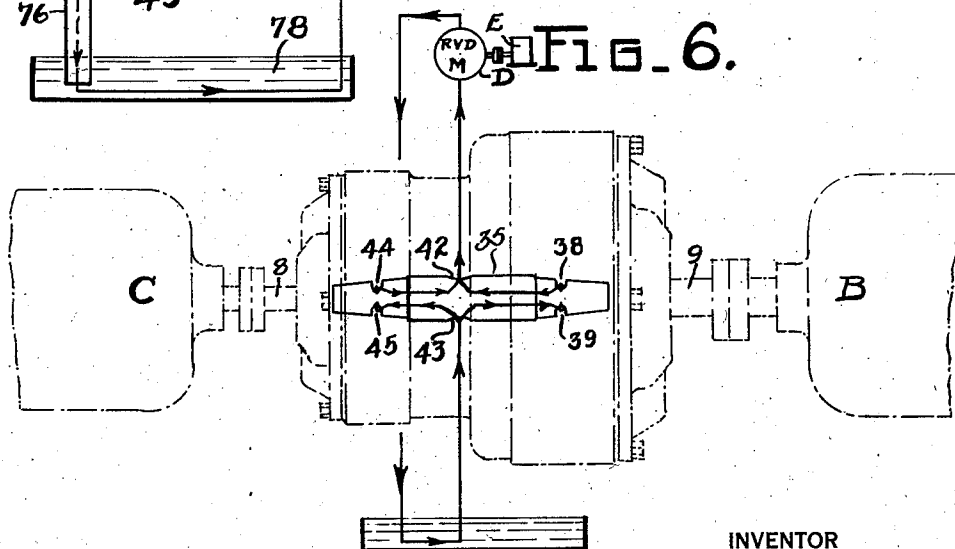


FIG. 6.



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MULTISTAGE DUAL SHAFT HYDRAULIC
PUMP OR MOTOR

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Application July 10, 1935, Serial No. 30,608

3 Claims. (Cl. 103—161)

This invention relates to hydraulic pumps and motors and particularly to a new two stage reversible hydraulic engine which may be operated as a pump for generating hydraulic fluid pressure by separate independently controlled prime movers simultaneously and as a motor for utilization of pressure fluid from a given source for operating a single driven member at different speeds or two separate driven members simultaneously or successively at greatly different and variable speeds.

One object is to provide a multistage motor for driving a given shaft or member at a wide range of speed while delivering the constant horsepower of a driving engine.

Another principal object of the present invention is to provide a multistage hydraulic machine in which each stage is reversible independently of the other when used as a motor and may be controlled independently of the other or in given as well as in any desirable variable relations thereto, and in which both units are reversible simultaneously as a pump so that the structure may be employed as a multistage pump, or as a multistage motor at high efficiency.

Another object is to hydraulically interconnect the stages of the pump for variable speed ratio of two independent prime movers in a manner assuring independent and flexible speed in each prime mover with proper power ratio input and fluid pressure equalization of the powers of the prime movers.

Another object is to provide a hydraulic engine for this purpose which is highly efficient and simple and compact in structure.

A more specific object is to provide a multistage pump driven by a high speed prime mover and a low speed prime mover simultaneously for generating pressure fluid in the common line of the stages.

Another specific object is to mechanically connect the stages of the pump for a fixed speed ratio of high and low speed prime movers, so as to insure accuracy in synchronization thereof through the pressure and proper distribution of component powers therebetween.

Another object is to provide a multistage motor in which each stage is of different capacity and provided with a separate driven shaft for low and high speed respectively.

Another specific object is to separate and dampen the torsional oscillation of the prime mover and the driven shaft respectively.

A more specific object is to provide a hydraulic motor mechanism for use in connection with a

power driven shaft by virtue of which the prime mover thereof can be operated at the speeds most suitable for highest efficiencies and economy of performance and, during the operation, the desired speeds of rotation of the shaft for given operations can be effected by gradual continuous increments throughout the range of speeds required for such operation without operatively disconnecting the shaft from the prime mover while effecting the desired changes.

Other objects and advantages will become apparent from the following specification wherein reference is made to the drawings in which

Fig. 1 is a horizontal sectional view through the axis of a pump of the present invention;

Fig. 2 is a cross sectional view taken on the lines 2—2 of Fig. 1 and 2a—2a of Fig. 3;

Fig. 3 is a horizontal sectional view similar to Fig. 1, illustrating a modification of the invention;

Fig. 4 is a horizontal top plan view illustrating the use of the present structure as a hydraulic motor combination for driving a single shaft;

Fig. 5 is a fragmentary cross sectional view taken on a plane indicated by the line 5—5 in Fig. 4; and

Fig. 6 is a diagrammatic side elevation illustrating the use of the structure of the present invention as a multistage pump.

Referring to the drawings, and particularly to Fig. 1, there is illustrated a dual stage hydraulic machine, comprising a rigid cylindrical casing 1 having a small diameter portion 2 at one end and a larger diameter portion 3 at the opposite end with a heavy rigid spider or radial wall 4 between said casing portions dividing the casing into a large diameter compartment and a small diameter compartment respectively. The ends of the casing portions 2 and 3 are closed by rigid supporting end covers 6 and 7 respectively through which project the impeller shafts 8 and 9 respectively of the dual hydraulic engine.

For brevity in illustration, the structural details will be described as a pump, the uses as a motor being referred to generally.

Since the pump stage units are the same in structure, differing only in dimensions, the pump unit included in the housing portion 3 only will be described in detail. With reference to Fig. 1, within the casing portion 3 is a rigid forged cylinder barrel 10 which is mounted at its ends in combination antifriction radial load and thrust bearings for rotation about its longitudinal axis, the impeller shaft 9 preferably being formed integral with the barrel 10. For supporting the

barrel within the casing portion 3, the wall 4 therein is provided with an axially extending annular flange 11 within which is mounted the outer race 12 of a set of tapered roller bearings or their equivalent. The barrel 10, in turn, is provided adjacent the wall 4 with an annular reduced diameter portion 13 on which is mounted the inner race 15 of the roller bearings. Between the respective races are interposed the tapered rollers 16 arranged to resist both radial load and axial thrust, the races, in turn, engaging the radial portion of the wall 4 and the radially extending end wall of the barrel 10 adjacent the hub 13 for resisting axial displacement of the races. Correspondingly, the opposite end of the barrel is mounted in tapered rollers 17, the outer race thereof being mounted in the end plate 7 in a manner similar to the race 12 and being adjustable axially for compensating for wear and diametral slackness of the end plate 7. Suitable shims may be interposed between the outer margin of the plate 7 and end wall of the casing portion 3 for effecting this adjustment. The barrel is sufficiently rigid to withstand deflection intermediate the bearings so as to reduce any tendency to bind on the valve pintle to be described.

The barrel 10 has a plurality of circumferentially spaced radial cylinders 18 positioned midway between the rollers 16 and 17 and each opening through suitable respective ports 18a into a central axial valve bore in the barrel 10. The barrel 10 has a radial flange 19 positioned in the zone of the cylinders 18 and in alignment with each cylinder there is provided in the flange a radial guideway 20, as better illustrated in Fig. 2. Mounted in the respective cylinders 18 are reciprocable plungers 21, each of which is provided with an enlarged head 22 which is reciprocable radially within the associated guideway 20. Each head 22 has a bore 23 which extends parallel to the barrel axis and in which is received an operating crosspin 24 having reduced end portions 25. The central or enlarged portion of the crosspin is received in the bore 23 and mounted therein on capillary cageless needle rollers 26 so as to be freely rotatable therein. The end portions 25 extend beyond the lateral faces of the heads 22 so as to cooperate with appropriate reactance means. The lateral faces of the heads 22 are parallel to each other and present flat plane surfaces of smooth finish.

In order to actuate the plungers through the medium of the pins 24, a pair of coaxial reactance rings 27 are provided at opposite sides of the plungers and ends of the flange. The flange likewise has flat parallel radial end walls which lie substantially in the planes of the side faces of the plunger heads, the flange being slightly narrower than the plunger heads. The rings 27 are provided with their adjacent walls flat and parallel so as to lie snugly adjacent the opposite faces of the flange 19 and lateral faces of the plunger heads 22, operating clearance only being provided therebetween. Each of the rings 27 is substantially square in cross section and is provided with annular grooves 28 opening through the radial walls adjacent the barrel flange and receiving the protruding ends 25 respectively of the crosspins 24 with sufficient clearance to permit rolling engagement between each pin end and one wall only of the associated groove at any given instant, either the radial outward wall or the inward wall.

The rings 27 are mounted in an annular reactance member or stator 30, the stator com-

prising a ring having its inner face beveled axially from the mid-portion and forming thereby the outer raceways 30a for tapered roller bearings 31. The rings 27 are provided with annular slanting grooves correspondingly formed to receive the tapered rollers 31, as indicated at 27a, so that standard tapered rollers may be used for mounting the rings in condition to resist both radial and axial thrust in a compact manner. As better illustrated in Fig. 2, the stator 30 is provided with slide supporting bearing surfaces 32 at diametrically opposite portions. The casing 3 is provided with complementary flat parallel bearing surfaces 33 so that the stator may be supported within the casing 3 with its axis parallel to the axis of rotation of the barrel while being shiftable to change the eccentricity of the reactance means relative to the barrel and its associated plungers. Control rods 34 are connected to the stator 30 and extend diametrically opposite from each other, through suitable bores in the casing 3 so that the stator and consequently the stroke of the pump may be adjusted from the outside. The bores and the control rods 34 are provided with slight clearance space so that the stator 30 may be self-adjusting with the rollers 31 in order that the rollers 31 match their tapered grooves in rings 27 respectively.

As set forth above the barrel 10 is provided with an axial bore with which the cylinders 18 are in communication, each through its individual port 18a. For effecting the proper valving relation a pintle 35 is provided, the pintle having an enlarged shank portion intermediate its ends by which it is fixedly secured in the partition wall 4. One protruding end of the pintle is received in the valve bore of the barrel 10 and is provided with a valve portion 35a which lies in the zone of the cylinders 18 and terminates as closely as practicable axially beyond the ports 18a so that the entire valving portion 35a of the pintle may be maintained completely covered with hydraulic fluid and a hydraulic pressure oil film provided thereabout for preventing elastic deflection and binding of the valve portion of the pintle as a result of hydrostatic unbalance. Beyond the valve portion remote from the wall 4 the pintle is provided with a reduced diameter race portion 35b on which operate capillary cageless needle rollers 36 which, in turn, are in rolling engagement with the walls of the barrel bore. The barrel bore is counterbored to a larger diameter at the end of the barrel adjacent the wall 4 so that a larger diameter portion of the pintle may be provided and extend as close as possible to the valving portion 35a. On this enlarged diameter are a pair of race portions 35c on each of which is a set of cageless roller bearings 37 which engage the circumferential wall of the counter-bore of the barrel. To provide a combination anti-friction and high pressure oil film bearing between the barrel and pintle and thus insure clearance for the instantaneous formation of the film between the pintle and barrel bore wall, the rollers preferably are of the capillary cageless combination or, if not true capillary rollers, are spaced preferably a capillary distance apart from each other. The pintle valve portion 35a is provided with a valve pressure port 38 and a diametrically opposite suction port 39, the ports being successively cooperable in valving relation with the cylinders consequent upon rotation of the barrel, and being reversible. The opposite end of the pintle is provided with a pressure port 44 and a suction port 45 corresponding to the

ports 38 and 39 respectively and in valving co-operation with the cylinders of the smaller barrel 50 of the pump. These ports are reversible simultaneously with the ports 38 and 39 by reversing the main ports.

As illustrated in Fig. 2, the pressure port 38 is connected with longitudinal ducts 40 formed in the pintle and the suction port 39 is connected with ducts 41 likewise formed in the valve pintle.

In the portion of the pintle by which it is mounted in the wall 4 is a main pressure port 42 and a diametrically opposite main suction or low pressure port 43. The main pressure port 42 is connected through the ducts 40 with the individual ports 38 and 44 of the stage units and the suction main port 43 is connected through the ducts 41 with the individual ports 39 and 45 of the stage units. The ducts 40 consequently provide communication between the ports 38 and 44 and one of the main ports 42 of the two stage units. The suction ducts 41 provide communication between the main port 43 and the individual ports 39 and 45. Thus the ports 38 and 44 connect with one main port 42 and the ducts 39 and 45 connect with the other main port 43 of the pump. In this manner both the barrel 10 and the barrel 50 may be operated independently by their respective shafts 9 and 8. As illustrated, the reactance of the smaller barrel 50 is also adjustable so that both the large and small units of the pump may be independently regulated for speed and torque and independently reversed.

Referring again to the barrel 10, it is oftentimes desirable and even necessary that coupling means be provided between the barrel and the reactance for positively coupling the same for starting the reactance in synchronism with the barrel and for maintaining the positive coupling engagement therebetween. For this purpose a plurality of bores 51 positioned one bore between each pair of adjacent cylinders or guideways 20 are provided in the flange 19, these bores extending parallel to the barrel axis. The bores are spaced preferably radially inwardly from the outer limits of the flange so that bridge portions 52 provide for retaining the strength of the flange at its outer periphery and for preventing deformation thereof under stress or during heat treating of the metal. In the reactance rings 27 are coaxial holes extending parallel to the barrel axis, the holes of one ring being aligned with those of the other and being positioned radially of the rings so as to lie within the radial limits of the grooves 28. Carried in each pair of aligned holes are coupling and clamping bolts 54. These holes correspond in number and, in the center position of the pump, to the bores 51 so that they may be aligned therewith. Coupling bolts 54 are received in each pair of aligned holes and extend through the bores 51, one end of each of the bolts 54 being in screw threaded engagement with one of the rings and the other end being provided with a head portion 54a so that the rings may be drawn toward each other.

Mounted on each bolt 54 is a sleeve 55 of proper diameter to snugly fit at its ends within the grooves 28 so that the ends of the sleeves engage the bottom or end wall of the grooves 28, thus fixing the spaced relation of the rings 27. Mounted on each sleeve 55 is a rotatable sleeve 56 of sufficient inside diameter to permit the formation of capillary oil films between them and the corresponding sleeves 55. The sleeves 56 lie entirely between the rings 27 with operating

clearance at the ends so that they may come into rolling contact with the circumferential walls defining the holes 51 at the maximum stroke of the pump unit only, except that upon starting, the sleeves 56 may engage the walls of the bores 51 at any stroke so as to start rotation of the reactance by the barrel or, if the reactance is rotated, may start the barrel without undue racing of the pins 25 in the grooves 28. When operating at full eccentricity, however, the diameter of the sleeves 56 and bores 51 are such that the sleeves remain in rolling engagement and roll entirely around the circumferential walls of the bores, thus affording a positive coupling means between the barrel and reactance through the medium of the flange 19.

As will more fully appear hereinafter, it is sometimes desirable that the barrels 10 and 50 of the pump be mechanically connected and thereby constrained to a fixed speed relation with respect to each other, as more fully described in my opening application, Serial No. 7809, filed February 23, 1935. For this purpose the structure illustrated in Fig. 3 is highly desirable. In Fig. 3, the parts corresponding to the parts illustrated in Fig. 1 are designated by corresponding primed numerals so that the parts and operation thereof are clear without additional description. However, in addition to the parts heretofore described, there is provided in the wall 4' a bore extending parallel to the barrel axis but disposed eccentrically thereto. Mounted within this bore is a sleeve 60 in which is rotatably mounted a shaft 62, the shaft 62 being supported in the sleeve 60 in sets of capillary cageless rollers 63 retained by suitable spacing washers. Keyed to the opposite ends of the shaft 62 are spiral gears 64 and 65 respectively. On the barrel 10' there is provided a spiral gear 66 complementary to the gear 64 on the corresponding end of the shaft 62 and in engagement therewith. A gear 67 complementary to the gear 65 and in engagement therewith at all times is provided on the barrel 50'. By this arrangement the barrels are geared together and mechanically constrained thereby to a fixed speed ratio with respect to each other at all times. The size of the gears 64 and 66 and the gears 65 and 67 may be changed to provide any fixed ratio desired. Generally, however, the larger pump unit is driven at a slower speed in inverse proportion to the power ratio than the smaller unit, to obtain the greatest possible efficiency for a greater power input. Consequently, the gear 64 is preferably smaller than the gear 65 so as to effect a step-down speed relation from the small high speed barrel 50' to the larger slow speed barrel 10'. Regardless of the speed ratio maintained between the barrels, however, by this mechanical connection, the reactances of the barrels may be separately adjusted, thereby adjusting the speed of the interconnected barrels when operated as a motor. Again, the stroke controlling means of both the large unit and the small unit may be mechanically secured for operation in a fixed ratio to each other when desired. In the present application, however, as distinguished from my copending application, Serial No. 7809, filed February 23, 1935, both units are provided with impeller shafts 8' and 9' respectively for driving connection with independent prime movers of fixed speed ratio to each other, when used as a pump and for connection to separate means, or selective gears of a train, to be driven by the structure when used as a motor.

Referring next to Figs. 4 to 6 inclusive, some applications of the hydraulic machine herein illustrated are discussed for showing the flexibility and advantages of the structures.

5 Referring to Figs. 4 and 5, there is illustrated a connection of the hydraulic machine of Fig. 3 for use as a motor. In this application of the machine, suitable brackets 60a and 61a are secured to the casing portions 2' and 3' respectively and carry bearings 62a and 63a respectively, in which is mounted a rotatable driven shaft 64a. On the opposite ends of the shaft 64a are spiral gears 66a and 67a respectively, the gear 66a preferably being of smaller diameter than the gear 67a. These gears are connected to the shaft by suitable spline connections 68a and 69a respectively, so that the gears may be shifted along the shaft by a suitable shifting fork 70 to operative and idle positions. On the shafts 8' and 9' respectively of the machine are mounted gears 72 and 73, the gear 73 being smaller than the gear 72 and in mesh with gear 67a in one shifted position of the gear 67a. The gear 72 is in mesh with an idling gear 74 positioned for cooperation with the gear 66a when the gear 66a is shifted by the fork 70 to proper position. The fork 70 is operable to shift both the gears 66a and 67a simultaneously so that one becomes operative and one idle concurrently.

30 In the form illustrated, the port 42' is considered the main pressure port and the port 43' is the main suction or low pressure port. As illustrated in Fig. 5, a return pipe line 76 is connected to the port 43' and communicates with a suitable sump 78. A working pressure pipe line 80 is connected with the port 42' and is provided with a valve 81 for purposes later to be described. The line 80 is connected to the pressure side of a reversible, variable delivery pump A, which supplies the operating pressure fluid, the pump A being also connected to the sump 78, as shown.

The reactances or stroke controlling means of the respective units are controlled independently by suitable hydraulic servo-motors through their pilot controls 82 and 83 which, in turn, operate control rods such as the rod 34' of Fig. 3. The servo controls 82 and 83 are more fully described in my copending application, Serial No. 749,746, filed October 24, 1934 and may utilize the pressure fluid from a separate pump for moving the reactances connected thereto in accordance with the setting of the pilot valves of the control through the medium of the rods 84 and 85 respectively.

55 In order to illustrate the operation of the structure as a motor and to illustrate its advantages, the operation will be described in connection with the driving of a spindle of a machine tool in which a low speed cycle of from 30 R. P. M. to 300 R. P. M. is used at one time and a high speed cycle of about 1200 R. P. M. is used at another time. Only occasionally are speed cycles of 600 R. P. M., 800 R. P. M. and other speeds between 300 R. P. M. and 1200 R. P. M. required. Fluid under pressure is supplied from a constant delivery pump A through the line 80, the valve 81 being open. The pressure fluid flows to both units or stages of the motor so that the barrels 10' and 50' may be operated concurrently or selectively. The reactances of the units are independently adjusted for operation in the desired relation to each other. If it is required that the shaft 64a is to be driven in the same direction by both units, the small unit may be reversed rel-

ative to the large unit so as to compensate for the reversal of drive due to the idler gear 74.

The slowest speed of 30 R. P. M. is obtained with a correspondingly large torque by setting both units at maximum eccentricity, thus utilizing the combined motor capacity for speed reduction. However, the large unit is so dominant in the capacity ratio, that no appreciable under-speed drive results from the addition of the capacity of the small unit.

If, now, speeds from 30 R. P. M. to 300 R. P. M. are required, it is only necessary to gradually reduce the stroke of the large unit and this may be done throughout this speed range without any appreciable change in efficiency or horsepower output. If the highest speed of the shaft 64a is required, the gears 67a and 73 are disassociated and the gears 72 and 74 and 76 associated. The capacity of the large unit is thereupon set at zero. This action, due to the much greater capacity of the large unit, will simultaneously increase the speed of the small unit to its critical speed at which the shaft is driven at 1200 R. P. M.

Due to the capacity ratio of the two units, the speed of the small unit will be controlled through a wide range by changes in the stroke or eccentricity of the large unit, as each unit is prevented from racing by the geared connection therebetween through gears 64, 65, 66 and 67 within the casing. If, therefore, the occasional intermediate speeds of 600 R. P. M., 800 R. P. M. etc. are required, these are obtained from the small unit by adjusting the stroke of the larger unit so that only the required part of the pressure fluid passes to the small unit. The fluid passing to the large unit is not wasted, but is added by the large unit to the same shaft 64a.

During these operations, the units are operated at high efficiency, since each dominates throughout its most favorable speed range. Thus the increase and decrease in speed is stepless and one unit is used as a valving means for controlling the volume of fluid passed to the other, thereby controlling the speed of the latter. Consequently, throughout their operating ranges, each unit may operate at substantially its full stroke to insure efficiency in operation. If over-speed is required, obviously this can be obtained by reducing the stroke of the small unit after the large unit is idle and at zero stroke.

A contribution to the higher efficiency of the combination resides in the fact that a large and rigid unit is necessary to deliver large torque at low speed efficiently, whereas a small unit is necessary to deliver low torque at high speed efficiently. In the structure shown, since the maximum fluid capacity of the large unit is several times that of the maximum fluid capacity of the small unit, low speed is obtained by the large unit and high speed by the small unit for the same total amount of pressure fluid. Each operates within a wide range of speeds and torque at substantially constant horsepower with relation to the speed of the pump A, both units being of economical design for the range in which they operate. Furthermore, both units have infinitely variable speeds with efficiency within their operating ranges.

When the machine in connection with which the transmission is to be used and the operating range and characteristics are such that a continuous rise in speed from the lowest to highest limit is required, the units are chosen with respect to each other so that when the capacity of the larger unit is reduced to the point where its

efficiency begins to decrease appreciably and therefore this relatively large capacity has passed to the small unit, the small unit will be operating at the same speed as the larger unit and at high efficiency.

In many instances, such as in power lathes and the like, it is desirable that an extremely high torque at very slow speed be provided during the working cycle of the lathe tool and that very high speed and at low torque be provided for rapid traverse of the lathe tool to starting position. The connection illustrated in Fig. 4 is primarily for such operations. These are readily effected and a further range of efficient operation provided due to the fact that the large unit is geared to the shaft 64 in step down speed relation, whereas the small unit is geared to the shaft 64 at substantially equal speed relation. Thus on the working cycle of the lathe, the fork 70 may be shifted to mesh the gears 67 and 73 and the large unit operated at full stroke. At the end of the working cycle, the fork 70 is shifted, disengaging the gears 67 and 73 and engaging the gear 66 and the idler gear 74, whereupon, the small unit being set at full stroke, the stroke of the large unit decreased to zero so that, the rapid traverse of the lathe tool is completed and the working head is returned to starting position.

It might be pointed out, that by closing the valve 81 and driving either the large unit or the small unit as a pump, the other unit may be operated as a motor.

In Fig. 6 is illustrated the use of the structure as a pump, wherein two different prime movers are utilized and are to drive a common member, or as a motor driving two separate driven means. In the first instance, one prime mover B is connected to the large unit shaft 9 and another prime mover C is connected to the small unit shaft 8. Both units are then operated as pumps concurrently or independently and supply fluid through the common main pressure port 42, the fluid circuit being as illustrated by the arrows in Fig. 6. Such an application would be useful in integrating the power of two prime movers, one of which is high speed and one of which is low

speed. In the example given, the prime mover B may be a low speed prime mover and the prime mover C a high speed prime mover so that the small unit may be operated at high speed and the large unit at low speed at which respective speeds they are most efficient. Both may deliver the same pressure and even the same fluid capacity through the common port to a hydraulically driven means, such as the hydraulic motor D, thus providing efficient synchronization of the output of the prime movers B and C, for driving the motor or for driving through the driving shaft of the motor, an engine E connected to the shaft of the motor.

Having thus described my invention, I claim:

1. A hydraulic pump mechanism comprising a large capacity rotary radial plunger, independently variable delivery pump unit, a smaller capacity rotary, radial plunger, independently variable delivery pump unit, independent driving means for the units respectively, and means connecting said units in a fluid circuit in parallel with each other.

2. A rotary, radial plunger pump or motor comprising a casing, a plurality of coaxial barrel units mounted for rotation in the casing, plunger and cylinder assemblies carried by the barrel units respectively, reactance means for the assemblies, valve means for the assemblies respectively, individual shafts on the units respectively operable at different rotative speeds and each shaft being connected to its associated unit for rotation together therewith and being coaxial therewith.

3. A rotary, radial plunger pump or motor comprising a casing, a large barrel and a small barrel rotatably mounted therein, shafts carried one by each barrel and respectively rotatable together with the associated barrel independently of the other barrel, plunger and cylinder assemblies carried by the barrels respectively, independently adjustable reactance means for the assemblies respectively, and valve means fitting the barrels and in valving cooperation with the assemblies of the associated barrels.

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