This invention relates to fluid pressure devices and systems which are adapted to transmit power by means of fluid pressure and in particular to devices and systems of this character which include a vane type fluid motor in which the vanes are urged into contact with the vane track by fluid pressure means. The present application is a continuation in part of my co-pending application filed February 17, 1940, Serial No. 319,399.

The widest present use for devices and systems of this general class is as hydraulic devices and systems, that is to say, devices and systems for handling, or whose motive fluid, but liquid, such for example, oil. The present invention will accordingly be described in connection with such use although it will be understood that the invention is also applicable to devices and systems operating with elastic fluids.

Vane type motors of the character mentioned above include a vane track that surrounds the rotor and vane assembly. For quiet and satisfactory operation of the motor it is practically essential that the outer ends of the vanes be urged into contact with the vane track when operation of the motor is started and that such contact be maintained continuously during its operation. In order to provide the track-contacting and track-following action of the vanes it is necessary to supplement the action of centrifugal force with an auxiliary force acting to urge the vanes outward, at least during the portion of their rotary travel in which they are passing through the intake area or areas of the motor, so that the outer ends of the vanes will be held firmly in contact with the surrounding vane track and thus provide a movable resistance to the pressure fluid admitted to the outer ends of the vane motor, whereby rotary motion is imparted to the rotor and driven shaft of the vane motor. In the vane motor forming part of the fluid pressure device and system of the present invention, fluid pressure means are utilized to provide this auxiliary force and this is accomplished by introducing or admitting, behind the inner ends of the vanes, pressure fluid having a pressure greater than but related to the pressure of the fluid admitted to the pressure areas at the outer ends of said vanes, as fully explained in my co-pending application filed March 28, 1948, Serial Number 198,449.

Thus fluid under two different but related operating pressures is used; the fluid having the higher of these two pressures, which for convenience is termed the "differential high pressure fluid," is admitted to the radially inner ends of the vanes, while fluid under the lesser of these two pressures (for convenience termed the "track pressure fluid" or "operating pressure fluid") is admitted to the pressure areas at the outer ends of the vane motor. In the fluid pressure device and system of the present invention these two different but related pressures are obtained by passing the supply of fluid going to the motor through "differential pressure" or resistance mechanism positioned in the fluid inlet conduit.

When the vane type motor is of constant capacity (i.e., constant fluid capacity or displacement per revolution of its rotor) its speed or operation or rotation is controlled by regulating the volume of operating pressure fluid supplied to the outer ends of its vanes. The speed of a variable capacity vane motor may also be regulated by altering the capacity or displacement per revolution of the rotor thereof, but the torque of the motor will then vary substantially inversely with the speed of rotation so that in all events the volume of operating pressure fluid that is supplied to the outer ends of the vanes determines the power that is transmitted by the vane motor.

As indicated above, certain parts of the disclosure are common to the present application and the above-mentioned application Serial No. 319,399, such, for example, as the provision of unitary means for performing the dual functions of regulating the speed of the vane type motor and providing differential high pressure fluid for urging the vanes of the motor into contact with the vane track. The arrangement of the present invention differs from that of the above-mentioned co-pending application Serial No. 319,399, however, in several important respects. For example, the present invention provides novel and improved means whereby either a single vane type motor or a plurality of vane type motors may be operated by pressure fluid supplied by a single source, with the speed of each motor varied or controlled at will (and independent of the speeds of the other motors when a plurality of motors are employed). By way of further example, in the arrangement of the co-pending application Serial No. 319,399 the speed of the motor is varied and controlled by varying and controlling either the volume of pressure fluid supplied by a variable delivery pump or by by-passing the unused portion of the fluid volume delivered by a constant capacity pump, whereas according to the present invention the speed of each vane type motor is varied and controlled solely by regulating and controlling the volume of fluid which is permitted to pass to or through each motor, without respect to the quantity of pressure fluid available from the source or the disposition of any surplus of the fluid pressure supply. Other differences will appear from comparison of the accompanying drawings with those of the co-pending applica-
tion Serial No. 310,399 and from the description which follows.

An object of the present invention is to provide an improved fluid pressure device and system of the character above indicated.

Another object is to provide an improved, simple and economical fluid pressure device and system including one or more vane type fluid motors and employing for each motor unitary means for performing the dual functions of regulating the speed of the corresponding motor and of providing fluid at a pressure sufficient to urge the vanes of said motor into contact with its vane track.

Another object is to provide a fluid pressure device and system of the character above set forth, and in which the speed of each vane type motor and the difference between the pressures of the operating pressure fluid and the fluid supplied for urging its vanes into contact with its vane track are held substantially constant irrespective of the load that is imposed on the vane motor and independent of the speed of any other motor employed in the fluid pressure system.

A further object is to provide such a fluid pressure device and system in which the speed of each vane type motor is held substantially constant independent of change in viscosity of the circulated fluid.

Other and more specific objects will appear from the description which follows.

The invention will be understood from a consideration of the accompanying drawings which illustrate, by way of example, several embodiments of the present invention.

In the accompanying drawings:

Fig. 1 is a diagrammatic view, partly in section, showing an illustrative embodiment of the present invention in a fluid pressure system that includes a plurality of vane type motors;

Fig. 2 is a fragmentary diagrammatic view, partly in section, showing a modification in which means are provided to compensate for change in viscosity of the circulated fluid;

Fig. 3 is a diagrammatic view, partly in section, showing another modification;

Fig. 4 is a longitudinal sectional view, taken along the line 4—4 of Fig. 5, of an illustrative embodiment of the vane type motor forming part of the fluid pressure device and system of the present invention;

Fig. 5 is a view in vertical section transverse the axis of rotation of the vane type motor and is taken along the line 5—5 of Fig. 4;

Fig. 6 is a view in vertical section transverse the axis of rotation of the vane type motor and is taken along the line 6—6 of Fig. 4, looking in a direction opposite to that of Fig. 5;

Fig. 7 shows an inner elevation of one of the members of the vane motor, for convenience termed an "end plate" or "cheek plate"; and

Fig. 8 is a sectional view of the cheek plate taken along the line 8—8 of Fig. 7.

The embodiment of the invention illustrated in Fig. 1 includes a plurality of vane type motors B, here shown as two in number although a smaller or greater number may be employed if desired. The vane type motors B may be of the same or of different constructions. For example, each motor B may be of either constant or variable displacement per revolution of its rotor, or one may be constant displacement and the other of variable displacement. For purposes of illustration, however, I have chosen a constant capacity vane type motor in which the vanes move inward and outward with respect to the rotor in a substantially radial direction; this illustrative motor forms no part per se, however, of the present invention but a part of said co-pending application Serial Number 198,449 and certain features of its construction are similar to those shown in co-pending application filed December 6, 1939, Serial Number 307,755. For convenience the same vane motor will be presumed to be employed which is illustrated in the accompanying drawings and will accordingly be first described.

Referring now to Figs. 4 and 8 inclusive, each motor B includes a casing 10 formed with an open-ended rotor cavity for the rotor 15 and associated parts as shown in Figs. 4 and 5. The rotor cavity is closed (Fig. 4) by an end head or cover member 11 which is attached to the casing 10 by cap screws 12. The rotor 15 is provided with a plurality of vanes 17 which are movably arranged in a substantially radial direction inward and outward in the vane slots 16. A vane track ring 25 surrounds the rotor and vane assembly and its inner circumferential surface 26 forms a track adapted to contact the radially outer ends of the vanes 17 as an axially extending end plate 11 serves to guide and control the vanes in their inward and outward movement; the surface 26 will hereinafter be referred to as the "vane track".

The rotor 15 and driven shaft 20 may be mounted and the two parts may be operatively connected with each other in any appropriate manner. In the present instance the rotor 15, shaft 20, their mountings and the operative connection there-between are as shown in Figs. 4 and 5, as an example of the method of mounting. The rotor 15 is supportedly mounted on the end of the shaft 20 which projects into the rotor cavity. For this purpose the end of the shaft 20 is formed with axially extending splines 21 (Figs. 4 and 5) and the rotor 15 is formed in its central opening with mating splines 18. The arrangement is such that the rotor 15 is freely movable in an axial direction on the shaft splines 21 while also permitting a limited angular or rocking motion of the rotor 15 relative to the shaft 20 in such a manner that the cheek plates 34 and 35, to be presently described, determine the axial and angular position of the rotor on the shaft and the plane of rotation of the rotor as fully explained in co-pending application Serial Number 307,755 above mentioned.

The rotor 15 is hydraulically balanced with respect to all forces imposed thereon by fluid pressure. Hydraulic balance of forces acting on the rotor in a radial direction is obtained by dividing the space intermediate the periphery of the rotor 15 and the vane track 26 into two equal and oppositely positioned fluid sections, each fluid section comprising a working chamber flanked by an inlet area and an outlet area. As shown in Fig. 5, the division between the two fluid sections is effected by co-operation of the rotor 15 and the outer ends of the vanes 17 with the vane track 26 at the regions of the vane track's least inclination. In the present embodiment is adjacent the horizontal centerline. The vane track 26 is preferably provided at each of these points of division with an arc 27, for convenience termed the "sealing arc," substantially concentric with the rotor 15 and extending in a circumferential direction for a distance equal
to at least the angular distance between a pair of adjacent vanes 17.

The working chambers of the two fluid sections are formed by means of two diametrically positioned arcs 31, preferably concentric with the rotor 15 and termed “working arcs,” which are located in the regions of greatest diameter of the vane track 26. The working chambers extend in a circumferential direction for an arcuate distance substantially equal to the distance between the outer ends of two adjacent vanes 17 which at any given instant are moving in contact with the working arcs 31. Operating pressure fluid is admitted between the vanes as they move through the inlet areas toward the working chambers and is discharged as the vanes recede therefrom through the outlet areas of the two fluid sections. The inlet area of each fluid section is thus at all times separated from the outlet area of the same fluid section by at least one of the vanes 17 and the difference in pressures on the opposite sides or faces of such vane causes rotation of the rotor 15, which in the figures is shown being driven by fluid pressure. As viewed in Fig. 5. The portions of the vane track 26 intermediate the sealing arcs 27 and working arcs 31 may be given any suitable curvature producing satisfactory rates of inward and outward movement of the vanes 17 as the rotor 15 revolves.

The sides or axial ends of the working chambers are closed by a pair of mating disc-shaped members 34 and 35 (Figs. 4, 5, 7 and 8), for convenience termed “end plates” or “check plates,” which are carried freely while in use, and the sides of the vanes 17 form substantially fluidtight running fits with the adjacent faces of the check plates 34 and 35. The check plate 34 will hereinafter be termed the “casing check plate” and the check plate 35 will be termed the “end head check plate.”

The check plates 34 and 35 are each provided with co-extensive mating ports (Figs. 4, 5 and 7), the ports of one check plate being axially opposed to the ports of the other check plate when the ports are in position in the casing 10 so that all forces exerted upon the rotor 15 and vanes 17 in an axial direction by fluid pressure are thus completely balanced. The ports in the check plates 34 and 35 will be best understood from Figs. 7 and 8, in which Fig. 7 shows an inner elevated or rotor face of the end head check plate 35. Referring to Fig. 7, each check plate is provided with a pair of diametrically opposed arcuate inlet slots or ports 36 and a similar pair of diametrically opposed outlet slots or ports 37; these ports are also partially shown in Fig. 5 and the inlet ports 36 are shown in the sectional view of Fig. 4 and the outlet ports 37 are shown in section in Fig. 8. Operating pressure fluid is admitted to the outer ends of the vanes 17 through the inlet ports 36 of the casing check plate 34 and, similarly, fluid discharged or exhausted by said vanes passes out through the outlet ports 37 of the same check plate. The ports 36 and 37 of the end head check plate 35 function principally as “balance ports” to contain fluid under the same pressure as that in the corresponding ports of the casing check plate 34 in order to produce hydraulic balance of the rotating parts, as already stated.

Each of the check plates 34 and 35 is also provided with two pairs of arcuate recesses or vane slot ports 38 and 39 in the faces thereof adjacent the rotor 15 as best shown in Fig. 7; the vane slot ports 38 are, however, also shown in the sectional view of the check plate 34 and the vane slot ports 39 are likewise shown in the sectional view of Fig. 8. These vane slot ports 38 and 39 are positioned to register successively with the inner ends of the vane slots 16 as the rotor revolves and the vane slot ports of each pair are positioned diametrically opposite each other. The arrangement is such that the inner end of each vane slot 16 connects with one of the vane slot ports 38 while the vane slot 17 therein is passing through the inlet area of each fluid section and also while traversing the outlet area of each fluid section; and the vane slot ports 39 of the casing cheek plate 34, and preferably of both check plates 34 and 35, are connected with the corresponding outlet ports of the casing check plate 34 by radial grooves 32 formed on the outer faces of said check plates, as indicated by dotted lines in Fig. 7 and shown in the sectional view of Fig. 8. Fluid pressure applied by the arcuate recesses on the sides of the vanes passes out through the outlet ports of the casing check plate 34.

As already stated, in order for the motor B to operate quietly and smoothly it is necessary to supplement the action of centrifugal force with an auxiliary force urging the vanes 17 into contact with the vane track 26 during at least the portion of their rotary travel in which the outer ends of said vanes are passing through the inlet areas of the motor. This is accomplished by introducing behind the inner ends of the vanes 17 through the vane slot ports 38, pressure fluid (hereinafter termed the “differential high pressure fluid”) having a pressure greater than but correlated with the pressure of the operating pressure fluid supplied to the inlet areas of the motor 38 where it acts upon the exposed outer ends of the vanes, as fully explained in co-pending application Serial Number 198,449 to which reference has already been made. Each of the vane slot ports 38 of the end head check plate 35 is accordingly provided with a hole 39 (Figs. 4 and 7) that registers with the inner end of one of the two passages 13 formed in the end head 11. The outer ends of the passages 13 connect with a passage 14 which in turn is properly connected with a conduit 45 through which differential high pressure fluid is supplied so that the differential high pressure fluid being obtained in a manner to be presently explained. The vane slot ports 38 of the casing check plate 34 are not directly connected with the differential high pressure fluid supply and act principally as “balance ports” to contain a supply of fluid (which is received through the vane slots 16), having the same pressure as the fluid in the vane slot ports 38 of the end head check plate 35, in order to substantially balance the hydraulic forces acting on the sides or axial ends of the vanes 17 and rotor 15 and thus prevent binding of the parts.
The fluid circuit of the motor B also includes a branched fluid inlet channel 40 (Figs. 4 and 6) and a branched fluid outlet channel 41, both of which are formed in the casing 10. The fluid inlet channel 40 is connected with the fluid supply conduit 42 and is also connected with the fluid inlet ports 36 of the casing check plate 34 as shown in Fig. 4. The fluid outlet channel 41 is similarly connected with the outlet or exhaust conduit 43 and with the outlet ports 37 of the casing check plate 34 by slanted passages, not shown, similar to the slanted passages 44.

In the embodiment of the invention illustrated in Fig. 1 the two vane type motors B are operated by pressure fluid supplied therefor through a branched fluid supply conduit or line 42 which in turn is supplied with pressure fluid by any suitable source, not shown, such, for example, as an accumulator, a reservoir or a pump with suitable output control or fluid escape means therefor. Fluid exhausted by the two motors passes through the branched discharge conduit 43 with which each motor is appropriately connected.

The differential high pressure fluid for urging the vanes 17 of each motor B into contact with its corresponding branch of the valve piston 60, is supplied by providing a variable orifice 70 in the branch of the conduit 42 leading to each motor B and the resistance to flow through each of said orifices creates the difference in pressures between the differential high pressure fluid going to the inner ends of the vanes 17 and the operating pressure fluid going to the outer ends of the vanes of the corresponding motor B. The conduit 45 of each motor B is accordingly connected with its corresponding branch of the conduit 42 at a point on the inlet side of the corresponding orifice 70. The volume of pressure fluid permitted to pass through each variable orifice 70 is regulated to provide the proper volume to produce a predetermined pressure drop thereacross, for any extent of opening of the corresponding orifice 70, so that the difference in pressures between the differential high pressure fluid and the operating pressure fluid is held substantially constant and change in the volume of fluid passing through said orifice 70 is effective responsive to the pressure drop actually existing across said orifice 70 relative to the predetermined pressure drop thereacross. The means by which this is accomplished will now be described.

A fluid flow or control valve means, broadly designated by the reference numeral 50, is provided for each vane motor B and, while only one of them is shown in section in Fig. 1, it will be understood that they are of identical structure and arrangement in this embodiment of the invention. Each control valve 50 includes a valve housing 51 having two valve bores 52 suitably closed at both of its ends as by covers 53 and 54 respectively. Each valve bore 52 is provided with an annular inlet port 55 and an annular outlet port 56 which are axially spaced from one another in the valve bore so that the portions of the supply conduit 42 connected with these ports have a somewhat offset positional relation to each other.

Suitably fitted in each valve bore 52 is a valve piston 60 having two heads 61 and 62 respectively which are of the same cross-sectional area and are spaced from one another by a reduced neck 63 having a tapered portion 64 adjacent the head 62. The head 62 controls the extent of connection between the inlet port 55 and outlet port 56, this control of connection being effected by the extent to which the head 62 closes or covers the outlet port 56 and hence the extent to which said outlet port 56 is connected with the inlet port 55 through the portion of the valve bore 52 intermediate the ports 55 and 56. It will be understood that partial covering or closing of the outlet port 56 by the head 62 presents a resistance to the flow of fluid therethrough, which resistance increases as the head 62 approaches its position in which it completely closes the outlet port 56 and prevents the passage of any fluid therethrough. The movement of the valve piston 60 in the valve bore 52 is preferably limited by proportioning of the parts to permit the head 62 to completely close the outlet port 56 in its extreme downward position of movement and to completely open or uncover the outlet port 56 in its extreme upward position. Similarly, the proportions are preferably made such that the inlet port 55 is at all times connected with the reduced neck 63 or tapered portion 64 of the valve piston 60.

The valve piston 60 is moved and its position is controlled responsive to the pressure drop actually existing across the orifice 70 relative to a predetermined pressure drop thereacross. The upper end of the valve bore 52 is connected with the inlet side of the orifice 70, as by a passage 72 here shown as branching from the passage 45 leading to the inner ends of the vanes 17. Similarly, the lower end of the valve bore 52 is connected, as by a passage 73, with the outlet side of the orifice 70 and the lower end of said valve bore 52 is also provided with a spring 65 which exerts a force supplementing the upward force exerted on the valve piston 60 by action of pressure fluid from the outlet side of said orifice 70. It is thus seen that pressure fluid passing the inlet side of the orifice 70 exerts a force on the valve piston 60 in a direction tending to move it downward to thereby cause the head 64 to close the outlet port 56, thus decreasing the volume of fluid permitted to pass through said outlet port 56; this force is opposed and balanced by the combined forces of the spring 65 and the action of the pressure fluid from the outlet side of the orifice 70 which exert a force tending to move the valve piston upward and to increase the volume of fluid permitted to pass through the outlet port 56.

The valve piston 60 is thus moved responsive to the pressure drop across the orifice 70 and takes a position to permit the passage through the outlet port 56 of just the proper fluid volume to produce a pressure drop across said orifice 70 equal in amount to the value determined by the spring 65. Upon any departure of the actual pressure drop from the predetermined amount thereof, the valve piston 60 is immediately moved in a direction to correctively alter the volume of fluid passing through the outlet port 55 so that the predetermined pressure drop across the orifice 70 is thus restored and maintained substantially constant at all times. These corrective changes and restoring movements of the valve piston 60 take place almost instantaneously and the adjustments are such as to set the corrective mechanism into operation upon slight departures in the pressure drop to be maintained across the orifice 70.

It will thus be seen that the spring 65 determines the amount of pressure drop to be maintained across the orifice 70 and hence the difference in pressures between the differential high pressure fluid and the operating pressure fluid. The compression of the spring 65 is accordingly
made such as to provide a pressure drop across the orifice 70 of an amount or value such that the differential high pressure fluid from the inlet side of the orifice 70 exceeds the pressure of the operating pressure fluid from the outlet side of the orifice by an amount sufficient to provide satisfactory action of the motor vanes 17. Position and movement of the valve piston 60 are determined and effected entirely by relative pressures existing on the inlet and outlet sides respectively of the orifice 70 and are substantially independent of absolute pressures. It is thus possible, for example, to operate two or more vane type motors, with each motor operating at a different but controlled speed, as is frequently desirable, and with the speed of each motor capable of easy and prompt variation. All of the motors in the system may be operated at the same speed, with assurance that the speed of each and all of them will be held substantially constant irrespective of variation of load or pressure imposed thereon and with almost instantaneous correction of variation from the predetermined speed. Further, the speed control and assured difference in pressures between the differential high pressure fluid and the operating pressure fluid is provided by means which are simple and inexpensive yet are dependable and prompt in action.

In the embodiment illustrated in Fig. 1 the speed of each motor B will be held substantially constant for any extent of opening of its corresponding variable orifice 70 provided the viscosity of the circulated fluid remains constant. Lubricating oil is usually employed as the circulated fluid in systems of this character and is subject to relatively large changes in viscosity as its temperature changes. Such viscosity changes alter the resistance to flow through the orifice 70 and therefore noticeably affect the volume of fluid that is permitted to pass there through (for any given opening of said orifice 70) which in turn causes corresponding change in the speed of the motor B unless means are provided to compensate for such viscosity changes. Such viscosity-compensating means are provided in the embodiment illustrated in Fig. 2 and will now be described. For convenience in illustration, only one branch of the supply conduit 42 which to a variable extent for has been shown in Fig. 2 but it will be understood that said conduit 42 may have either one or a plurality of branches, each of which is provided with a motor B and associated mechanism therefor.

The modified fluid flow control valve means 50' of Fig. 2 is generally similar to the control valve means 50 of Fig. 1 except for the provision of the viscosity-compensating means. It includes a valve housing 51' having a valve bore 52' provided with axially spaced annular inlet and outlet ports 55' and 56' respectively which are connected with the portions of the supply conduit 42. A valve piston 60' is slidably fitted within the valve bore 52' and is provided with a pair of heads 61' and 62' separated by a reduced portion 63' having a tapered portion 64' adjacent the head 62'. The head 62' is adapted to cover or close to varying extents the outlet port 55', to regulate the passage of fluid therethrough, in the same manner as already explained in connection with the valve piston 60 of Fig. 1. The valve piston 60' also includes, however, a pair of extension rods 77 and 78.
of equal diameter, which extend from the heads 76 of and 82 respectively and project through suitable openings in the closures for the ends of the valve bore 52' in such manner that they form substantially fluid tight fits therewith. The upper and lower ends of the valve bore 52' are connected with the inlet and outlet sides respectively by the passages 12 and 13 and a spring 65' in the lower end of the valve bore 52' surrounds the rod 71 and exerts a force on the valve piston 60 tending to move it upward, supplementing the upward force exerted thereon by action of pressure fluid from the outlet side of the orifice 70.

It will thus be seen that this portion of the modified control valve means 50' of Fig. 2 is substantially identical with that of the control valve means 50 of Fig. 1 from which it differs, as described up to this point, principally with respect to the provision of the extension rods 17 and 78. This portion of the control valve means 50' is therefore capable of functioning in the same manner as described in connection with Fig. 1 and may, if desired, be so employed without regard to the viscosity compensating means which will now be described.

The viscosity compensating means includes a pair of cylinders or bores 79 and 80 respectively, for convenience termed "compensating cylinders," which in the present instance are formed in the members 75 and 76 which close the ends of the valve bore 52'. The compensating cylinders are provided with slidably fitted pistons, termed "compensating pistons," operatively connected with the valve piston 60 and the ends of the rods 71 and 78 are utilized as the compensating pistons in the present embodiment. Each compensating piston is of such size that its cross-sectional area equals the cross-sectional area of one of the end portions of the valve piston 60' which are exposed to the pressure fluid in the ends of the valve bore 52', that is to say, the cross-sectional area of the compensating piston equals the cross-sectional area of the head 61', or 62', minus the cross sectional area of the corresponding rod 71 or 78; this relation is here obtained by making the rods 71 and 78 of such size that the cross sectional area of each of them is one-half the area of a section through the heads 61' or 62' of the valve piston 60'.

The viscosity compensating mechanism also includes an auxiliary fluid circuit, which may be termed the "compensating circuit," which in turn includes a small constant capacity pump 83 adapted to be continuously driven at a constant speed. The pump 83 receives its supply of oil or other fluid through an inlet conduit 84 which is adapted to be connected in any suitable manner with a supply of fluid having the same viscosity as the fluid simultaneously passing through the orifice 70; as here shown the inlet conduit 84 is connected with the discharge conduit 43, preferably at a point in said conduit 43 where little or no pressure exists. The pump 83 is also provided with a discharge conduit 85 leading to a suitable reservoir, not shown, and having an orifice 86 which is here shown as a variable orifice although a fixed orifice may be employed since the size thereof is not adjusted during operation.

With this arrangement, the amount of the pressure drop across the orifice 86 will be constant as long as the viscosity of the circulated fluid is constant but will vary immediately and conformably with any change taking place in the viscosity of the fluid. As the fluid passing through the orifice 86 is of substantially the same viscosity as the fluid passing through the orifice 70, it will be seen that change in viscosity of the circulated fluid will produce identical changes in the amounts of the pressure drop across both orifices for the fluid passing through them. The rate of fluid flow through the orifice 86 is constant for the reason that the pump 83 is of constant capacity and is driven at constant speed. Hence the change in the pressure drop across the orifice 86 is an exact measure of the corresponding change, due to change in viscosity, which takes place during the same interval in the amount of pressure drop across the orifice 70 for any particular rate of fluid flow there-through.

According to this embodiment, the change taking place in the amount of the pressure drop across the orifice 86 is employed to correspondingly modify the amount of pressure drop to be maintained across the orifice 70. The compensating cylinder 78 is therefore connected, as by a passage 81, with the discharge conduit 85 at a point on the inlet side of the compensating cylinder 80 which is similarly connected, as by a passage 82, with said discharge conduit 84 at a point on the outlet side of said orifice 86. The compensating cylinders 79 and 80 are thus supplied with fluid having the same pressures as the pressures existing on the inlet and outlet sides respectively of the orifice 86.

Two additional opposing forces are thus brought to bear upon the valve piston 60' by the compensating pistons. These two opposing forces have a net difference tending to move the valve piston 60' upward, this net difference corresponding to and varying with the amount of the pressure drop existing across the orifice 86. The effect, therefore, is that of a force tending to move the valve piston 60' upward and which varies commensurably with the amount of the pressure drop across the orifice 86; hence likewise varies with the viscosity of the fluid.

The net difference of forces thus exerted upon the valve piston 60' by the compensating pistons combines with the cross-sectional area of each of the portion of the circulated fluid and in exact accordance with the change occurring in the amount of the pressure drop across the orifice 70, with a constant size of opening thereof and a constant rate of fluid flow there-through, resulting from viscosity change. In other words, decrease in viscosity of the fluid reduces the amount of pressure drop across the orifice 70 produced by a constant flow of fluid there-through and therefore reduces the net difference of the forces exerted upon the valve piston 60' by fluid from the inlet and outlet sides of the orifice 70, which net difference or resultant tends to move the valve piston 60' downward; this decrease in viscosity simultaneously also reduces, by exactly the same amount, the sum of the combined opposing forces which tend through addition of compensating pistons to move the valve piston 60' upward. In the same manner, increase in viscosity of the fluid simultaneously and equally increases the forces tending to move the valve.
piston downward and those tending to move it upward. The relative balance of forces acting upon the valve piston 60' is therefore undis-
turbed by change in viscosity of the fluid. Change in viscosity thus affects the amount of pressure drop to be maintained across the orifice 70, and alters the amount thereof in exact accordance with the effect of such change in viscosity upon the amount of the pressure drop actually taking place across the orifice 70 with a constant flow rate of fluid therethrough. The compensating mechanism therefore cooperates with the other parts of the mechanism to hold substantially constant the volume of operating pressure fluid passing through the orifice 70 to the outer ends of the motor vanes 17 and hence to hold the speed of the motor B substantially constant in irrespective of change in viscosity of the circulated fluid. The difference in pressures between the operating pressure fluid and the differential high pressure fluid will vary, however, with change in viscosity of the circulated fluid and the spring 55' is so made such that the drop across the orifice 70, and hence the differ-
ence in pressures of the operating pressure fluid and differential high pressure fluid, is suf-
ficient to provide satisfactory action of the motor vanes 17 when the pressure drop across the orifice 70 is minimized for the particular fluid em-
ployed. This provides satisfactory operation of the motor at all times and with all viscosities of the fluid.

It may here be noted that only one compensating circuit is required for use in connection with plurality of modified control valve means 50' employed in a fluid system, as the compensating cylinders 79 and 80 of each of said modified control valve means 50' may be connected with the inlet and outlet sides respectively of the same compensating orifice 63.

In the embodiments illustrated in Figs. 1 and 2 the variable orifice 70 and control valve means for each motor B are positioned in the supply line in advance of the motor. It is essential that the variable orifice 70 be positioned in advance of its corresponding motor B in order that it may provide the differential pressure relation be-
tween the differential high pressure fluid and the operating pressure fluid but the fluid flow or control valve means may be located at other points in the fluid system. For example, the control valve means may be positioned in the branch of the discharge conduit 43 leading from the corresponding motor B as illustrated in Fig. 3 and either the control valve means 50 of Fig. 1 or the modified control valve means 50' of Fig. 2 may be employed at this location in the fluid system.

The control valve means 50 illustrated in Fig. 3 is identical with the control valve means 50 of Fig. 1, the difference between the arrange-
ments of these two figures being solely in the po-

The valve piston 60 moving responsive to the pressure drop across the orifice 70 to regulate and control the volume of fluid that is permitted to pass therethrough the variable orifice 70 is regulated and maintained substantially constant at an amount producing a pressure drop across the orifice 70 equal to the predetermined drop thereacross set by the spring 55. In the arrangements of both Figs. 1 and 3, therefore, the control valve means 50 functions to control the speed of the corresponding motor B and to maintain the required differ-
ence in pressures between the differential high pressure fluid and the operating pressure fluid. In the arrangement of Fig. 1 this control is ef-
fected by varying the resistance to flow from the source of pressure fluid to the motor B, whereas in the arrangement of Fig. 3 control is effected by varying the resistance to discharge of fluid by the motor B which in turn regulates the vol-
ume of fluid permitted to pass into said motor B through the corresponding orifice 70.

Each of the embodiments shown and described thus provides accurate and dependable means for regulating the speed of the motor B, or motors B, and one of them provides corrective action to compensate for all variations in the motor's speed except the variation due to the leakage or "slip" of fluid past the rotor 15 and vanes 17 from the motor's inlet areas to its outlet areas, which is usually small. The same means em-
ployed for controlling the speed of the motor B is likewise utilized to provide a pressure drop in the supply line whereby the difference in pres-
sures is obtained between the differential high pressure fluid supplied to the inner ends of the motor vanes 17 to urge them into contact with the vane track 26 and the operating pressure fluid supplied to the outer ends of the vane 17 to cause rotation of the rotor 15, with differ-
ence in pressures is essential for satisfactory opera-
tion of the motor as hereinbefore explained.
A simple, economical arrangement is thus provided in which the several parts cooperate to pro-
vide satisfactory operation at the desired speed, with many attendant advantages, some of which have already been mentioned.

As previously stated, each embodiment of the invention described herein may be used in a fluid system including either one or a plurality of vane type fluid motors. Different embodiments of the invention may, however, be employed in the same fluid system if desired; for example, the modified control valve means 50' of Fig. 2 may be employed in a fluid system that also employs the control valve means 50 according either to the arrangement shown in Fig. 1 or in Fig. 2 or both of them.

It will be understood that the several embodi-
ments of my invention have been described for the purpose of illustrating the operation and con-
struction of the apparatus of my present invention and that changes, some of which have been indicated, may be made without departing from the spirit of the invention.

I claim:

1. In a fluid pressure power transmission sys-
tem, a fluid pressure supply line, resistance means in said supply line, a vane motor having a rotor provided with a plurality of vane movable in-
wardly and outwardly thereof, said vane motor also having a vane track to guide said vane in their in and out movement, a discharge line for the exhaust of fluid from said vane motor, means for sup-
plying pressure fluid from the inlet side of said resistance means to one of the ends of said vane to urge said vane into contact with said vane track, means for simultaneously supplying pressure fluid from the outlet side of said resis-
tance means to the other ends of said vane to
cause rotation of said rotor and motor speed control means responsive to the difference in pressures on the inlet and outlet sides of said resistance means, said motor speed control means including a valve bore having inlet and outlet ports connected in one of said lines and an element movable responsive to the difference in pressures on the inlet and outlet sides of said resistance means active to regulate the volume of fluid entering said inlet port.

2. In a fluid pressure system comprising a vane type fluid motor and a supply line connected thereto, said vane motor having a vane track and a rotor provided with a plurality of vanes movable inwardly and outwardly thereof, said vanes being urged into operating position in contact with said vane track at least in part by fluid pressure means, in combination, means for controlling the operation of said motor including resistance means in said supply line for producing a pressure drop whereby two pressures are obtained in the portions of said supply line at opposite ends of said resistance mechanism; means supplying fluid having the higher of said pressures to one of the ends of said vanes to urge them into operating position in contact with said vane track and valve means in said supply line having an element responsive to the pressure drop across said resistance means and active to regulate the volume of fluid passing therethrough.

3. In a fluid pressure system comprising a vane type fluid motor having a supply line and a discharge line connected thereto, said vane motor having a vane track and a rotor provided with a plurality of vanes movable inwardly and outwardly thereof, said vanes being urged into operating position in contact with said vane track at least in part by fluid pressure means, in combination, means for controlling the operation of said motor including resistance means in said supply line for producing a pressure drop whereby two pressures are obtained in the portions of said supply line at opposite ends of said resistance mechanism; means supplying fluid having the higher of said pressures to one of the ends of said vanes to urge them into operating position in contact with said vane track and valve means in said discharge line having an element responsive to the pressure drop across said resistance means and active to regulate the volume of fluid passing from said motor.

4. In a fluid pressure system, a fluid pressure supply line, an orifice in said line, a vane motor having a rotor provided with a plurality of vanes movable inwardly and outwardly thereof, said vane motor also having a vane track to guide said vanes in their in and out movement, means for supplying pressure fluid from the inlet side of said orifice to one of the ends of said vanes to urge them into contact with said vane track, means for simultaneously supplying pressure fluid from the other end of said orifice to the other ends of said vanes to cause rotation of said rotor, and means responsive to the difference in pressures on the inlet and outlet sides of said orifice to hold substantially constant at a predetermined value the difference in pressures on the inlet and outlet sides of said orifice active to regulate the volume of fluid passing therethrough to hold substantially constant at a predetermined value the difference in pressures on the inlet and outlet sides of said orifice.

5. In a fluid pressure system, a fluid pressure supply line, an orifice in said line, a vane motor having a rotor provided with a plurality of vanes movable inwardly and outwardly thereof, said vane motor also having a vane track to guide said vanes in their in and out movement, means for supplying pressure fluid from the inlet side of said orifice to the other ends of said vanes to cause rotation of said rotor, and means responsive to the difference in pressures on the inlet and outlet sides of said orifice to hold substantially constant at a predetermined value the difference in pressures on the inlet and outlet sides of said orifice active to regulate the volume of fluid passing therethrough to hold substantially constant at a predetermined value the difference in pressures on the inlet and outlet sides of said orifice.

6. In a fluid pressure power transmission system, a fluid pressure supply line, resistance means in said supply line, a vane motor having a rotor provided with a plurality of vanes movable inwardly and outwardly thereof, said vane motor also having a vane track to guide said vanes in their in and out movement, means for supplying pressure fluid from the inlet side of said resistance means to one of the ends of said vanes to cause rotation of said rotor, motor speed control means responsive to the difference in pressures on the inlet and outlet sides of said resistance means, means for simultaneously supplying pressure fluid from the outlet side of said resistance means to the other ends of said vanes to cause rotation of said rotor, and viscosity means cooperating with said last named means to modify the value of the pressure drop to be maintained across said orifice conformably with change in the viscosity of the circulated fluid.

7. In a fluid pressure power transmission system, a fluid pressure supply line, resistance means in said supply line, a vane motor having a rotor provided with a plurality of vanes movable inwardly and outwardly thereof, said vane motor also having a vane track to guide said vanes in their in and out movement, means for simultaneously supplying pressure fluid from the outlet side of said resistance means to one of the ends of said vanes to cause rotation of said rotor, motor speed control means responsive to the difference in pressures on the inlet and outlet sides of said resistance means, means for simultaneously supplying pressure fluid from the outlet side of said resistance means to the other ends of said vanes to cause rotation of said rotor, and viscosity means cooperating with said last named means to modify the value of the pressure drop to be maintained across said orifice conformably with change in the viscosity of the circulated fluid.

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