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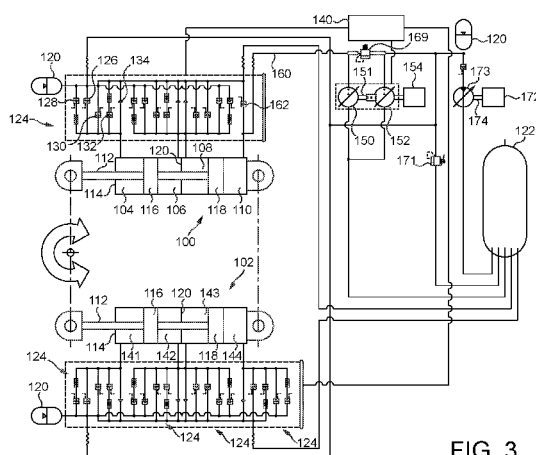


FIG. 3

(57) Abstract: The invention relates to a hydraulic fluid power transmission comprising: actuator apparatus having a plurality of chambers, each of the chambers having a respective fluid driving surface configured to drive or be driven by hydraulic fluid therein; a discretised pressure control system configured to selectively connect one or more first chambers of the said plurality of chambers to one of a plurality of hydraulic fluid sources or sinks, at least two of the plurality of hydraulic fluid sources or sinks having different fluid pressures; a continuous pressure control system configured to control the pressure of hydraulic fluid, or a flow rate of hydraulic fluid, input to or output from one or more second chambers of the said plurality of chambers, the pressure or flow rate of the hydraulic fluid input to or output from the second chambers being thereby variable within a range of pressures or flow rates respectively; and a controller configured to control the discretised and continuous pressure control systems to thereby regulate a property of the actuator affected by the hydraulic fluid pressure in the said chambers.



1 HYDRAULIC FLUID POWER TRANSMISSION

2
3 Field of the invention

4
5 The invention relates to a hydraulic fluid power transmission and a method of
6 transmitting hydraulic fluid power.

7
8 Background to the invention

9
10 The load applied by hydraulic actuators can be controlled in two main ways:
11 continuous smooth pressure control to the actuator or actuators; or switching the total
12 area of an actuator or actuators between high and low pressure to produce discrete
13 or “quantised” load steps (rather than a smoothly varying function).

14
15 A straightforward and commonly used means of continuously controlling the pressure
16 applied to an actuator is by using a proportional control valve to regulate the pressure
17 in the actuator chamber directly. Fluid is pumped up to a high pressure and stored in
18 an accumulator. When the actuator is to be driven a proportional control valve is used
19 to throttle much of this pressure away to produce an intermediate pressure that is
20 applied to the actuator to drive it at a prescribed load or speed. When the actuator
21 must resist motion a control valve throttles pressure generated in the actuator back to
22 low pressure reservoir. While being simple and potentially permitting high
23 instantaneous powers this approach is of course very inefficient and wasteful as

1 control can only be effected in either mode by dissipating excess pressure energy as
2 heat and noise.

3
4 Another common method is to use a closed hydraulic circuit where a single
5 bidirectional hydraulic pump-motor (operable to function as a pump or a motor in
6 different operating modes) is used to alternately take fluid from one side of a
7 bidirectional actuator and pump it into the other. This allows for continuous and high
8 bandwidth control of load and motion but instantaneous power is limited to that of the
9 pump-motor, no energy storage or smoothing is possible, and while usually better
10 than the purely dissipating system above (in most circumstances) part load efficiency
11 is poor due to the full instantaneous power rating of the pump-motor machine and the
12 fact that it must run continuously.

13
14 A third method, illustrated in Figure 1, uses a pair of pump-motors 10, 12 on a
15 common shaft 14 which creates a true continuously variable transmission with energy
16 storage. One pump-motor 10 exchanges fluid between a high pressure energy store
17 16 and a low pressure reservoir 18, under control of its displacement, to provide a bi-
18 directional torque to the common shaft 14. The other pump-motor 12 can then use
19 this torque to transmit flow bi-directionally to or from the actuator 20. Thus power can
20 be transmitted between the actuator 20 and the energy store 16. A hydraulic pump or
21 motor 21 coupled to a prime mover or generator 22 may be provided in
22 communication with the energy store 16 to thereby provide an overall power source
23 or sink of the average power requirements of the system. This system allows for
24 power exchange and storage and a much smaller prime mover or generator rating but
25 suffers from the instantaneous power and part load efficiency limits of the rotating
26 machines.

27
28 As well as avoiding the need to rate the prime mover or generator for the
29 instantaneous power requirements (as does the quantised system described below),
30 this system allows continuously variable transmission of flow, load and power without
31 the discrete steps of a quantised system. On the other hand, while common in
32 mobile and industrial applications where cost and efficiency is not a primary concern
33 (for example in offshore lifting equipment, or in large earth moving equipment),
34 conventional fluid power transforming systems would have fundamental drawbacks
35 that prevent their useful application in renewable energy applications where the high
36 peak-to-average power and flow ratios mean that the high capital costs and high
37 losses at part load would be unacceptable given the wider economic drivers.

1
2 Quantised transmissions typically use digitally controlled fast acting valves to actively
3 pressurise and de-pressurise discrete actuator areas. This alters the aggregate area
4 over which a nominally steady pressure is applied to vary the total applied force in a
5 series of steps rather than as a smooth function. This is known as a 'quantised'
6 system and the process of approximating the desired continuous system or function
7 with a series of steps is known as 'quantisation'. In hydraulic transmissions such an
8 approach provides for rapid control of force with relatively very efficient power
9 transmission between the actuators and a local energy store. The working pressure
10 in the energy store may be controllable via a secondary transmission
11 (generation/supply) system.

12
13 Quantised approximation to continuous control can be achieved by selecting from the
14 available discrete load steps in real time. A quantised fluid power transmission uses
15 digitally controlled hydraulic valves to admit flow between the individual areas of the
16 load bearing actuators (e.g. either side of hydraulic cylinder pistons or between the
17 ports of rotary hydraulic actuators) and the high or low pressure parts of the circuit.
18 Hence the net forces applied by each actuator area, and hence the load as a whole,
19 can be controlled in real time by combining different chamber pressures in sequence
20 with accurate timing.

21
22 An example of a quantised transmission is shown in Figure 2 for the case of a simple
23 fixed displacement rotary actuator 30. In this example, single actuator chambers of
24 an actuator 30 operate in opposing directions, the actuator chambers being
25 selectively connected to a high pressure energy store 32 or a low pressure reservoir
26 34 by way of respective pairs 36, 38 of electronically controlled low and high pressure
27 valves 40, 42. The valves may be switched open or closed to admit flow to or from
28 the actuators from or to the high or low pressure galleries to which they are
29 connected to control the pressure in each actuator chamber. There are three possible
30 load states of the actuator 30: zero; and one step in either direction, the magnitude of
31 the load steps being a function of the relative actuator area (e.g. the area of a fluid
32 driving surface of a piston of the actuator) and the step pressure applied. Additional
33 actuators can be added in parallel together to increase the available load and number
34 of load steps. The greater the number of steps the closer the stepped load profile will
35 approximate a continuous load control function (but the greater the number of
36 components, overall complexity and cost).

37

1 The quantised transmission may be hydraulically coupled (e.g. by way of energy
2 store 32) to a hydraulic pump or motor 43 coupled to a prime mover or generator 44,
3 thereby providing an overall power source or sink to the system.

4
5 For any applied load the absorbed power passes to and from the energy storage
6 system without passing through any finite displacement and inefficient rotating
7 machines. The flow only has to pass through the switched inlet and outlet valves and
8 any plumbing connections. These can be sized to accept effectively unlimited flow
9 potential at very little cost and complexity. The quantised transmission thus provides
10 a route between the actuator(s) and the power pack (or electricity generator)
11 components through a set of digitally controlled valves with no fixed losses and
12 minimal restriction of the fluid. This offers lower losses and improved efficiency
13 compared to a design using conventional variable displacement pump/motors to vary
14 the pressure in the primary actuators. This is because the losses associated with
15 conventional variable displacement pump/motors include leakage, churning, friction,
16 and parasitic losses in the boost system that are not present in the quantised
17 transmission using digital valves to pressurise and depressurise discrete actuation
18 steps.

19
20 Furthermore, the quantised system can transmit power and flow over a range only
21 limited by the fluid resistance through the digital valves and fluid connections
22 (connections required for any system), whereas the finite rating of any variable
23 displacement pump/motor arranged in line with the actuator(s) must also limit the
24 power rating of the whole primary actuation system. The latter is a far greater
25 restriction in terms of power rating and equipment cost, size, and poor part-load
26 efficiency.

27
28 For a given power handling capability, the cost of the digital control manifolds
29 required to implement the quantised transmission are relatively low when compared
30 to variable displacement pump-motors. The quantised system has relatively flat
31 efficiency across the power range: there are zero fixed losses (except for the
32 relatively tiny power requirements of the digital control solenoids), flow losses lie only
33 in fluid transit, and volumetric losses are only significant for very small actuator
34 motions. A directly coupled variable displacement pump/motor transmission would
35 suffer excessive losses at low powers. Sizing a variable displacement pump/motor to
36 meet a given power range makes it very inefficient towards the bottom of that power
37 range. Therefore, for applications involving large ratios of peak to average power and

1 flow, such as in renewable energy power take-off or in heave compensation systems,
2 conventional systems are ill-suited.

3
4 A major drawback of the quantised approach lies in the application of discrete load
5 steps rather than a continuously variable load. Hysteresis, is required in the
6 quantisation process to avoid limit cycling (instability where the quantised system
7 switches rapidly back and forth between two states due to interaction with the step
8 load and the dynamic response of the system affecting the demand signal) around
9 transitions so at practical step sizes, the quantisation process must introduce a
10 significant phase shift from the ideal load signal to maintain stable operation. This
11 limits the performance of the load and motion control system as a whole to well below
12 that possible with a continuous load control.

13
14 The switching of pressures in the hydraulic cylinders leads to a stepped load profile
15 when a smoothly varying one would be preferable from a control point of view. The
16 steps in load can cause significant limitations to the gains that may be attained in a
17 force or motion control loop. In many applications, a quantised system is not viable
18 due to the need for absolute motion control, the need to hold static loads, and/or the
19 requirement to smoothly control the motion of elements with low inertia. Continuously
20 varying the pressure in the hydraulic actuators is a necessary or attractive feature for
21 the vast majority of applications. The only way to achieve this with a digital quantised
22 system alone would be to have a very large number of very small actuator load steps
23 which will not be attractive or practical for many applications.

24
25 For the vast majority of applications to date, continually varying the pressure applied
26 to the actuator has been used in preference to the step changes that are present in
27 the quantised transmission. However, a digitally controlled quantised system offers
28 major advantages in efficiency of power transmission and in power and flow capacity
29 for a given cost and size of equipment. Although quantised transmission offers
30 potentially greater efficiencies than the continuous transmission it cannot be used for
31 a number of applications due to its stepped output.

32
33 An improved hydraulic transmission would therefore be desirable.

1 Summary of the invention

2

3 A first aspect of the invention provides a hydraulic fluid power transmission
4 comprising: actuator apparatus having a plurality of chambers, each of the chambers
5 having a respective fluid driving surface configured to drive or be driven by hydraulic
6 fluid therein; a discretised pressure control system configured (e.g. programmed) to
7 selectively connect one or more first chambers of the said plurality of chambers to
8 one of a plurality of hydraulic fluid sources or sinks, at least two of the plurality of
9 hydraulic fluid sources or sinks having different fluid pressures; a continuous
10 pressure control system configured (e.g. programmed) to control the pressure of
11 hydraulic fluid, or a flow rate of hydraulic fluid, input to or output from one or more
12 second chambers of the said plurality of chambers, the pressure or flow rate of the
13 hydraulic fluid input to or output from the second chambers being thereby variable
14 within a range of pressures or flow rates respectively. The hydraulic fluid power
15 transmission typically further comprises a controller configured to control the
16 discretised and continuous pressure control systems to thereby regulate a property of
17 the actuator apparatus affected by the hydraulic fluid pressure in the said chambers.

18

19 Typically the said at least two of the hydraulic sources or sinks have different
20 pressures, there being a discrete step between their pressures.

21

22 As the one or more first chambers are selectively connected to one of a plurality of
23 hydraulic fluid sources or sinks, a direct transmission of hydraulic fluid from a
24 hydraulic fluid source to the said first chambers, or to a hydraulic fluid sink from the
25 said first chambers, is provided. This hydraulic fluid does not need to flow via any
26 finite displacement or inefficient rotating machines, which means that the
27 instantaneous powers are effectively unlimited and the energy losses are much lower
28 than for a continuously variable transmission.

29

30 As the pressure or flow rate of hydraulic fluid flowing into or out of the second
31 chamber(s) can be controlled within a range of pressures or flow rates respectively,
32 the continuously variable transmission of hydraulic fluid flow, load and power can be
33 achieved without the discrete steps of the discretised pressure control system.

34

35 By providing both discretised and continuous pressure control systems, the efficiency
36 benefits of the discretised pressure control system can be achieved without the
37 discrete steps in the pressure(s) of the first chamber(s) being propagated to the said

1 property of the actuator apparatus. This makes the hydraulic fluid transmission
2 according to the first aspect of the invention particularly suitable for handling high
3 instantaneous power flows efficiently.

4
5 Typically the controller is configured to control the continuous pressure control
6 system (and typically to control the discretised pressure control system) such that a
7 contribution of the second chambers (and typically thus the continuous pressure
8 control system) to the said property of the actuator apparatus at least partially
9 (typically continuously) compensates for a difference between a contribution of the
10 said first chambers (and typically thus the discretised pressure control system) to the
11 said property of the actuator apparatus and a value of the said property of the
12 actuator apparatus indicated by a received demand.

13
14 Typically the said received demand is a smoothly varying function of the said property
15 of the actuator apparatus. Typically the contribution of the said first chambers (and
16 typically thus the discretised pressure control system) to the said property of the
17 actuator apparatus comprises one or more steps. By compensating for a difference
18 between a value of the said property of the actuator apparatus indicated by the
19 received demand and a contribution of the said first chambers (and typically thus the
20 discretised pressure control system) to the said property of the actuator apparatus,
21 steps in the contribution of the said first chambers to the said property of the actuator
22 apparatus can be effectively smoothed, thereby allowing the said property of the
23 actuator apparatus to be more accurately regulated in accordance with the (typically
24 smooth) received demand.

25
26 Typically the hydraulic transmission comprises more said first chambers than said
27 second chambers. It may be that there are at least two (or at least three) first
28 chambers per second chamber.

29
30 By providing more first chambers than second chambers, the efficiency of the
31 hydraulic fluid power transmission is improved because discretised fluid
32 transmissions typically transmit hydraulic fluid more efficiently than continuous fluid
33 transmissions.

34
35 It may be that a single second chamber is provided.

36

1 It may be that more than one first chamber is (typically more than two first chambers,
2 more typically more than three first chambers are) provided.

3
4 The said property of the actuator apparatus may be, for example, any of the following:
5 a particular performance property (e.g. speed of movement or torque delivered to or
6 by the actuator apparatus); an output torque; an output force; a speed of rotation; a
7 speed of movement; an electrical power output; a power input or output; a linear or
8 rotary displacement; a resistance to movement of an actuator shaft; a fluid pressure
9 or rate of fluid flow to or from the actuator apparatus; or a force or pressure acting on
10 or exerted by the actuator apparatus.

11
12 The said property of the actuator apparatus is typically an output property of the
13 actuator apparatus.

14
15 The received demand is typically a demand for the said property of the actuator
16 apparatus. Typically the received demand is a received signal indicative of a
17 demanded (e.g. target) value of the said property of the actuator apparatus. For
18 example, it may be that the received demand is: a received signal indicative of a
19 demanded or target value of a performance property (e.g. speed of movement or
20 torque delivered to or by the actuator apparatus) of the actuator apparatus; a
21 received signal indicative of a demanded (e.g. target) value of an output torque of the
22 actuator apparatus; a received signal indicative of demanded (e.g. target) value of an
23 output force of the actuator apparatus; a received signal indicative of a demanded
24 (e.g. target) value of a speed of rotation of the actuator apparatus; a received signal
25 indicative of a demanded (e.g. target) value of a speed of movement of the actuator
26 apparatus; a received signal indicative of a demanded (e.g. target) value of an
27 electrical power output of the actuator apparatus; a received signal indicative of a
28 demanded (e.g. target) value of a power input or output of the actuator apparatus; a
29 received signal indicative of a demanded (e.g. target) value of a linear or rotary
30 displacement of the actuator apparatus; a received signal indicative of a demanded
31 (e.g. target) value of a resistance to movement of an actuator shaft of the actuator
32 apparatus; a received signal indicative of a demanded (e.g. target) value of a fluid
33 pressure or rate of fluid flow to or from the actuator apparatus; or a received signal
34 indicative of a demanded (e.g. target) value of a force or pressure acting on or
35 exerted by the actuator apparatus.

36

1 It may be that the received demand is derived from a manual control input from a
2 user. Alternatively, it may be that the received demand is an automatically computer
3 derived control input (e.g. a desired resistance of a hydraulic ram to movement of a
4 buoyant body member of apparatus for extracting electrical power from water waves).

5
6 It may be that the received demand is a demand for a particular performance property
7 (e.g. speed of movement or torque delivered to the actuator apparatus) of manually
8 operated hydraulic machinery comprising the hydraulic transmission and the actuator
9 apparatus.

10
11 It will be understood that the plurality of hydraulic sources and sinks may comprise
12 one or more hydraulic fluid sources and one or more hydraulic fluid sinks. The
13 plurality of hydraulic sources and sinks may comprise two or more hydraulic sources.
14 The plurality of hydraulic sources and sinks may comprise two or more hydraulic fluid
15 sinks.

16
17 Typically at least one or each second chamber has a load range greater than or equal
18 to that of at least one or each said first chamber. Typically each second chamber
19 may have a load range 10-20% greater than one or each said first chamber to ensure
20 that the second chamber has enough load capacity to correct for steps in a real
21 system with control hysteresis, pressure drops and other real system effects. It will
22 be understood that the load range of the said second chamber is the range from the
23 minimum contribution (e.g. a minimum pressurised fluid consumption or output) of the
24 second chamber to meet the demand to the maximum contribution (e.g. a maximum
25 pressurised fluid consumption or output) of the second chamber to meet the demand.

26
27 Typically the pressure of the hydraulic fluid input to or output from the second
28 chambers is variable within a range of pressures between a first pressure and a
29 second pressure. It may be that the discretised pressure control system is configured
30 (e.g. programmed) to selectively connect one or more first chambers of the said
31 plurality of chambers to one of a plurality of hydraulic fluid sources or sinks, the
32 plurality of hydraulic fluid sources or sinks comprising a hydraulic fluid sink having a
33 pressure equal to the said first pressure and a hydraulic fluid source having a
34 pressure equal to the second pressure. However, it may of course be that the
35 pressure ranges of the discretised and continuous pressure control systems are
36 different. For example the difference between the first and second pressures may be
37 greater than the pressure difference between the said hydraulic fluid source and sink.

1
2 It may be that the discretised pressure control system is configured to selectively
3 connect the one or more first chambers to one of a plurality of hydraulic fluid sources
4 or sinks to thereby selectively change the pressure(s) within the said first chamber(s)
5 between discrete pressure values (the said discrete pressure values being
6 determined by the pressures of the hydraulic fluid sources or sinks). Typically the
7 discretised pressure control system is configured to selectively change the
8 pressure(s) within the said first chamber(s) between discrete pressure values
9 discretely with respect to time (i.e. non-continuously over time).

10
11 It may be that the continuous pressure control system is configured to control the
12 pressure or flow rate of hydraulic fluid input to or output from one or more second
13 chambers of the said plurality of chambers continuously with respect to time.

14
15 It may be that the continuous pressure control system comprises an electrical
16 generator configured to convert rotation of a rotatable shaft (e.g. a rotatable shaft of a
17 variable displacement pump-motor or of a pair of variable displacement pump-
18 motors) of the continuous pressure control system into electricity (in order to take
19 power out of the transmission) and/or a prime mover (e.g. electric motor or engine
20 such as an internal combustion (e.g. diesel powered) engine) configured to provide
21 power to the transmission (e.g. to allow the actuator apparatus to do work on a load).

22
23 It may be that the hydraulic transmission further comprises a variable displacement
24 hydraulic motor configured to convert pressurised hydraulic fluid received from a high
25 pressure hydraulic fluid store to rotation of a rotatable shaft, and an electrical
26 generator driven by the said rotation of the rotatable shaft to generate electricity.
27 Typically the said variable displacement hydraulic motor and electrical generator are
28 separate from (i.e. not part of) the continuous pressure control system (although of
29 course the motor may be in communication with a high pressure hydraulic fluid store
30 and/or a hydraulic fluid reservoir to which the continuous pressure control system is
31 also selectively connected).

32
33 It may be that the hydraulic transmission further comprises a prime mover (e.g.
34 electric motor or engine such as an internal combustion engine) and a variable
35 displacement hydraulic pump configured to convert rotation of a rotatable shaft of the
36 prime mover to pressurised hydraulic fluid (which is typically provided to a high
37 pressure hydraulic fluid store such as an accumulator). Typically the said prime

1 mover and variable displacement hydraulic pump are separate from (i.e. not part of)
2 the continuous pressure control system (although of course the pump may be in
3 communication with a high pressure hydraulic fluid store and/or a hydraulic fluid
4 reservoir to which the continuous pressure control system is also selectively
5 connected).

6
7 By flow rate of hydraulic fluid input to or output from one or more second chambers of
8 the said plurality of chambers, we typically mean the volume of hydraulic fluid flowing
9 into or out of the said one or more second chambers per unit time. However, it will be
10 understood that this flow rate may be delivered by (for example) controlling a volume
11 of hydraulic fluid flow per cycle of rotation of a rotatable crankshaft (e.g. a rotatable
12 crankshaft of one or more pump-motors of the continuous pressure control system,
13 the pump-motors being operable to function as a motor or a pump in different
14 operating modes).

15
16 It will be understood that the actuator apparatus typically comprises one or more
17 coupled actuators. It will also be understood that the actuator apparatus may drive or
18 be driven by a hydraulic fluid (or in some embodiments a mixture of these).

19
20 Typically the pressure or flow rate of the hydraulic fluid input to or output from the
21 second chambers is variable within a continuous range of pressures or a continuous
22 range of flow rates respectively.

23
24 Typically the continuous pressure control system is configured to continuously control
25 the pressure or flow rate of hydraulic fluid input to or output from the said second
26 chambers. It may be that the continuous pressure control system is configured to
27 continuously vary the pressure or flow rate of hydraulic fluid input to or output from
28 the second chambers such that a contribution of the second chambers to the said
29 property at least partially (typically continuously) compensates for a difference
30 between a value of the said property indicated by a or the received demand and a
31 contribution of the said first chambers to the said property.

32
33 Typically the second chambers are different from the first chambers in the sense that
34 none of the first chambers are second chambers and vice versa. It will be understood
35 that one or more first chambers may be of identical construction to one or more
36 second chambers.

37

1 Typically the controller is configured to control the discretised and continuous
2 pressure control systems to thereby regulate a property of the actuator apparatus
3 affected by the hydraulic fluid pressure in the said chambers to follow a or the
4 received demand for the said property of the actuator apparatus.

5
6 Typically the controller is configured to generate a discretised demand and a
7 continuous demand from the said received demand for the said property of the
8 actuator apparatus, the controller being configured to control the discretised pressure
9 control system to follow the discretised demand and to control the continuous
10 pressure control system to follow the continuous demand.

11
12 Typically the discretised demand is determined by offsetting (i.e. applying an offset
13 to) the received demand for the said property of the actuator apparatus (typically by a
14 constant amount). Typically the discretised demand is determined by offsetting (i.e.
15 applying an offset to) the received demand to determine an offset demand signal and
16 then quantising the offset demand signal, the quantising typically taking into account
17 the number of first chambers (and typically the direction in which they are configured
18 to act).

19
20 It may be that the hydraulic fluid transmission comprises a single second chamber. It
21 may be that the discretised demand is offset from the received demand in a direction
22 opposite from that in which the second chamber is configured to act (i.e. to contribute
23 to the output) to determine the offset demand.

24
25 Typically the discretised demand is determined by offsetting (i.e. applying an offset
26 to) the received demand by half the load range available from an individual second
27 chamber of the said second chamber(s).

28
29 By determining the discretised demand by offsetting (i.e. applying an offset to) the
30 received demand (typically by half the load range available from an individual second
31 chamber), a single second chamber acting in one direction only can be made to
32 correct steps in the contribution of the first chambers to the demand. This is because
33 the offset biases the continuous demand signal to always remain in the right sense
34 for the second chamber to follow it.

35

1 This allows the number of second chambers to be minimised (e.g. a single second
2 chamber can be provided), which thereby helps to increase the efficiency of the
3 transmission as a whole and reducing its cost.

4
5 Alternatively, it may be that a plurality of second chambers are provided, at least two
6 of the said second chambers being operable to act in opposition to each other. In this
7 case, it is not necessary to offset the discretised demand from the received demand.
8 However, as the number of second chambers is increased, the efficiency of the
9 transmission as a whole may be reduced and the cost may increase.

10
11 It may be that two or more (or three or more) said first chambers are connected in
12 parallel. It may be that two or more (or three or more) said first chambers are
13 connected in series.

14
15 It may be that two or more (or three or more) said second chambers are connected in
16 parallel. It may be that two or more (or three or more) said second chambers are
17 connected in series.

18
19 Typically the hydraulic fluid transmission comprises one or more pressure sensors
20 configured to measure the pressure in each of the first chambers, the said pressure
21 sensors being in (typically electronic) communication with the controller. Typically
22 the hydraulic fluid transmission comprises one or more pressure sensors configured
23 to measure the pressure in each of the second chambers, the said pressure sensors
24 being in (typically electronic) communication with the controller.

25
26 Typically the hydraulic fluid transmission comprises one or more pressure sensors
27 configured to measure the pressure in each of the said hydraulic fluid sources and/or
28 each of the said hydraulic fluid sinks, each of the said pressure sensors being in
29 (typically electronic) communication with the controller.

30
31 Typically the said hydraulic fluid sources or sinks comprise one or more high
32 pressure hydraulic fluid sources or sinks and one or more low pressure hydraulic fluid
33 sources or sinks, the high pressure hydraulic fluid sources or sinks having greater
34 hydraulic pressures than the low pressure hydraulic fluid sources or sinks.

35
36 Typically the controller is configured to determine the continuous demand by:
37 determining an output from the said first chambers by identifying one or more first

1 chambers selectively connected to a high pressure hydraulic fluid source or sink;
2 measuring the pressures in each of the said identified first chambers (typically by
3 receiving pressure measurement signals from pressure sensors in the said
4 chambers); and deriving the continuous demand from the determined pressures in
5 the said first chambers and the said received demand for the said property of the said
6 actuator apparatus.

7
8 Typically the continuous pressure control system is configured to take into account
9 one or more physical properties of the actuator apparatus to determine a flow rate or
10 pressure of hydraulic fluid delivered to or received from the said one or more said
11 second chambers. For example, the said physical properties may include any one or
12 more of: a speed of movement of one or more pistons of the actuator apparatus;
13 volume(s) of the said one or more second chambers; and a rate of change of volume
14 of the said one or more chambers. The continuous pressure control system is
15 typically configured to take into account one or more properties of the hydraulic
16 working fluid within the second chamber(s), such as the fluid stiffness, to determine a
17 flow rate or pressure of hydraulic fluid delivered to or received from the said one or
18 more said second chambers.

19
20 Typically the continuous pressure control system is configured to adjust a flow rate or
21 pressure of hydraulic fluid delivered to or received from the said one or more said
22 second chambers to reduce an error between a measured pressure (e.g. by one or
23 more pressure sensors configured to measure the pressure in the said second
24 chamber) in the said second chamber(s) and the continuous demand (typically taking
25 into account one or more physical properties of the actuator apparatus as discussed
26 above).

27
28 Typically the continuous pressure control system comprises one or more variable
29 displacement pump/motors operable to operate as a pump or as a motor in different
30 operating modes.

31
32 Typically the continuous pressure control system comprises a variable displacement
33 pump/motor (operable to operate as a pump or as a motor in different operating
34 modes) in a common shaft arrangement with another unit exchanging hydraulic fluid
35 with a high pressure hydraulic fluid source. Typically the said another unit comprises
36 a second variable displacement pump/motor (operable to operate as a pump or as a
37 motor in different operating modes).

1
2 Typically the variable displacement pump/motor(s) is (are) under active computer
3 programmed control (typically by the said controller).
4

5 Optionally the variable displacement pump/motor(s) is (are) a (synthetically
6 commutated) variable displacement pump/motor(s), typically each comprising at least
7 one working chamber of cyclically varying volume (typically a plurality of working
8 chambers of cyclically varying volume), a high pressure manifold, a low pressure
9 manifold and a plurality of valves which regulate the flow of fluid between the at least
10 one working chamber and the low and high pressure manifolds. Typically at least
11 one valve associated with the or each working chamber is an electronically controlled
12 valve operable in phased relationship to cycles of working chamber volume to select
13 the net volume of working fluid displaced by the respective working chamber during
14 each successive cycle of working chamber volume.
15

16 Typically the discretised pressure control system comprises a plurality of
17 electronically controllable valves which regulate the flow of fluid between each said
18 first chamber and one of the said hydraulic sinks or sources. Each said valve is
19 typically operable between an open position in which hydraulic fluid can flow between
20 a said first chamber and a hydraulic fluid source or sink and a closed position in
21 which there is no hydraulic fluid flow through the valve.
22

23 It may be that the common shaft is coupled to a prime mover or energy sink, and
24 wherein the controller is configured to control the net shaft torque to provide a
25 smoothed net power transfer between the common shaft and the prime mover or
26 energy sink.
27

28 It may be that the continuous pressure control system comprises one or more
29 variable flow control valves each of which is configured to regulate the flow of
30 hydraulic fluid between a said second chamber and a respective one of a plurality of
31 hydraulic fluid sources or sinks.
32

33 Typically the continuous pressure control system is configured to control the pressure
34 or flow rate of hydraulic fluid into or out of a said second chamber responsive to a
35 (typically externally driven) change of volume of that chamber (e.g. to directly match
36 the change in volume of that chamber).
37

1 Typically the continuous pressure control system is configured to control the flow rate
2 or pressure of hydraulic fluid into or out of a said second chamber to cause a desired
3 change in pressure taking into account the volume of the said second chamber.

4 Typically the continuous pressure control system is configured to control the flow rate
5 or pressure of hydraulic fluid into or out of a said second chamber to cause a desired
6 change in pressure taking into account an effective bulk modulus of the fluid volume.

7
8 Typically the controller is configured to synchronise a change in pressure of a said
9 second chamber with a corresponding change in pressure in one or more said first
10 chambers.

11
12 Typically the fluid driving surfaces of the said chambers are coupled to each other
13 such that the forces exerted by hydraulic fluid on the respective fluid driving surfaces
14 of the chambers are additive or subtractive.

15
16 Typically the discretised pressure control system comprises one or more inlet valves
17 configured to selectively connect a said first chamber to a hydraulic fluid source,
18 and/or one or more outlet valves configured to selectively connect a or the said first
19 chamber to a hydraulic fluid sink.

20
21 It may be that the one or more inlet valves comprise a pair of inlet valves connected
22 in parallel with each other.

23
24 It may be that the one or more outlet valves comprise a pair of outlet valves
25 connected in parallel with each other.

26
27 Typically a first one of the valves in each pair permits a greater hydraulic fluid flow
28 rate therethrough than a second one of the valves in that pair.

29
30 Typically the second one of the said valves transitions from a closed state in which no
31 hydraulic fluid flows therethrough to an open state in which hydraulic fluid flows
32 therethrough (typically at a maximum hydraulic fluid flow rate for the valve) more
33 quickly than the first one of the said valves.

34
35 Typically the controller is configured to open the said second one of the valves before
36 opening the said first one of the valves in order to change the pressure in one or
37 more first chambers.

1
2 The second one of the said pair of valves controls the timing and transition profile of
3 the transition of the first chamber between connection to a hydraulic fluid source and
4 a hydraulic fluid sink or vice versa. That is, the second one of the said pair of valves
5 controls the timing and transition profile of the transition of the first chamber between
6 relatively low and high pressure states. Thus, the timing and profile of the said
7 transition can be controlled (e.g. slowed) to permit a pressure change in the said
8 second chamber(s) to synchronise with a pressure change in the first chambers.

9
10 The invention extends in a second aspect to apparatus for extracting electrical power
11 from a renewable energy source comprising a hydraulic fluid transmission according
12 to the first aspect of the invention. In this case, the renewable energy source is
13 typically configured to drive the actuator apparatus. Typically the said property of the
14 actuator apparatus is a property of the actuator apparatus which affects the
15 resistance of the actuator apparatus to movement caused by the renewable energy
16 source. For example, the said property may be an actuator pressure.

17
18 The invention also extends in a third aspect to apparatus for extracting electrical
19 power from water waves comprising a hydraulic fluid transmission according to the
20 first aspect of the invention.

21
22 Typically the apparatus comprises one or more pairs of buoyant body members
23 connected to each other by a coupling which permits relative rotation between the
24 body members. The apparatus typically comprises a power extraction system
25 comprising the said actuator apparatus, the said actuator apparatus having one or
26 more coupled actuators each having a first end coupled to a first body member of the
27 said pair of body members and a second end coupled a second body member of the
28 said pair of body members such that the actuator(s) resists and extracts power from
29 the relative rotation between the body members.

30
31 The invention extends in a fourth aspect to a heave compensation winch comprising
32 a hydraulic fluid transmission according to the first aspect of the invention.

33
34 In this case, the hydraulic fluid transmission is typically configured to convert stored
35 hydraulic pressure and/or shaft power from a prime mover (such as an electric motor)
36 into a varying load on a cable which varies the speed of the winding in and out of the

1 cable. The said property of the actuator apparatus in this case is typically the load on
2 the said cable or the speed of the winding in and out of the cable.

3
4 A fifth aspect of the invention provides a method of transmitting hydraulic fluid power,
5 the method comprising: providing an actuator apparatus having a plurality of
6 chambers, each of the chambers having a respective fluid driving surface configured
7 to drive or be driven by hydraulic fluid therein; selectively connecting said one or
8 more first chambers to one of a plurality of hydraulic fluid sources or sinks, at least
9 two of the said plurality of hydraulic fluid sources or sinks having different fluid
10 pressures; and controlling the pressure or flow rate of hydraulic fluid input to or output
11 from said one or more second chambers of the said plurality of chambers by varying
12 the pressure or flow rate of hydraulic fluid flowing into or out of the said second
13 chambers within a range of pressures or flow rates respectively to thereby regulate a
14 property of the actuator apparatus affected by the hydraulic fluid pressure in the said
15 chambers.

16
17 The method may further comprise controlling the pressure or flow rate of hydraulic
18 fluid input to or output from said one or more second chambers by varying the
19 pressure or flow rate of hydraulic fluid flowing into or out of the said second chambers
20 within a range of pressures or flow rates respectively such that a contribution of the
21 second chambers (and typically thus the continuous pressure control system) to the
22 said property of the actuator apparatus at least partially (typically continuously)
23 compensates for a difference between a contribution of the said first chambers to the
24 said property of the actuator apparatus and a value of the said property of the
25 actuator apparatus indicated by a received demand.

26
27 The preferred and optional features discussed above in respect of any aspect of the
28 invention are preferred and optional features of each aspect of the invention.

29 30 Description of the Drawings

31
32 An example embodiment of the present invention will now be illustrated with
33 reference to the following Figures in which:

34
35 Figure 1 is a schematic hydraulic circuit diagram of a hydraulic transmission which
36 uses continuous pressure control enabled by a pair of variable displacement pump-
37 motors on a common shaft;

Figure 2 is a schematic hydraulic circuit diagram of a hydraulic transmission which uses quantised pressure control enabled by two pairs of electronically controlled valves configured to selectively connect each of a pair of actuator chambers to a high pressure energy store and a low pressure reservoir;

Figure 3 is a schematic hydraulic circuit diagram of a hybrid hydraulic transmission having a pair of actuators each with four hydraulic fluid chambers acting about a pivot to allow torque and power to be transmitted about it, the pressure of all four chambers of one actuator and the pressure of three of the other being controlled by a discretised pressure control system and the pressure within the remaining actuator of the said other actuator being controlled by a continuous pressure control system;

Figure 4 is a block diagram of an algorithm for calculating a demand to be met by the continuous pressure control system of Figure 3;

Figure 5 is a block diagram of an algorithm implemented by the continuous pressure control system to meet the demand calculated by the algorithm of Figure 4;

Figure 6 is a graph showing the various demand and output signals from example quantised and continuous pressure control systems; and

Figure 7 is a block diagram illustrating the hydraulic transmission of Figure 3 used in a heave compensation winch.

Detailed Description of an Example Embodiment

Figure 3 is a schematic hydraulic circuit diagram of a hybrid hydraulic fluid transmission comprising first and second linear hydraulic actuators 100, 102 acting about a pivot to allow torque and power to be transmitted about it, the first and second actuators 100, 102 each having four actuator chambers 104, 106, 108, 110 and 141, 142, 143, 144 respectively. The hydraulic actuators 100, 102 are hydraulic rams of apparatus for extracting energy from water waves, the actuators 100, 102 being driven by the relative rotation of buoyant body members (not shown) provided in a body of water (e.g. an ocean), the relative rotation being caused by waves in the body of water, the buoyant body members being coupled to each other by a coupling which permits relative rotation between the body members and which causes

1 movement within each of the actuators 100, 102 of a shaft 112 into and out of a
2 cylinder 114, thereby causing first and second pistons 116, 118 provided on the shaft
3 112 to reciprocate in the cylinder 114 to perform work. The pistons 116, 118 of the
4 actuators 100, 102 each have opposing respective fluid driving surfaces configured to
5 drive or be driven by hydraulic fluid in the respective chambers 104-110 (in respect of
6 the first actuator 100), 141-144 (in respect of the second actuator 102). A divider 120
7 is provided between the pistons 116, 118 of each actuator 100, 102 so that the
8 pistons 116, 118 of each actuator 100, 102 operate in series with each other.
9 Chambers 104, 106 are provided on either side of the first piston 116 of the first
10 actuator 100, while chambers 108, 110 are provided on either side of the second
11 piston 118 of the first actuator 100. Chambers 141, 142 are provided on either side
12 of the first piston 116 of the second actuator 102, while chambers 143, 144 are
13 provided on either side of the second piston 118 of the second actuator 102. The
14 pressures in the chambers 104-110, 141-144 are controlled in order to control the
15 resistance of the movement of the pistons 116, 118 within the cylinder 114 of each
16 actuator 100, 102, in order to maximise the energy extracted by the hydraulic rams.

17
18 Three chambers 104, 106 and 108 of the first actuator 100 and all four chambers
19 141-144 of the second actuator 102 (hereinafter referred to as first chambers) are
20 selectively connected to a high pressure hydraulic fluid accumulator 120 and a low
21 pressure hydraulic fluid reservoir 122 by way of respective hydraulic fluid lines and
22 respective banks 124 of electronically controlled valves coupled between the
23 respective chambers 104-108, 141-144 and the accumulator 120 and the reservoir
24 122. The banks 124 thus operate as a discretised pressure control system. Each of
25 the banks 124 of electronically controlled valves are identical, so only the bank 124
26 coupled between the chamber 104 of the first actuator 100 and the accumulator 120
27 and reservoir 122 is described below for brevity.

28
29 The bank 124 comprises first and second electronically controlled high pressure
30 valves 126, 128 arranged in parallel and configured to selectively connect the
31 chamber 104 to the accumulator 120. Each of the high pressure valves 126, 128
32 have open states in which hydraulic fluid can flow through them between the
33 chamber 104 and the accumulator 120 and closed states in which hydraulic fluid
34 cannot flow through them between the chamber 104 and the accumulator 120. The
35 first high pressure valve 126 has a greater fluid flow area therethrough when it is in its
36 open state than the second high pressure valve 128. This is illustrated in Figure 3 by
37 way of a hydraulic impedance in series with the second high pressure valve 128. The

1 second high pressure valve 128 (e.g. by virtue of having a valve member with a
2 smaller cross sectional area than that of the first high pressure valve 126) is capable
3 of transitioning between its open and closed states more quickly than the first high
4 pressure valve 126, but the maximum fluid flow rate through the second high
5 pressure valve 128 (when open) is less than the maximum fluid flow rate through the
6 first high pressure valve 126 (when open).

7
8 The bank 124 further comprises first and second electronically controlled low
9 pressure valves 130, 132 arranged in parallel and configured to selectively connect
10 the chamber 104 to the reservoir 122. The first low pressure valve 130 has a greater
11 fluid flow area therethrough when it is in its open state than the second low pressure
12 valve 132. This is illustrated in Figure 3 by way of a hydraulic impedance in series
13 with the second high pressure valve 132. The second low pressure valve 132 (e.g.
14 by virtue of having a valve member with a smaller cross sectional area than that of
15 the first low pressure valve 130) is capable of transitioning between its open and
16 closed states more quickly than the first low pressure valve 130, but the maximum
17 fluid flow rate through the second low pressure valve 132 (when open) is less than
18 the maximum fluid flow rate through the first low pressure valve 130 (when open).

19
20 The bank 124 further comprises a large cross sectional flow area low pressure check
21 valve 134 operable to selectively connect the chamber 104 to the low pressure
22 reservoir 122 in order to maintain the pressure in the chamber 104 at least at the
23 pressure in the low pressure reservoir (which may be charged above atmospheric
24 pressure), for example when the chamber 104 is increasing in volume (by movement
25 of the shaft 112 and the piston 116), to prevent cavitation. The check valve 134 is
26 configured to open when the pressure in the chamber 104 falls below the pressure of
27 the reservoir 122.

28
29 The valves of the bank 124 control the timing and the transition profiles (i.e. the way
30 in which the pressure changes over time) of the chamber 104 between a high
31 pressure state, when it is connected to the accumulator 120, and a low pressure state
32 when it is connected to the reservoir 122. More specifically, the second low or high
33 pressure valves 128, 132 open first, to begin the pressure transition gradually. When
34 the pressure transition is underway, the first low or high pressure valves 128, 132 are
35 opened to allow the full rate of hydraulic fluid flow through the combination of first and
36 second low or high pressure valves, to thereby complete the pressure transition. This
37 prevents extremely quick large pressure transitions from occurring, which could

1 cause a shock to the system. In addition, the timing profiles of the transitions are
2 typically controlled to permit a pressure change in chamber 110 to synchronise with a
3 pressure change in one or more of the first chambers 104-108, 141-144 and to
4 optimise the performance and accuracy of the overall system. Typically in practice
5 the timing profiles of the transitions of the chamber 104 between the low and high
6 pressure states ensures that the control latency of the chamber 104 is greater than
7 the control latency of the chamber 110.

8
9 The valves of the banks 124 are typically electronically controlled solenoid valves
10 which are controlled by a controller 140 provided in electronic communication with the
11 valves. The controller 140 acts to selectively open and close the valves by turning a
12 solenoid on and off, the valves being biased to the open or closed positions by a
13 (typically passive) biasing mechanism such as a spring. The banks 124 of low and
14 high pressure valves are thus part of a discretised control system under the control of
15 the controller 140 which is configured to selectively connect the first chambers 104-
16 108, 141-144 to the accumulator 120 and the reservoir 122. The controller typically
17 comprises one or more computer processors configured to run computer program
18 code. It may be that the controller 140 is distributed across a plurality of processors.

19
20 As explained in the Background of invention, use of this discretised control method
21 alone leads to an undesirable stepped output.

22
23 Chamber 110 (which will be referred to as a second chamber hereinafter) is
24 connected to a continuous pressure control system comprising a first variable
25 displacement hydraulic pump-motor 150 coupled to a second variable displacement
26 hydraulic pump-motor 152 by a common shaft 151, the variable displacement
27 hydraulic pump-motors 150, 152 each being operable to function as a pump or a
28 motor in different operating modes. The variable displacement hydraulic pump-
29 motors 150, 152 are also coupled to an electricity generator 154 by way of the
30 common shaft 151 (which is operable to drive the generator 154). Connection
31 between the chamber 110 and the continuous pressure control system is by way of a
32 hydraulic line 160, and the chamber 110 is also selectively connected to the low
33 pressure reservoir 122 by an electronically controlled valve 162 which is controlled
34 by, and in electronic communication with, the controller 140. Valve 162 will either
35 open passively in the event that the pressure in the chamber 110 falls below the
36 pressure in the reservoir 122 (to prevent cavitation) or actively on the command of the

1 controller 140 to disable flow from the pump-motors 150,152 into or out of the
2 chamber 110.

3
4 Depending on the pressure requirements in the second chamber 110, one of the first
5 and second variable displacement hydraulic pump-motors 150, 152 operates in a
6 motoring mode to convert high pressure fluid (received from the chamber 110 or from
7 the accumulator 120) to torque on the shaft 151 and output low pressure fluid to the
8 reservoir 122. The other of the first and second variable displacement hydraulic
9 pump-motors 150, 152 operates in pumping mode to convert torque of the shaft to
10 pressurised fluid which is either provided to the accumulator 120 or to the chamber
11 110. The displacements of the hydraulic pump-motors 150, 152 can be varied in
12 each case to consume or provide hydraulic fluid from or to the second chamber 110
13 having a pressure which is variable across a (typically continuous) range.

14
15 It will be understood that if the continuous pressure control system is required to
16 consume high pressure fluid from the chamber 110, the first variable displacement
17 hydraulic pump-motor 150 functions in the motoring mode, converting high pressure
18 fluid from the chamber 110 to a torque on the shaft 151. The second variable
19 displacement hydraulic pump-motor operates in pumping mode, converting torque
20 from the shaft 151 to high pressure fluid provided to the accumulator. In addition,
21 some of the torque on the shaft 151 is converted to electricity by the generator 154.
22 This helps to reduce the number of rotating machines required to generate power,
23 thereby improving efficiency of power generation. The pump-motors 150, 152 also
24 provide an efficient means for starting and speed control of the shaft. The quantity of
25 high pressure fluid consumed can be controlled by adjusting the displacements of the
26 hydraulic pump-motors 150, 152.

27
28 If the continuous pressure control system is required to provide high pressure fluid to
29 the chamber 110, the second variable displacement hydraulic pump-motor 152
30 functions in motoring mode, converting high pressure hydraulic fluid from the
31 accumulator 120 to torque on the shaft 151. In this case, the first variable
32 displacement hydraulic pump-motor 150 functions in pumping mode, converting
33 torque on the shaft 151 to pressurised hydraulic fluid provided to the chamber 110.
34 The quantity of high pressure fluid provided can be controlled by adjusting the
35 displacements of the hydraulic pump-motors 150, 152.

36

1 In practice, the pressure in the second chamber 110 is controlled by the pump-motors
2 150, 152 demanding a little less or more than the flow required to match the
3 'geometric flow' that the chamber 110 is providing or absorbing as a result of the
4 motion of the actuator shaft 112. The 'geometric flow' is the flow required to maintain
5 a constant pressure in the chamber 110, taking into account increases or decreases
6 in the volume of the chamber 110 as a result of motion of the actuator shaft 112. If
7 the pump-motors 150, 152 are absorbing flow from the chamber 110, and if they
8 absorb a little less flow than the 'geometric flow' being generated, the pressure in the
9 chamber 110 will rise. Conversely if the pump-motors 150, 152 absorb a little more
10 than the geometric flow, the pressure in the second chamber 110 will fall. In the
11 inverse case where the actuator 100 is absorbing flow from the pump-motors 150,
12 152, if the pump-motors 150, 152 provide greater flow to the chamber 110 than the
13 'geometric flow' required, the pressure in the chamber 110 will rise. If the pump-
14 motors 150, 152 provide less than the geometric flow, the pressure in the chamber
15 110 will fall. If the actuator is stationary, there is no 'geometric flow' of course and the
16 pump-motors 150, 152 only need to deliver a small flow to pressurise the chamber
17 110 accordingly. It is through this mechanism that the pump-motors 150, 152 can
18 deliver continuous pressure control over the chamber 110, and thereby continuous
19 load control.

20
21 In one example, the variable displacement pump-motors 150, 152 comprise
22 (synthetically commutated) variable displacement pump/motor(s), each comprising at
23 least one working chamber of cyclically varying volume (typically a plurality of
24 working chambers of cyclically varying volume), a high pressure manifold, a low
25 pressure manifold and a plurality of valves which regulate the flow of fluid between
26 the at least one working chamber and the low and high pressure manifolds, at least
27 one valve associated with the or each working chamber being an electronically
28 controlled valve operable in phased relationship to cycles of working chamber volume
29 to select the net volume of working fluid displaced by the respective working chamber
30 during each successive cycle of working chamber volume. The quantity of
31 pressurised fluid consumed or provided by the continuous pressure control system
32 can be varied by the controller 140 by activating and deactivating working chambers
33 of the pump-motors 150, 152 on each cycle of working chamber volume (by way of
34 controlling opening and closing of respective valves of the pump-motors).

35
36 The desired resistance of the actuator 100 to movement of the shaft 112 caused by
37 relative rotation of the buoyant body members is provided to the controller 140 in the

1 form of a computer derived load demand signal (functioning as the received
2 demand). The first chambers 104-108, 141-144 and the second chamber 110 work
3 to meet the load demand. As will be explained below, the contribution of the second
4 chamber 110 (and thus the continuous pressure control system) to meet the load
5 demand can be selected to at least partially (preferably fully) compensate for a
6 difference between the received demand and a contribution of the first chambers
7 104-108, 141-144 (and thus the discretised pressure control system) to meet the load
8 demand, thereby significantly reducing (preferably removing) the steps in the
9 resistance profile caused by the contribution of the first chambers 104-108, 141-144
10 (and thus the discretised pressure control system).

11
12 As shown in Figure 6, the demand 180 received by the controller is a smooth curve.
13 The controller 140 is configured to generate from the demand signal 180 a
14 discretised load demand 220 (also shown in Figure 6), the discretised demand 220
15 being the portion of the demand to which the first chambers 104-108, 141-144 are
16 required to meet. The controller 140 is configured to cause the first chambers 104,
17 141-144 to meet the discretised demand 220 as best they can such that the first
18 chambers 104-108, 141-144 provide the main load and power flow. The difference or
19 error between the load delivered by the first chambers 104-108, 141-144 and the
20 received demand 180 forms a continuous demand 222 for the second chamber 110
21 to deliver.

22
23 In this example the (single) second chamber 110 is operable to provide a force in only
24 one direction (to the left in the view of Figure 3). A single second chamber 110 acting
25 in one direction only can be made to correct the steps in the discrete system by
26 generating the discretised demand 220 by: offsetting (i.e. applying an offset to) the
27 received demand 180 by (typically) half of the load range of the second chamber 110
28 (the load range of the second chamber being the range from the minimum
29 contribution of the second chamber 110 to meet the demand to the maximum
30 contribution of the second chamber 110 to meet the demand) in a direction opposite
31 to that in which the second chamber 110 is configured to provide an active force to
32 generate an offset demand 181; and then quantising the offset demand 181 to
33 generate the discretised demand 220. This biases the correction signal to always
34 remain in the right sense for the second chamber 110 to correct it. To further clarify,
35 this approach and effect is illustrated clearly in Figure 6. The received demand signal
36 180 is offset down by half of the range of the second chamber which can be seen
37 from the maximum and minimum range of the continuous demand 222. This means

1 that (in this example) the continuous demand 222 inherently remains positive and
2 within the load range that can be applied by the second chamber 110.

3
4 For alternative embodiments in which the actuator 100 is a rotary actuator, the
5 direction of fluid flow into and out of the actuator chambers could be reversed as
6 required as an alternative to the above approach.

7
8 It will be understood that if at least one second chamber 110 (i.e. at least one
9 chamber in respect of which hydraulic fluid provided thereto or consumed therefrom
10 having a continuously controlled pressure) is provided which is operable to provide a
11 force in one direction and at least one second chamber 110 is provided which is
12 operable to provide a force in the opposite direction, it would not be necessary to
13 offset the discretised demand from the received demand as the continuous pressure
14 controlled chambers could act to compensate for errors between the discrete
15 pressure controlled chambers 104-108, 141-144 in either direction.

16
17 If there is relative clockwise motion of the buoyant body members around the pivot
18 axis shown in Figure 3 (between the ends of the double headed arrow between
19 actuators 100, 102), the first actuator 100 will retract (the shaft 112 of the first
20 actuator 100 moves to the right in the view of Figure 3) and the second actuator will
21 extend (the shaft 112 of the second actuator 102 moves to the left in the view of
22 Figure 3). In this case, the volumes of the first chambers 104, 108 of the first actuator
23 100 will increase and the volumes of the first chamber 106 and the second chamber
24 110 of the first actuator 100 will decrease. The volumes of the first chambers 141,
25 143 of the second actuator 102 will increase and the volumes of the first chambers
26 142, 144 of the second actuator 102 will decrease. The opposite is of course true if
27 there is relative anticlockwise motion of the buoyant body members. In either case,
28 as the shafts 112 of the first and second actuators 100, 102 move, the controller 140
29 controls the states of the high and low pressure valves in the valve blocks 124
30 associated with each first chamber 104-108, 141-144 to select the pressure of each
31 first chamber 104-108, 141-144 as low or high (by selectively connecting them to the
32 reservoir 122 or accumulator 120 respectively) to meet the discretised demand 220.
33 Pressure within each of the first chambers 104-108, 141-144 can be controlled to
34 either resist or assist movement of the shafts 112 of the actuators 100, 102 caused
35 by the relative rotation of the buoyant body members, although it will be understood
36 that the overall aim is that the relative rotation causes a net flow to the accumulator
37 120 (and to enable power to be extracted from the system).

1
2 The pressure within the second chamber 110 is also controlled by controlling the
3 pressure and/or volume of working fluid flowing into the chamber 110 from the pump-
4 motors 150, 152. However, as explained above, while the pressure within the first
5 chambers 104-108, 141-144 is restricted to either the pressure of the accumulator
6 120 or the pressure of the reservoir 122, the pressure within the second chamber 110
7 is controllable within a continuous range of values by the pump-motors 150, 152.
8 This property can be used to smooth the steps in the output provided by the first
9 chambers 104-108, 141-144, to thereby improve the quality of the output as a whole,
10 while retaining much of the benefits (e.g. improved efficiency) of the discrete pressure
11 control system of the first chambers 104-108, 141-144.

12
13 In order to calculate a continuous demand 222 to smooth the steps in the output
14 provided by the first chambers 104-108, 141-144, the controller 140 is configured to
15 receive pressure measurement signals from pressure sensors within the first
16 chambers 104-108, 141-144, identify which of the chambers 104-108, 141-144
17 contributes to the demand and derive the continuous demand 222 from the received
18 pressure measurements and the said identification. As illustrated in Figure 4, the
19 continuous demand 222 is typically derived by subtracting the measured (stepped)
20 contributions of the chambers 104-108, 141-144 from the received load demand,
21 taking into account the area of the relevant fluid driving surface of piston 118 of the
22 first actuator 100.

23
24 Figure 5 is a block diagram of an example algorithm implemented by the controller
25 140 to control the continuous pressure control system (and thus the pressure in the
26 second chamber 110) to meet the continuous demand 222. The algorithm receives
27 the continuous pressure demand 222 as an input to a difference operator 192 which
28 also receives as another input the measured pressure 194 in the second chamber
29 110 (typically measured by a pressure sensor configured to measure the pressure in
30 the second chamber 110, the pressure sensor 110 being in electronic communication
31 with the controller 140), the difference operator 192 being configured to subtract the
32 measured pressure 194 from the continuous pressure demand 222 to provide a
33 difference signal 196. This feedback loop provides fine control to correct minor errors
34 in the pressure of the second chamber 110 to meet the continuous pressure demand
35 222.

36

1 The difference signal 196 is input to (in this example) a proportional integral (PI)
2 control block 198 which in turn provides a control signal 199 to a summing operator
3 200. Also provided to the summing operator 200 is an input 202 which enables the
4 algorithm to take into account measurements (e.g. measurements of speed,
5 acceleration) of the motion of the actuator shaft 112 (e.g. rate of change of volume
6 204 of the second chamber 110) and knowledge of the compliance of the volume of
7 the second chamber 110 (i.e. volume 206 of chamber 110 and the stiffness 208 of the
8 hydraulic working fluid) to determine the required flow to or from the second chamber
9 110. For example, measurements of the motion of the actuator shaft 112 can be
10 used by the algorithm to determine the flow rate required to maintain constant
11 pressure in the chamber 110 taking into account motion of the actuator shaft 112.
12 Knowledge of the compliance of the volume of the second chamber 110 can be used
13 by the algorithm to determine change in fluid volume at the working pressure required
14 to achieve a given pressure change. As shown in Figure 5, knowledge of the
15 compliance of the volume of the second chamber 110 can be taken into account by
16 differentiating the continuous pressure demand with respect to time and multiplying
17 the differentiated continuous pressure demand by the volume of chamber 110 divided
18 by the fluid stiffness. This is then taken into account, together with the rate of change
19 of actuator volume, to provide the input 202 to the summing operator 200.

20
21 The output of the summing operator 200 is fed as a flow demand signal to the pump-
22 motors 150, 152 which delivers a flow rate of hydraulic fluid to or from the second
23 chamber 110 as close as possible to the flow demand. It will be understood that the
24 flow rate of hydraulic fluid flowing into or out of the second chamber 110 typically
25 refers to the volume of hydraulic fluid flowing into or out of the second chamber per
26 unit time, but that this may be delivered by controlling a volume of hydraulic fluid flow
27 to or from the pump-motors 150, 152 per cycle of rotation of the rotatable shaft 151.
28 The delivered flow rate, together with the volume 230 of the chamber 110 (which is
29 determined by the externally induced motion 232 of the actuator shaft 112 of the first
30 actuator 100), determine the pressure in the second chamber 110.

31
32 It will be understood that, in an alternative embodiment, the pressure within the
33 chamber 110 can be controlled by providing the pump-motors 150, 152 with a
34 continuous pressure demand signal, in which case the pump-motors 150, 152 are
35 operable to determine the required flow rate of hydraulic fluid into or out of the
36 chamber 110 to meet with pressure demand signal and to control the displacement
37 (e.g. by activating and deactivating working chambers) accordingly.

1
2 This combination of open and closed loop control enables the second chamber 110
3 to accurately correct for the difference between the contribution to the overall demand
4 by the first chambers 104-108, 141-144 and the overall demand itself, reducing
5 (preferably removing) the effect of the steps in the contribution from the first
6 chambers 104-108, 141-144.

7
8 As explained above, Figure 6 shows the discretised load demand 220 and the
9 continuous demand 222 (being the error between the load delivered by the first
10 chambers 104-108, 141-144 and the overall demand 180). Also shown in figure 6 is
11 the overall output 224 provided by the chambers 104-110, 141-144 which faithfully
12 follows (and is therefore overlaid on) the overall demand 180, other than for a portion
13 224a where the demand cannot be met due to saturation (i.e. the demand exceeds
14 the capability of the actuator 100).

15
16 Referring back to Figure 3, a system pressure relief valve 169 is connected between
17 the pump-motors 150, 152. A further system pressure relief valve 171 is provided
18 between the reservoir 122 and the accumulator 120. A further electrical generator
19 172 is connected to the accumulator 120 by way of a variable displacement motor
20 173 configured to receive pressurised fluid from the accumulator 120 and to convert
21 received pressurised fluid from the accumulator 120 into torque on a shaft 174 which
22 drives the generator 172 to generate electrical energy.

23
24 Typically the valves in the banks 124 are selected and tuned to provide the best
25 overall response in combination with the second chamber 110. For example, the
26 valves in the banks 124 may be selected such that the peak rate of change of the
27 pressure acting in the first chambers 104-108, 141-144 can be matched by the
28 second chamber 110. This allows the contribution of the second chamber 110 to
29 compensate for a step change contributed by a first chamber 104-108, 141-144 to the
30 load.

31
32 The controller 140 may of course include constraints to ensure that the overall output
33 remains smooth and controlled, and to prevent overload or reaching endstops (for
34 example). In some applications, constraints may be applied by the controller 140 to
35 avoid the overall demand signal 180 from exceeding the capabilities of the hydraulic
36 transmission. The controller 140 may restrict the rate of change of the total load to lie
37 within the load range of the second chamber 110, which is in turn a function of its flow

1 range limit and the compliance of its volume. The controller 140 may similarly be
2 configured to constrain the peak rate of change of pressure acting in the first
3 chambers 104-108, 141-144 to lie within the load range of the second chamber 110.

4
5 The fact that most of the flow and power is handled by the first chambers 104-108,
6 141-144 (and thus the discrete pressure control system), and the fact that the high
7 instantaneous power through the second chamber 110 is exchanged directly between
8 the two pump-motors 150, 152 (rather than through a prime mover or generator)
9 makes the hydraulic transmission disclosed herein particularly suitable for handling
10 large instantaneous power flows while maintaining a very high system efficiency.

11
12 Further variations and modifications may be made within the scope of the invention
13 herein described.

14
15 For example, although the apparatus for extracting energy from water waves aims to
16 provide a net flow to the accumulator 120 (and to extract energy from the system by
17 way of generators 154, 172), the hydraulic transmission described herein is equally
18 applicable to an actuator system which causes a net flow from the accumulator 120
19 (that is, an actuator system which is driven by hydraulic fluid from the hydraulic
20 transmission to perform work on a load). In this case, the electricity generator 154 is
21 typically replaced by a prime mover, such as a diesel engine or electric motor, so as
22 to add power to the system. Additionally or alternatively, the electrical generator 172
23 and variable displacement hydraulic motor 173 may be replaced by a prime mover
24 (such as a diesel engine or electric motor) and a variable displacement hydraulic
25 pump, again so as to add power to the system. Indeed, the hydraulic transmission
26 described herein is generally applicable to any system where high power flows
27 across all four quadrants are required ('four quadrant' means a system that can apply
28 a load in either direction during motion in either direction).

29
30 Figure 7 is a block diagram of a heave compensation winch comprising an actuator
31 300 comprising a plurality of chambers (not shown) and a hydraulic transmission 302
32 configured to selectively control the connection of each of a plurality of the said
33 plurality of first chambers of the actuator to a high pressure fluid store 304 and a low
34 pressure hydraulic fluid reservoir 306 in accordance with a load demand. The
35 hydraulic transmission 302 is also configured to control the pressure in a second
36 chamber of the said plurality of chambers within a continuous range of pressures. In
37 accordance with the principles discussed above, the hydraulic transmission is

1 configured to control the pressure in the second chamber to at least partially
2 compensate for a difference between the contribution of the first chambers to a
3 received demand. In this case, the demand is typically a varying load on a cable
4 which varies the speed of the winding in and out of the cable by the winch. It will be
5 understood that the winch is an example of an application where there is a net flow of
6 power to the actuator. Accordingly, the electricity generator 154 is typically replaced
7 by a prime mover, such as a diesel engine or electric motor, so as to add power to the
8 system and/or the electrical generator 172 and variable displacement hydraulic motor
9 173 are replaced by a prime mover (such as a diesel engine or electric motor) and a
10 variable displacement hydraulic pump, again so as to add power to the system.

11
12 Although seven first chambers 104-108, 141-144 and a single second chamber 110
13 are illustrated in Figure 3, it will be understood that any number of first chambers and
14 second chambers could be provided. The best choice of design will depend on the
15 specific application requirements. Typically a minimum of two first chambers and one
16 second chamber will be provided.

17
18 Although the chambers 141-144 are illustrated in Figure 3 as being in parallel with the
19 chambers 104-108, it will be understood that they could alternatively be connected in
20 series.

21
22 In some applications, it is possible to use a single pump-motor 150 in place of the pair
23 of pump-motors 150, 152 of the continuous pressure control system of Figure 3.

24
25 In some embodiments, the variable displacement pump-motors 150, 152 can be
26 replaced by variable flow control valves, each of which is configured to regulate the
27 flow of hydraulic fluid between the second chamber 110 and the accumulator 120 or
28 the reservoir 122.

29
30 Although a linear actuator 100 is described above, the hydraulic transmission may
31 additionally or alternatively comprise one or a number of rotary actuators comprising
32 one or more chambers whose pressures are controlled discretely and one or more
33 chambers whose pressures are controlled over a continuous range of pressures in a
34 similar way to the linear actuator 100 to create a rotary output directly or through a
35 gear system, or indeed a linear output using, for example, a rack and pinion system.

36

Claims

1. A hydraulic fluid power transmission comprising: actuator apparatus having a plurality of chambers, each of the chambers having a respective fluid driving surface configured to drive or be driven by hydraulic fluid therein; a discretised pressure control system configured to selectively connect one or more first chambers of the said plurality of chambers to one of a plurality of hydraulic fluid sources or sinks, at least two of the plurality of hydraulic fluid sources or sinks having different fluid pressures; a continuous pressure control system configured to control the pressure of hydraulic fluid, or a flow rate of hydraulic fluid, input to or output from one or more second chambers of the said plurality of chambers, the pressure or flow rate of the hydraulic fluid input to or output from the second chambers being thereby variable within a range of pressures or flow rates respectively; and a controller configured to control the discretised and continuous pressure control systems to thereby regulate the hydraulic fluid pressure in the said chambers.
2. The hydraulic transmission according to claim 1, wherein the controller is configured to receive a demand indicative of a demanded value of a parameter of the actuator apparatus affected by the hydraulic fluid pressure in the said chambers, and wherein the controller is configured to control the discretised and continuous pressure control systems to follow the received demand.
3. The hydraulic transmission according to claim 1 or claim 2, wherein the controller is configured to control the discretised and continuous pressure control systems to thereby regulate a parameter of the actuator apparatus affected by the hydraulic fluid pressure in the said chambers.
4. The hydraulic transmission according to any one of claims 1 to 3, wherein the controller is configured to control the discretised and continuous pressure control systems such that the hydraulic fluid pressure in the second chambers at least partially compensates for a difference between the hydraulic fluid pressure in the said first chambers and the hydraulic fluid pressure indicated by a or the received demand.

- 1 5. The hydraulic fluid power transmission according to any one of claims 1 to 4,
2 wherein the controller is configured to control the discretised and continuous
3 pressure control systems to thereby regulate the hydraulic fluid pressure in
4 the said chambers to follow a or the received demand.
5
- 6 6. The hydraulic fluid transmission according to claim 4 or claim 5 wherein the
7 controller is configured to generate a discretised demand and a continuous
8 demand from the said received demand, the controller being configured to
9 control the discretised pressure control system to follow the discretised
10 demand and to control the continuous pressure control system to follow the
11 continuous demand.
12
- 13 7. The hydraulic fluid transmission according to claim 6 wherein the discretised
14 demand is determined by offsetting the received demand.
15
- 16 8. The hydraulic fluid transmission according to claim 7 wherein the discretised
17 demand is determined by offsetting the received demand by half the load
18 range available from an individual second chamber of the said second
19 chamber(s), the load range being the range from a minimum contribution of
20 the individual second chamber to meet the received demand to a maximum
21 contribution of the individual second chamber to meet the received demand.
22
- 23 9. The hydraulic fluid transmission according to any one preceding claim
24 wherein the continuous pressure control system comprises one or more
25 variable displacement pump/motors operable to operate as a pump or as a
26 motor in different operating modes.
27
- 28 10. The hydraulic fluid transmission according to claim 9 wherein the continuous
29 pressure control system comprises a variable displacement pump/motor in a
30 common shaft arrangement with another unit exchanging hydraulic fluid with a
31 high pressure hydraulic fluid source.
32
- 33 11. The hydraulic fluid transmission according to claim 10 wherein the common
34 shaft is coupled to a prime mover or energy sink, and wherein the controller is
35 configured to control the net shaft torque to provide a smoothed net power
36 transfer between the common shaft and the prime mover or energy sink.
37

- 1 12. The hydraulic fluid transmission according to any one of claims 1 to 8 wherein
2 the continuous pressure control system comprises one or more variable flow
3 control valves each of which is configured to regulate the flow of hydraulic
4 fluid between a said second chamber and a respective one of a plurality of
5 hydraulic fluid sources or sinks.
6
- 7 13. The hydraulic fluid transmission according to any one preceding claim
8 wherein the continuous pressure control system is configured to control the
9 pressure or flow rate of hydraulic fluid into or out of a said second chamber
10 responsive to a change of volume of that chamber.
11
- 12 14. The hydraulic fluid transmission according to any one preceding claim
13 wherein the continuous pressure control system is configured to control the
14 flow rate or pressure of hydraulic fluid into or out of a said second chamber to
15 cause a desired change in pressure taking into account the volume of the said
16 second chamber and an effective bulk modulus of the fluid volume.
17
- 18 15. The hydraulic fluid transmission according to any one preceding claim
19 wherein the controller is configured to synchronise a change in pressure of a
20 said second chamber with a corresponding change in pressure in one or more
21 said first chambers.
22
- 23 16. The hydraulic fluid transmission according to any one preceding claim
24 wherein the fluid driving surfaces of the said chambers are coupled to each
25 other such that the forces exerted by hydraulic fluid on the respective fluid
26 driving surfaces of the chambers are additive or subtractive.
27
- 28 17. A method of transmitting hydraulic fluid power, the method comprising:
29 providing an actuator apparatus having a plurality of chambers, each of the
30 chambers having a respective fluid driving surface configured to drive or be
31 driven by hydraulic fluid therein; selectively connecting said one or more first
32 chambers to one of a plurality of hydraulic fluid sources or sinks, at least two
33 of the said plurality of hydraulic fluid sources or sinks having different fluid
34 pressures; and controlling the pressure or flow rate of hydraulic fluid input to
35 or output from said one or more second chambers of the said plurality of
36 chambers by varying the pressure or flow rate of hydraulic fluid flowing into or
37 out of the said second chambers within a range of pressures or flow rates

- 1 respectively to thereby regulate the hydraulic fluid pressure in the said
- 2 chambers.

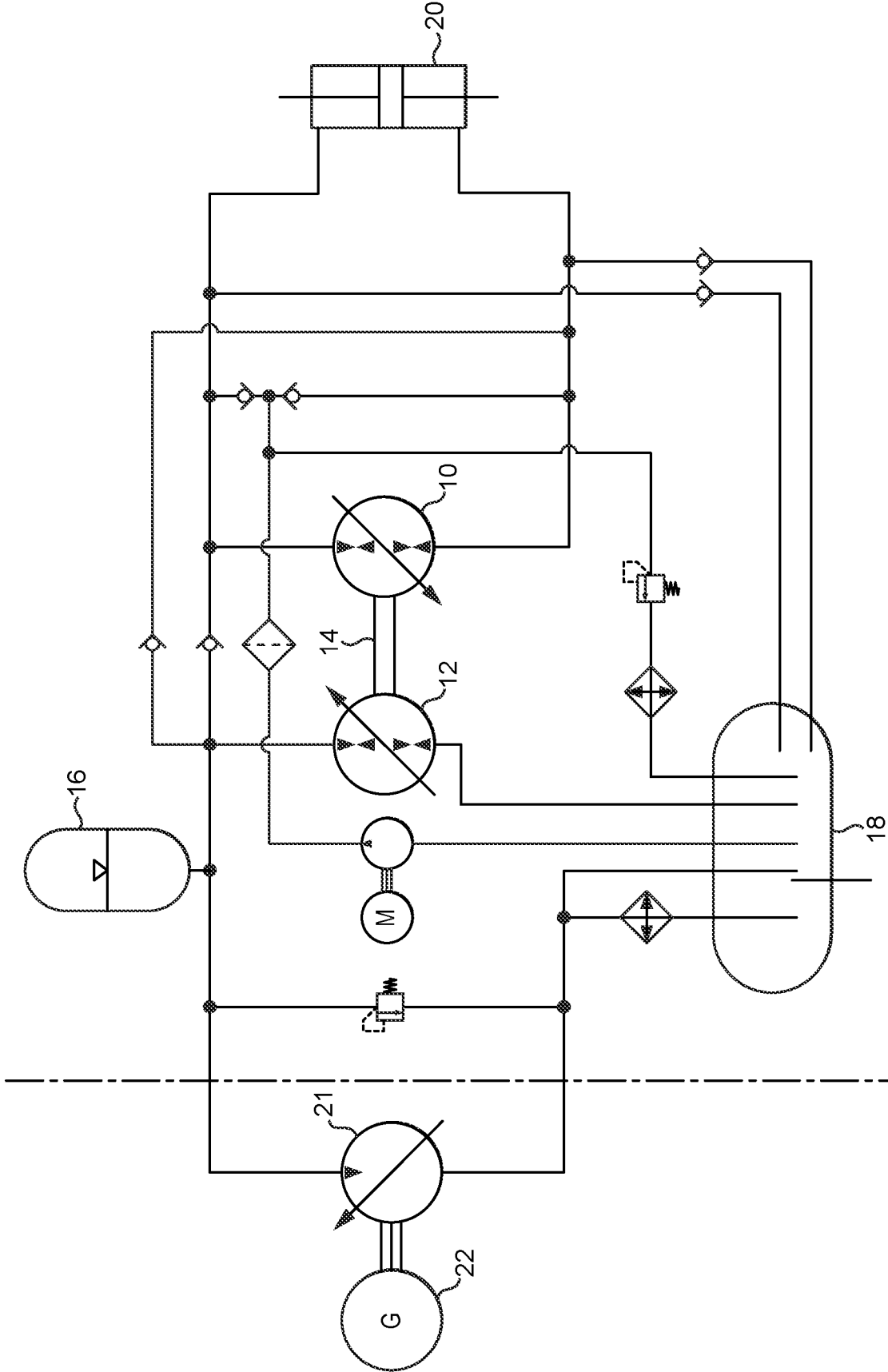


FIG. 1

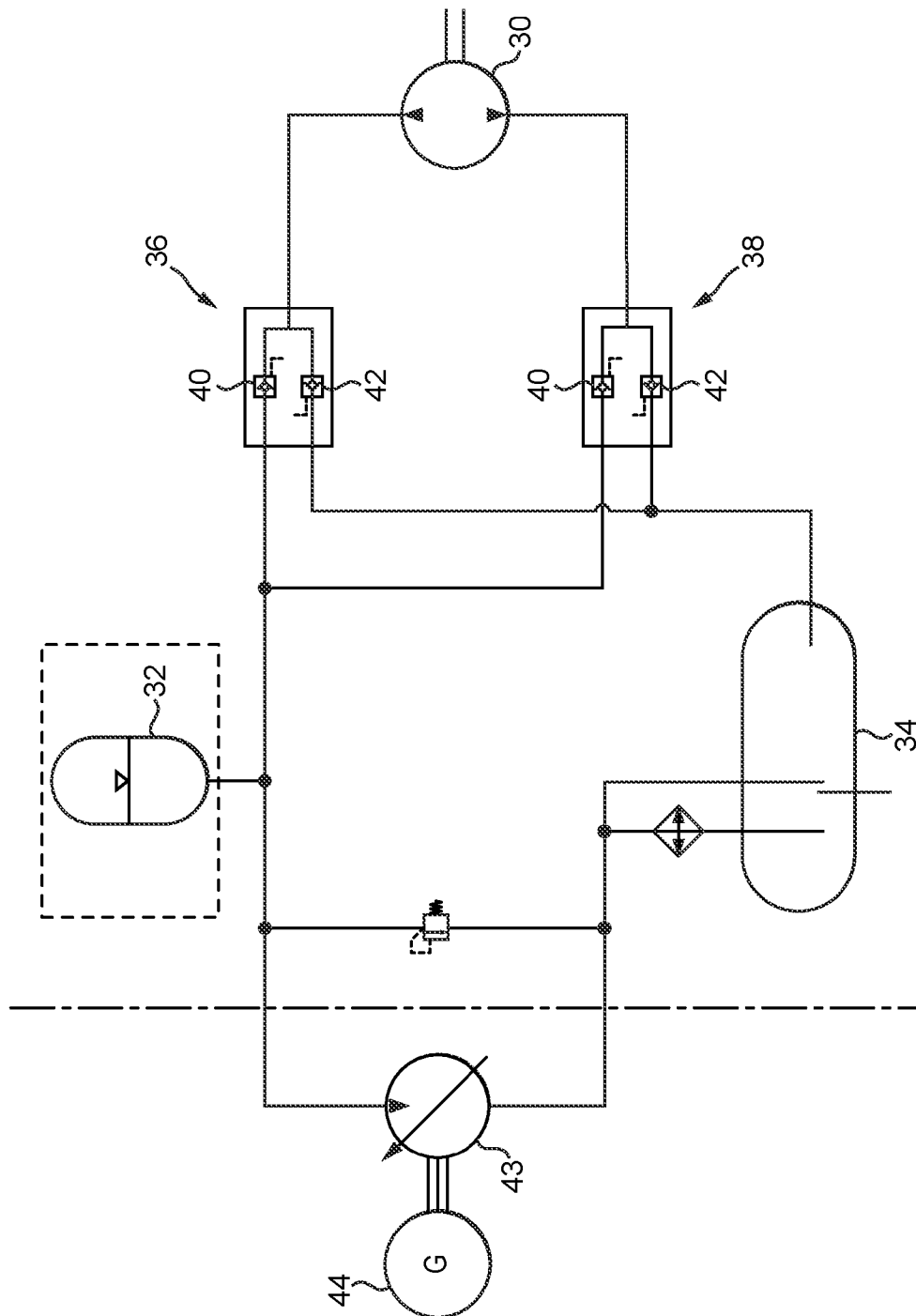


FIG. 2

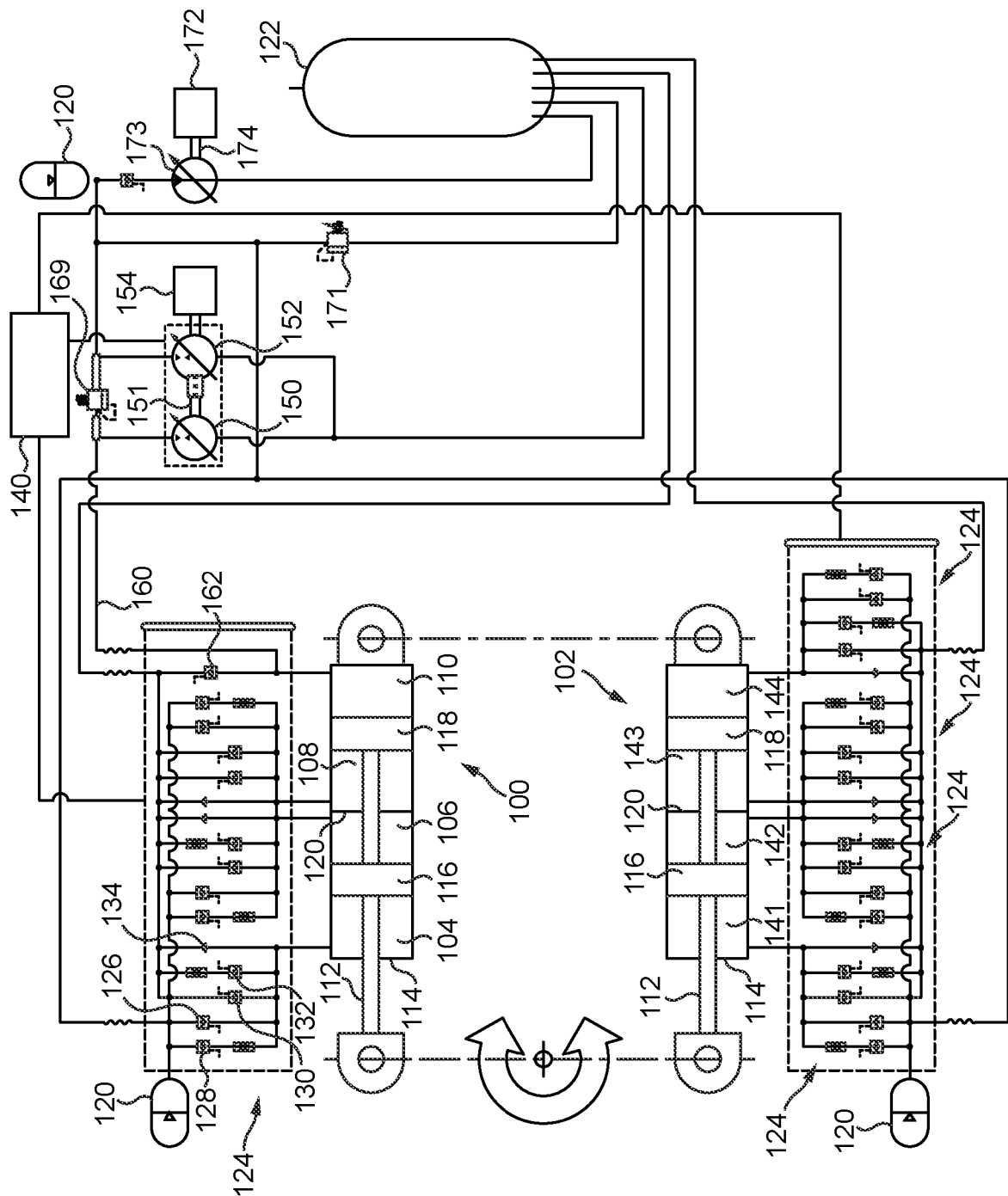


FIG. 3

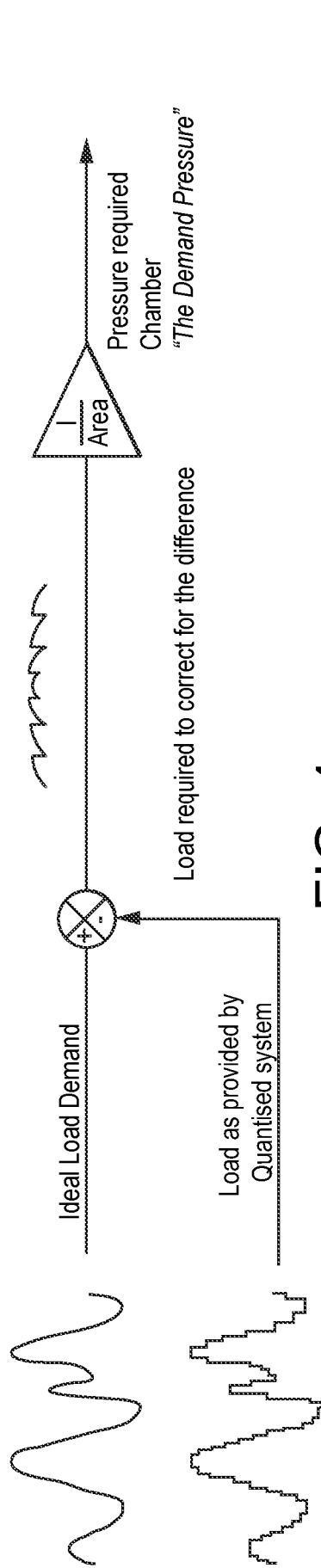


FIG. 4

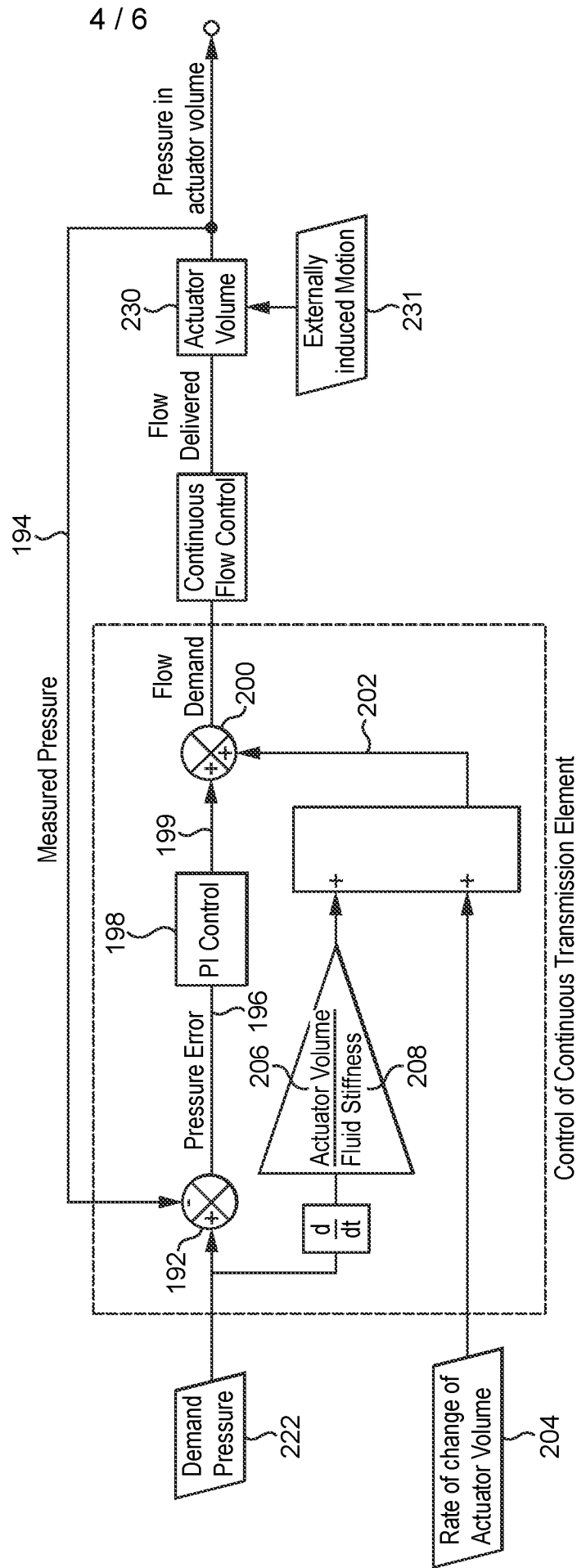


FIG. 5

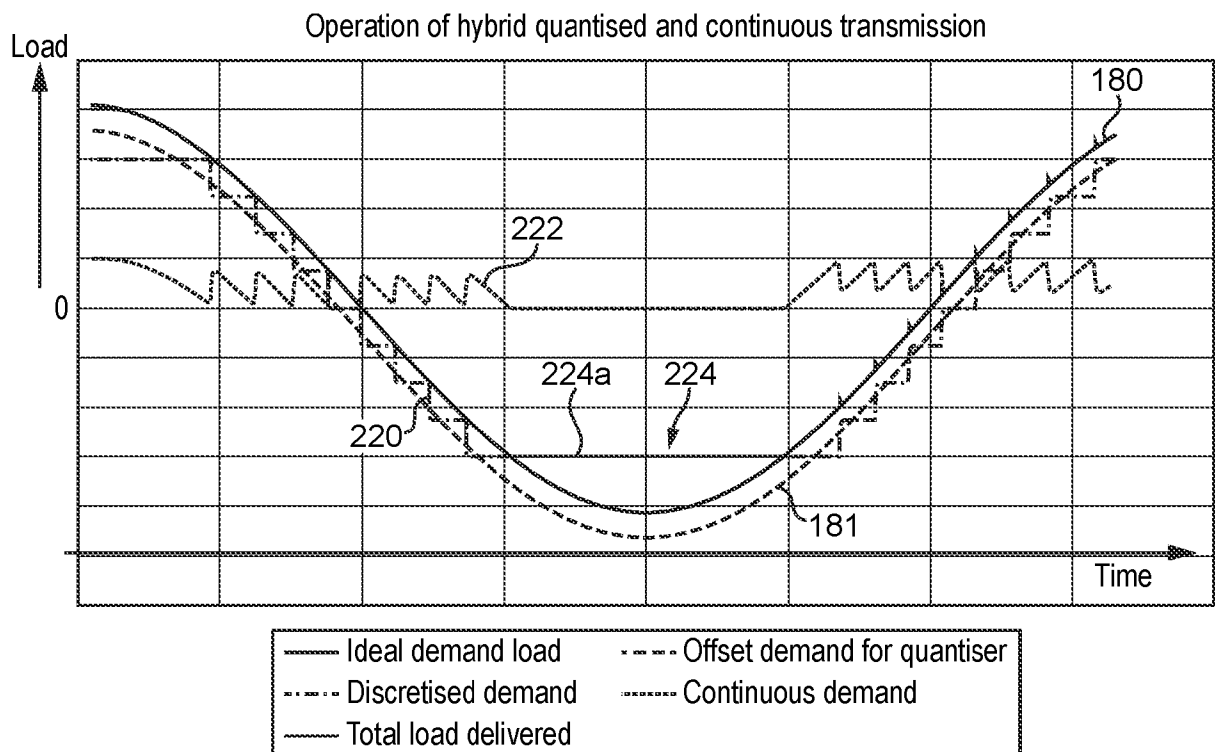


FIG. 6

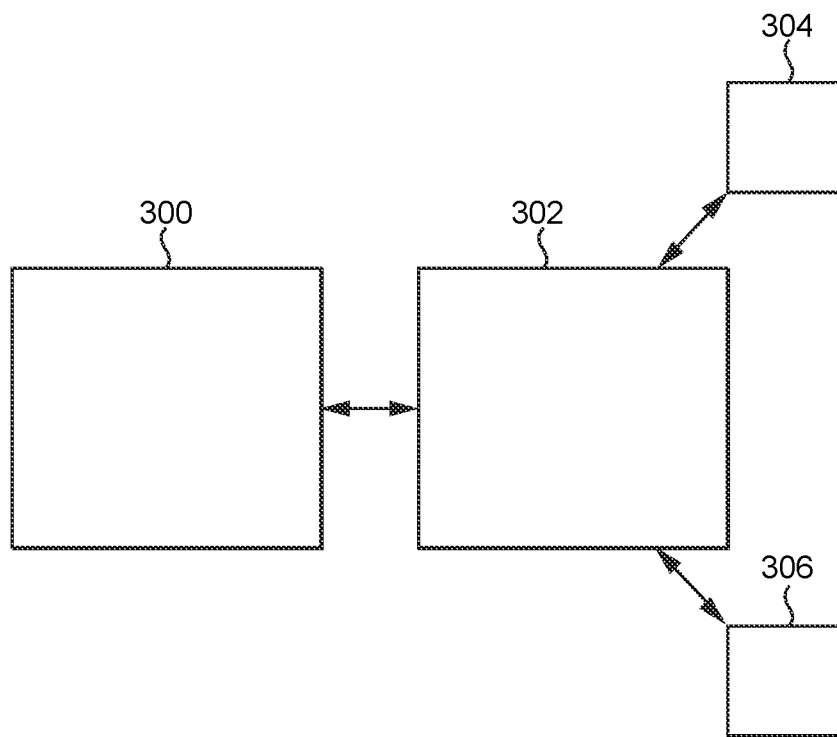


FIG. 7