April 26, 1966 H. D. FANSHAWE 3,247,802

ROTARY PUMPS AND MOTORS

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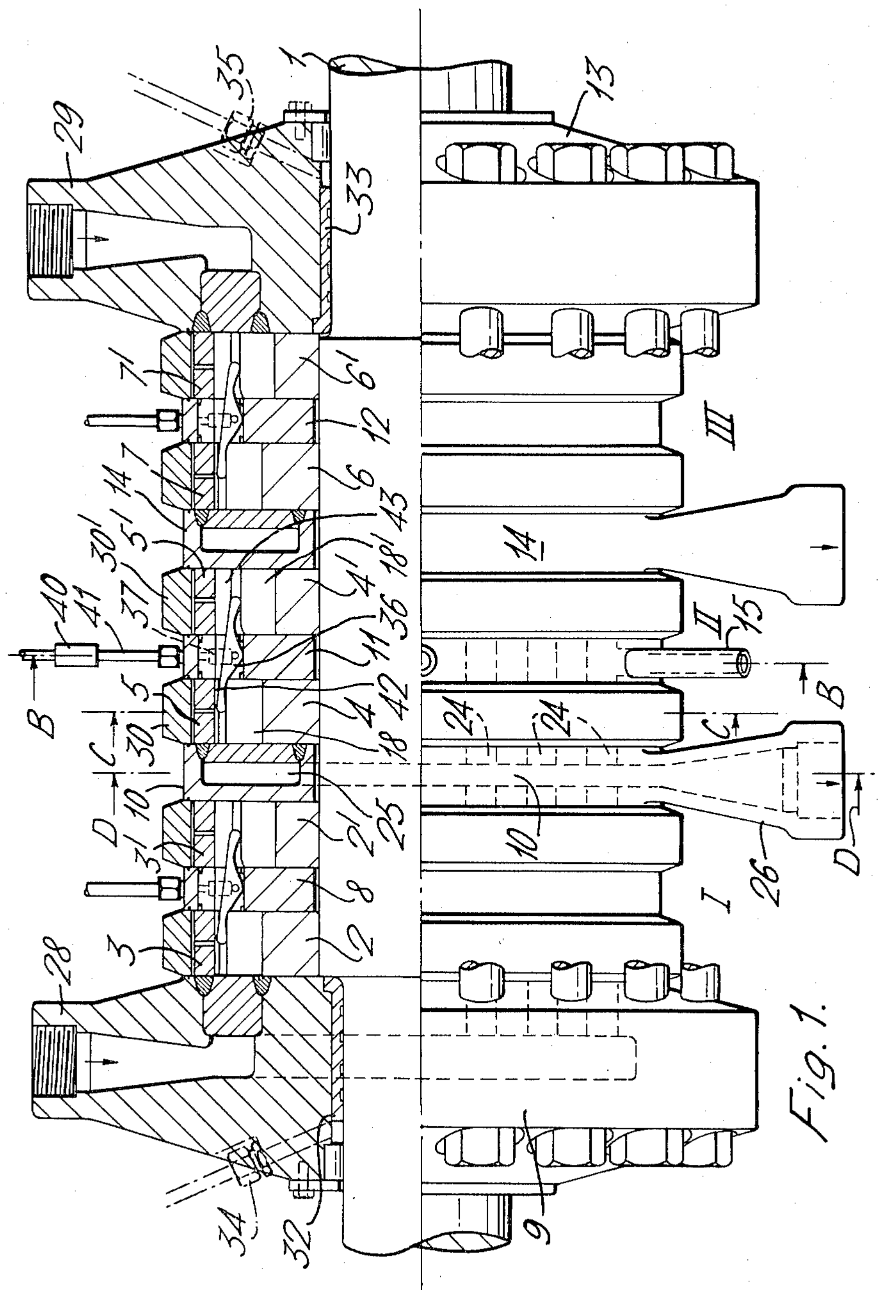

Fig. 1.

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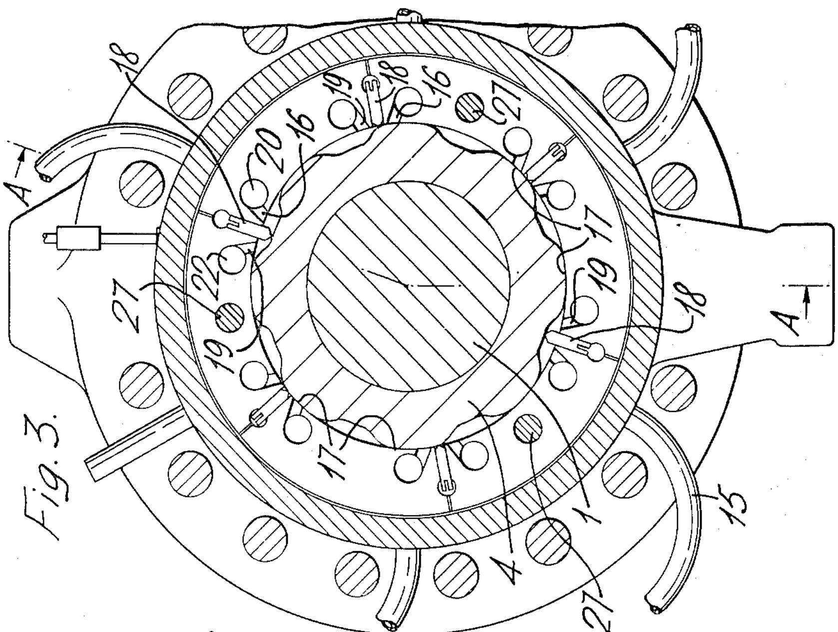

Fig. 3.

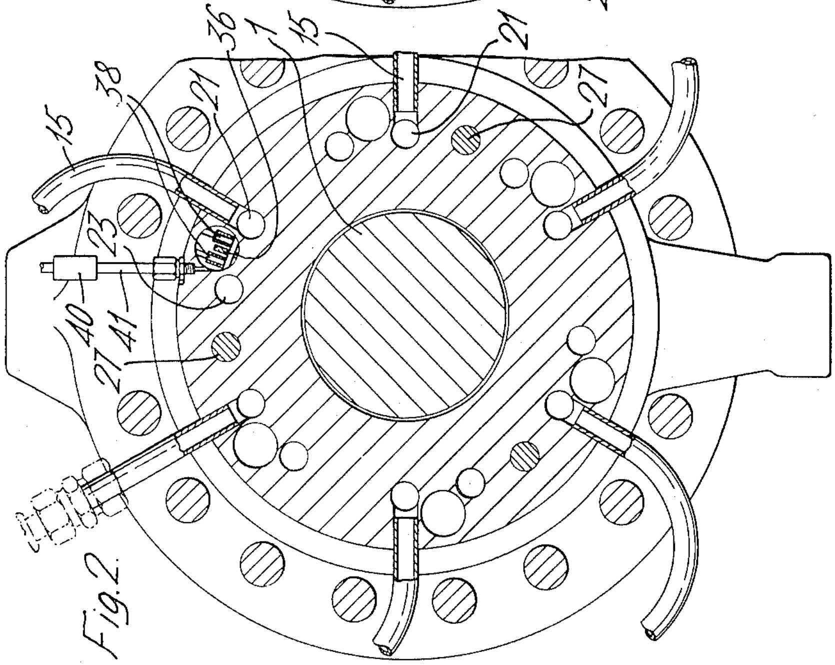

Fig. 2.

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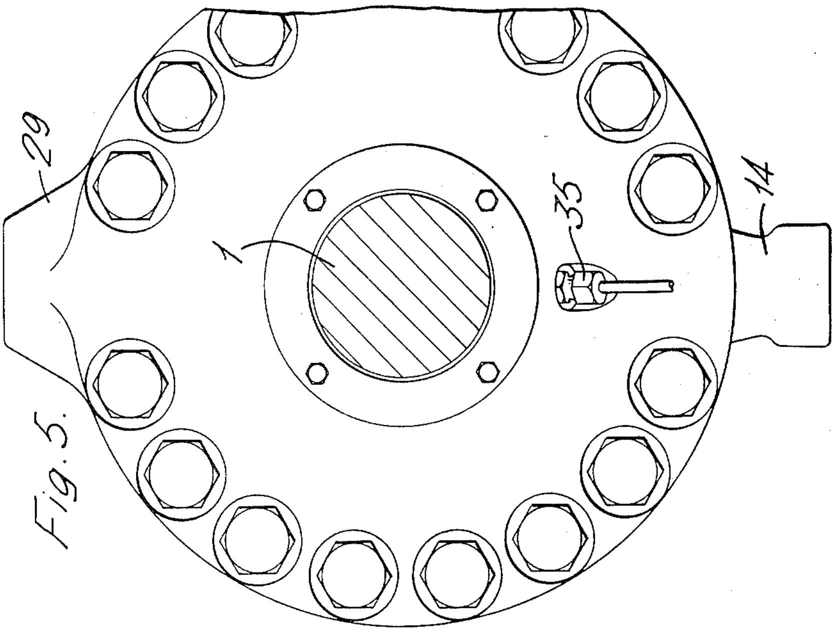

Fig. 5.

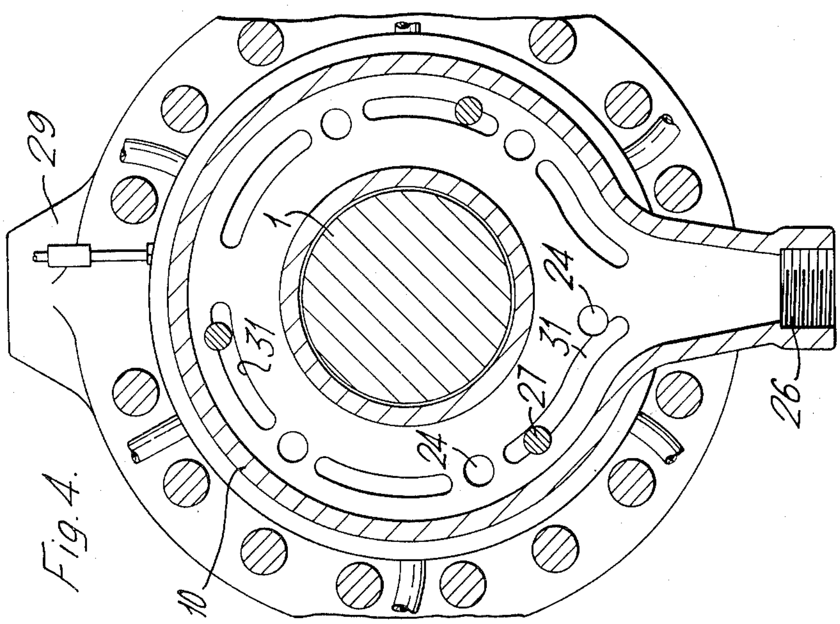

Fig. 4.

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ROTARY PUMPS AND MOTORS

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This invention relates to rotary hydraulic motor means and concerns in particular such motor means for operation under a number of different conditions of loading from a source of limited output. It will be appreciated that the invention is also applicable to rotary pumps where alternative quantities of fluid are required to be delivered.

It happens that in certain engineering projects, for instance in connection with deep mining or earth borehole drilling, rotary gear is required for a number of different purposes. Thus in borehole drilling it is desirable to provide a, so-called, power swivel for rotating the drilling string for the drilling operation, the swivel being used also for other operations, some less and some more onerous than drilling. Heretofore it has been difficult without providing change speed gear arrangements to enable a swivel to provide the widely different torque requirements for these different operations, because some limit has to be placed on the size of motor means that can be used; it may be for economic reasons, or merely from the point of its sheer size. Such gear arrangements have added to the cost, dimensions and weight of the motor means, and an object of the present invention is to provide an alternative power means which, or the use of which, obviates the necessity for such gear.

It is a common feature of the projects mentioned above, that supplies of high pressure fluid are required for certain essential operations and the present invention is also directed to enabling such high pressure fluid supplies to be used for the power means.

According to one feature of the invention hydraulic rotary power means is provided which comprises a plurality of stages that are adapted to be independently supplied with actuating fluid, each stage having at least one rotor unit associated with the or a driving shaft of the said power means and one or more co-operating stator units, each said co-operating rotor and stator units being relatively contoured to provide compartments to which actuating fluid is adapted to be fed and having one or more vane members between the or each co-operating stator and rotor, said vane members being adapted to co-operate with said compartments in operation to provide reaction to said fluid for rotating its associated rotor, whereby said shaft may be rotated by supplying fluid to one or another said stage, or a combination of said stages.

The said stages may be substantially identical with each other, or one or more may be such as to give rise to greater torque than one or more other stages.

By providing for selection of stages for rotating the shaft, the effect given is substantially the same, under suitable conditions, as that of introducing, as described above, change speed gear mechanism between the motor and an output shaft. It will be evident, however, that the present proposal can yield a power means which is substantially smaller and more compact than heretofore; besides, the use of valves which control the supply of fluid to selected stages, is far simpler than the use of a change speed gear mechanism which probably also requires the provision of clutch means.

One hydraulic motor unit embodying the invention will now be described, by way of example, with reference to the accompanying drawings of a high pressure hydraulic swivel motor for service with a borehole drilling rig, the motor being arranged to provide power for a number of different duties, such as referred to above.

FIGURE 1 of the drawings shows a part-sectional side elevation of the motor; FIGURE 2 a section on the line B—B of FIGURE 1, FIGURE 3 a section on the line C—C of FIGURE 1, FIGURE 4 a section on the line D—D of FIGURE 1 and FIGURE 5 and end view of the motor. FIGURE 1 is a part-section on the line A—A in FIGURE 3.

In the drawings, three stages I, II and III of rotor and stator units are provided on a common shaft **1** (FIGURE 1). Each stage comprises a pair of rotors and co-operating stators, designated **2** and **3**, **2'** and **3'** in stage I, **4** and **5**, **4'** and **5'**, in stage II and **6** and **7**, **6'** and **7'** in stage III; all the rotors are keyed to the shaft **1**.

Stators **3** and **3'** are separated by a sandwich piece **8** and are clamped between an inlet manifold **9** for operating fluid and an exhaust manifold **10** by means of bolts referred to below; the clearances between the sandwich piece and the manifold walls and the rotors are arranged to be such as to prevent excessive leakage when operating under maximum fluid pressure.

Similarly, sandwich pieces **11** and **12** are provided for stages II and III respectively. The input manifold for stage III is indicated at **13** and the exhaust manifold at **14**. Stage II exhausts into the manifold **10** (possibly also into manifold **14**), the input arrangements for this stage comprising individual feed tubes **15** with entries spaced around the periphery, there being six feed tubes in all (FIGURE 2).

Each unit, rotor and stator, is substantially identical with the others and in each stage, the two rotors are staggered in relation to each other and are shaped to provide substantially constant torque and speed of the motor during each revolution, given a constant supply of operating fluid.

Thus in FIGURE 3, rotor 4 is so shaped as, in co-operation with stator **5**, to provide eight compartments, as shown, into which operating fluid is arranged to be successively fed through inlet ports **16** as each of the eight lobes **17**, which are dimensioned to give close clearance from the inner wall of the stator, is urged past one of the six vanes **18** by reaction of the pressurised fluid against the vanes in other compartments. Fluid trailing the vanes is forced by the backs of the vanes into the exhaust ports **19**.

The inlet ports **16** are fed from supply ducts **20** through the stator **5** and ducts **21** through the sandwich piece **11**, the ducts **21** being aligned with one end of the ducts **20**; and similar ducts **20'** in the stator **5'** are aligned at one end with the other ends of ducts **21**. The ducts **21** are in individual communication with a supply manifold (not shown) through pipes **15** and valve means are arranged to connect or disconnect the supply of operating fluid to the pipes **15**; the other ends of ducts **20** and **20'** are blocked by the end plates of the manifolds **10** and **14** respectively. Alignment is secured by means of three spigots **27**.

The exhaust ports **19** lead to ducts **22** which pass through the stator **5** and align with ducts **23** and **24** in the sandwich piece **11** and an end wall of the manifold **10**, respectively; similar exhaust ducts in stator **5'** align with the other ends of ducts **23** in the sandwich piece and with ducts **24'** in the end wall of the manifold **14**.

All the ducts **24** in the manifold **10** communicate with the annular duct **25** in the manifold, and this leads to a single point take off **26** which is arranged to be connected to a suitable return line for the operating fluid. Similar arrangements are made in respect of manifold **14**.

The torque transmitted to the shaft will depend upon the operating pressure of the fluid and, to be able to make use of comparatively high pressure supply, say, 5,000 p.s.i., and yet to keep the size of the motor within acceptable limits, each stator is enveloped in a pressure ring, such as at **30** and **30′** for stage II of the motor. These pressure rings are of high tensile steel and serve to enclose a pressurised space surrounding the stators to counteract straining of the stators, thereby enabling their thickness to be kept down to reasonable dimensions. Moreover the end walls of the manifolds in contact with the stators are made of thickness sufficient to resist undue straining when under the influence of high pressure fluid which seeps between the rotors and the end walls, and therefore to prevent excessive leakage of the fluid. In addition, in view of the somewhat long unsupported area of the annuli of the manifolds, suitable spacers are provided in the annuli (designated **31** in annulus—see FIGURE 4).

There will inevitably be a certain degree of leakage towards the shaft, on account of the clearances which are required to permit rotation of the rotors, and the bearings **32**, **33** at each end of the shaft are provided with labyrinth seals to minimise this. To avoid excessive build-up of pressure in the bearing cases, however, the leakage fluid is drained through connections **34**, **35**; these connections may lead to a tank at atmospheric pressure, in spite of the fact that the exhaust from the motor may be to a main at an elevated pressure for reasons connected with other uses of the fluid supply system.

By supplying operating fluid to the manifold that feeds the pipes **15**, the rotors **4** and **4′** are energised to rotate the shaft. To rotate the shaft by means of rotors **2** and **2′**, operating fluid is supplied to the inlet **28** of manifold **9**, and to rotate it by means of rotors **6** and **6′**, operating fluid is supplied to inlet **29** of manifold **13**. If operating fluid is supplied to all three stages, then, all stages being substantially identical, three times the quantity of fluid is required at a given pressure, compared with that required for operating one stage at the same pressure.

It will be evident, therefore, that if the supply of fluid is limited, different combinations of torque and speed of rotation can nonetheless be obtained at the limit of output of the supply, by selecting the number of stages of the motor to be energised.

Thus, for instance, it may be required to provide a torque of 36,000 lbs.-ft. for one operation, and to maintain a speed of 300 r.p.m. for other operations of a power motor. Consider using a single stage rotor hydraulic motor for this purpose: when operating at maximum pressure and maximum capacity, it would be necessary (to attain both requirements at the same time and to produce that torque at the given rate of rotation) for the supply to provide about 2,100 horsepower. If the power available at the supply is limited to, say, 350 horsepower, then only one-sixth of the required power is available, and the requirements could not be met simultaneously; nor could they be met separately unless the 350 horsepower were coupled to a pump able to provide the same throughput capacity and the same pressure. If, however, the high torque were not to be required at more than a fraction of the maximum (300 r.p.m.) speed of rotation, and the torque required on the motor at the higher speeds were to be similarly a fraction of the maximum 36,000 ft. lbs. so that the 350 horsepower represents an adequate power supply, the use of a gear box would enable both the maximum torque and the maximum speed to be obtained for appropriate operations, when coupled to a pump with a fraction of the above maximum capacity.

As stated above, however, the use of the gear box leads to complications which may be unacceptable, or at least inconvenient, in certain cases and, by use of the present invention, the same effect may be produced by selection of the number of stages of the motor to supply power. With all three stages supplying power, it may be arranged that the high torque requirement is attained at the expense of rate of rotation, while with only one stage operating, the

speed of rotation is maintained but the torque available is considerably reduced.

It is evident that the invention is also applicable to pumps because the motor means described will act as a pump if powered by suitable means, and in that case similar arguments apply. Thus the pump can be arranged to provide pressurised fluid under selected conditions for variable conditions of drive. It is understood therefore, that the scope of the invention shall include pumps.

It is of course not essential that the stages of a motor or pump in accordance with the invention should be similar in design and in size. Thus in alternative construction a similar set of requirements as are fulfilled by the construction described above may be met by providing only two stages but by making one stage a fraction, say about two-thirds, the size of the other. Then these stages may be arranged to be used alternatively simply to provide two different conditions of operation and in parallel to provide a third condition. It will be evident, too, that each stage need not be duplex as in the design shown in the accompanying drawings; other arrangements may be made for the attainment of substantially uniform torque and speed, which is the reason for the adoption of the duplex arrangements.

A possible snag arises in connection with an hydraulic motor in accordance with the invention, when a stage is cut off from supply of operating fluid. Even if the supply connection of the disconnected stage is reconnected to exhaust to prevent cavitation, its rotor still has to rotate with the shaft; a disadvantage which must arise from excessive churning of the residual fluid in the stage is overcome by use of a second feature of this invention.

According to another feature of the invention, provision is made in a rotary pump or motor of the kind hereinbefore specified, for vane withdrawal whereby the rotor(s) may idle without excessive churning of fluid that may be trapped in the compartments.

In a hydraulic pump or motor of the kind wherein the rotor is shaped to provide two or more compartments in co-operation with the stator, the compartments being, in effect, swept by vanes sliding in the stator and acting between the rotor and the stator, positive pressure means being provided to bias said vanes into contact with the rotor, in accordance with yet another aspect of the invention, ancillary means are provided whereby the outer edges of the vanes may be relieved of high pressure fluid. Then, in order that the vanes shall be forced into their slots to allow movement of the trapped fluid past the vanes, valve means on the exhaust side (instead of the pressure side) of the stage is shut, enabling pressure to build up in the stage, and the said ancillary means, which may be a vent valve, is operated to relieve the pressure from behind the outer edges of the vanes, so that the pressurised fluid in the compartments then forces the vanes outwardly into their slots. Once in the slots, the fluid in the stage, being pressurised, maintains them there, until the vent valve is closed again when the stage is to be coupled.

Special arrangements frequently have to be made for the vanes positively to be forced against the rotor, at least initially, and in this case a force is applied to the vanes **18** (reference is here made to stage II by way of example) by means of a rocker arm **36** under the influence of springs **37**, **38**; then, once powered rotation has been established, additional hydraulic pressure is obtained from pressurised fluid which is bled to the outer edges of the vanes from the high pressure side of the rotor/stator compartments. This bleed occurs naturally up the high pressure face of each vane, due to the attitude which the vanes adopt in the slots when under load. Such high pressure, acting on the edges of the vanes, provides the necessary seal between their inner edges and the rotor. A pressure differential could not, however, build up internally across the vanes if the basic biasing pressure provided by the springs were not available at the commencement of feed of operating fluid to the stage; but if the vanes are maintained in contact with the rotor when the feed is cut off and the feed port connected to exhaust the vanes will force the liquid trapped in the compartments of the stage backwards and forwards through the inlet and outlet ports.

In accordance with the other aspect of the invention therefore, a vent valve is provided, comprising a valve **40** which is included in a pipe **41** leading from the spaces **42** and **43**, at the back of the vane slots, to a tank at atmospheric pressure.

It will be appreciated that this feature of the invention relating to withdrawal of vanes may be applied to hydraulic pumps in a similar manner. In that case, however, it would have to be the inlet of the pump or the stage which is closed to enable sufficient pressure to build up on the pump or in the stage to move the vanes against the action of the biasing springs when the venting valve was open.

Alternatively, if the motor has a pressurized exhaust, the supply may be disconnected, and if the pump has a pressurized suction, the delivery port may be disconnected, in both cases the vent valve from the back of the vane slots being opened to tank; then the differential between exhaust and tank pressures, or between suction and tank pressures, respectively, must be arranged to be sufficient to force the vanes back into their slots, so that the rotor(s) may turn freely.

I claim:

1. Rotary hydraulic motor or pumping means comprising a shaft, a plurality of independently operable stages associated with said shaft, each stage including at least one rotor unit fixed to and rotatable with said shaft and at least one co-operating stator unit, each pair of co-operating rotor and stator units being relatively contoured to provide compartments therebetween which are adapted to receive hydraulic fluid, at least one vane member extending radially between the co-operating stator and rotor units of each stage, each vane member being adapted to co-operate with the compartments of the associated stage during rotation of the rotor of that stage, and means for independently controlling the flow of hydraulic fluid to and from each of said stages, whereby said stages may be brought into use singly or in combination to provide different conditions of operation.

2. Rotary hydraulic motor or pumping means as claimed in claim **1**, wherein said stages are substantially identical in capacity to each other.

3. Rotary hydraulic motor or pumping means as claimed in claim **1**, comprising three stages of substantially identical capacity, said control means being so operable that only one stage is in use, or that two stages or all three stages are in use in parallel, whereby three different conditions are available for operation of the motor or pumping means.

4. Rotary hydraulic motor or pumping means in accordance with claim **1**, comprising two stages substantially different in capacity from each other, whereby two different conditions of operation may be made available by operating each stage separately and a third condition by operating both stages in parallel.

5. Rotary hydraulic power means comprising an output shaft, a plurality of independently operable, hydraulically actuated stages associated with said shaft, each stage including a rotor unit fixed to and rotatable with said shaft and a co-operating stator unit, the co-operating rotor and stator units of each stage being so contoured as to provide compartments therebetween which are adapted to receive pressurized hydraulic fluid, at least one vane member for each pair of rotor and stator units, said vane member being arranged to extend radially between its stator unit and its rotor unit and to be reactive to hydraulic fluid in said compartments, and means for independently controlling the flow of pressurized hydraulic fluid to and from each of said stages, whereby said output shaft may be powered under different conditions of speed and/or torque.

6. Rotary hydraulic pumping means comprising a driving shaft, a plurality of independently operable hydraulic pumping stages associated with said shaft, each of said stages including a rotor unit fixed to and rotatable with said shaft and a co-operating stator unit, the co-operating rotor and stator units of each stage being so contoured as to provide a plurality of compartments therebetween adapted to receive hydraulic fluid, at least one vane member for each pair of rotor and stator units, said vane member being arranged to extend radially between its stator unit and its rotor unit, whereby when power is applied to said driving shaft said vane member causes fluid in any of said compartments to be forced out of the compartment, and means for independently controlling the flow of hydraulic fluid to and from each of said stages, whereby said stages may be brought into use singly or in combination so that said fluid may be delivered by the pumping means under selected output conditions.

7. Rotary hydraulic motor or pumping means of the kind in which at least one rotor unit co-operates with at least one stator unit to provide at least two compartments that are adapted to receive hydraulic fluid and in which at least one vane member is arranged to bridge the gap between said rotor and said stator units so as, in effect, to sweep said compartments in turn, comprising means for causing withdrawal of said vane member from said compartments whereby the rotor may idle without excessive churning of fluid that may be trapped in the compartments.

8. Rotary hydraulic motor or pumping means of the kind in which a rotor is shaped to provide in co-operation with a stator a plurality of compartments adapted to receive pressurized hydraulic fluid, and in which the compartments are, in effect, swept by at least one vane sliding in the stator and acting between the rotor and the stator, the outer edge of said vane extending into a space to which pressurized fluid is normally supplied so as to exert pressure behind said outer edge, comprising positive pressure means for biasing said vane into contact with the rotor, and ancillary means for relieving the outer edge of said vane of the pressure of said pressurized fluid.

9. Rotary hydraulic motor or pumping means as claimed in claim **8**, wherein said ancillary means comprises a duct connected to the fluid-containing space behind the outer edge of said vane, and a vent valve in said duct.

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