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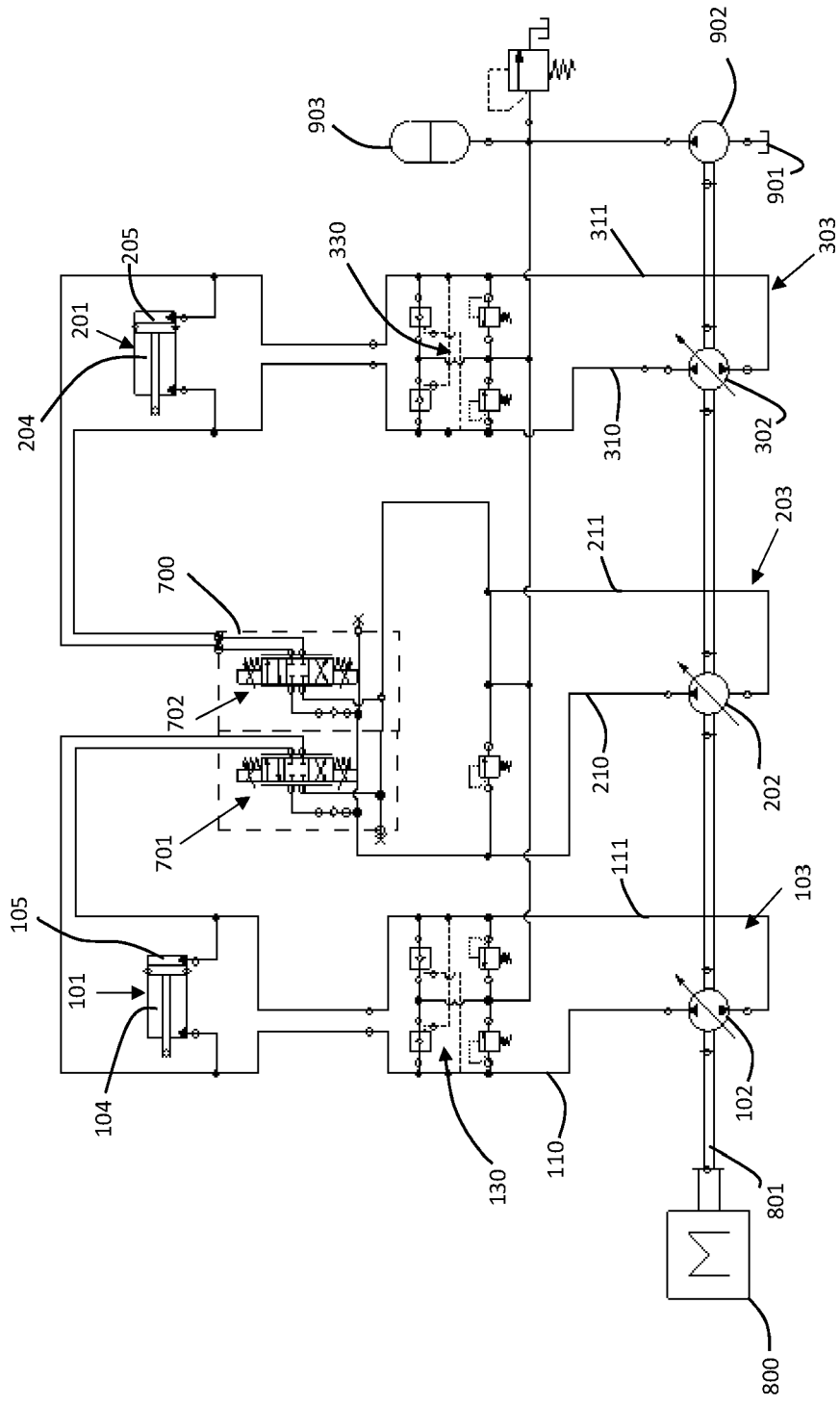


Figure 1a

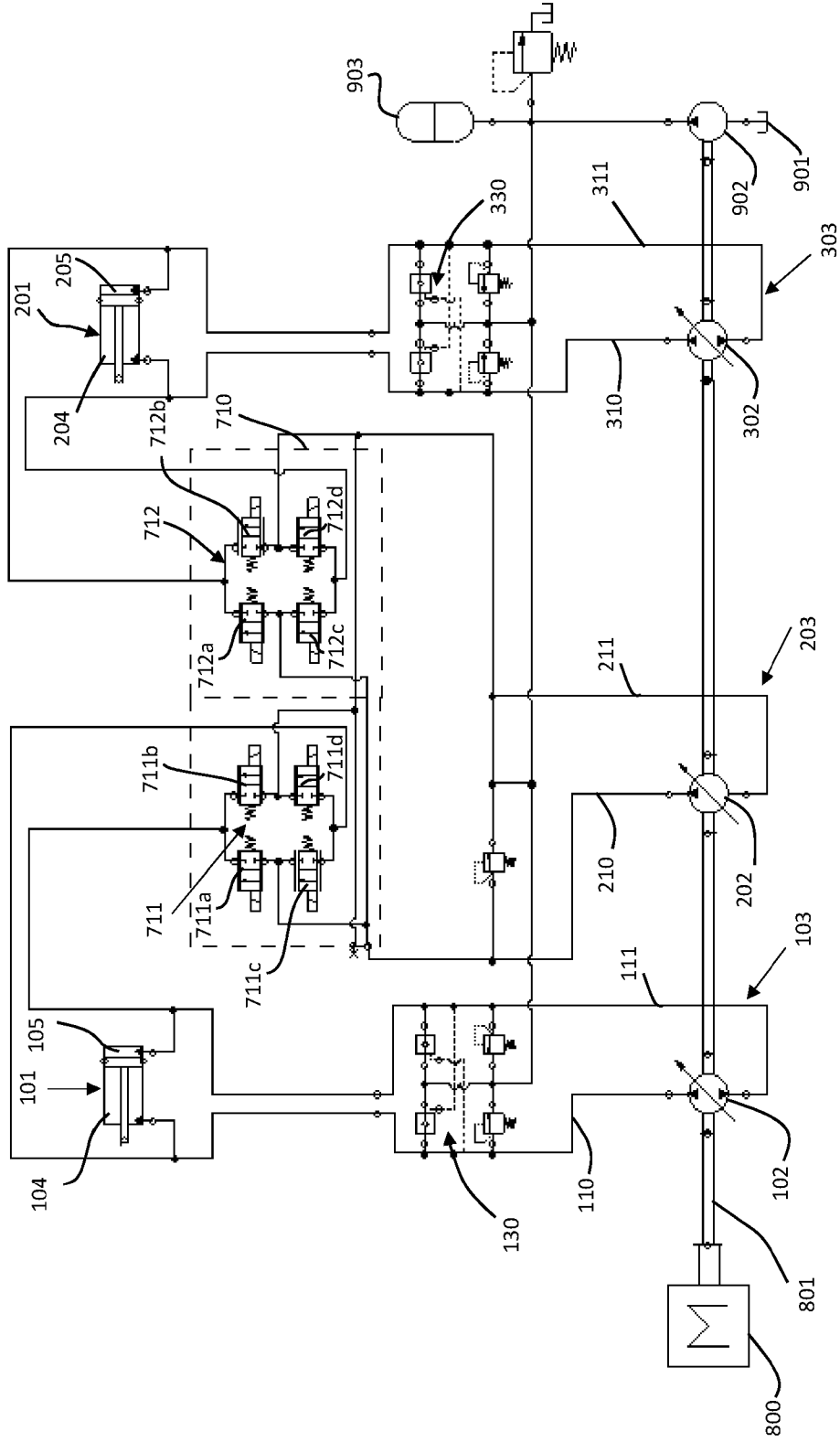


Figure 1b

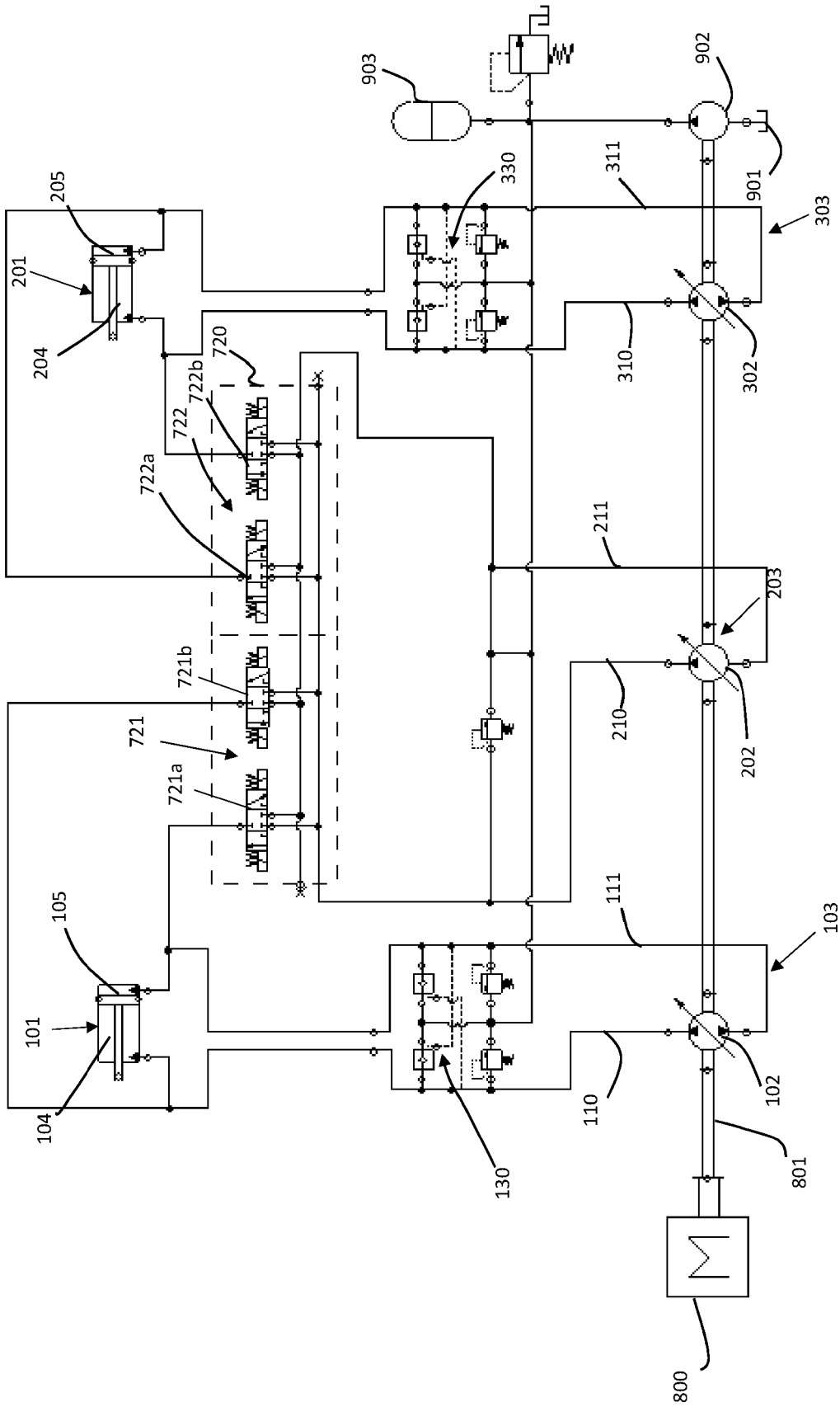


Figure 1c

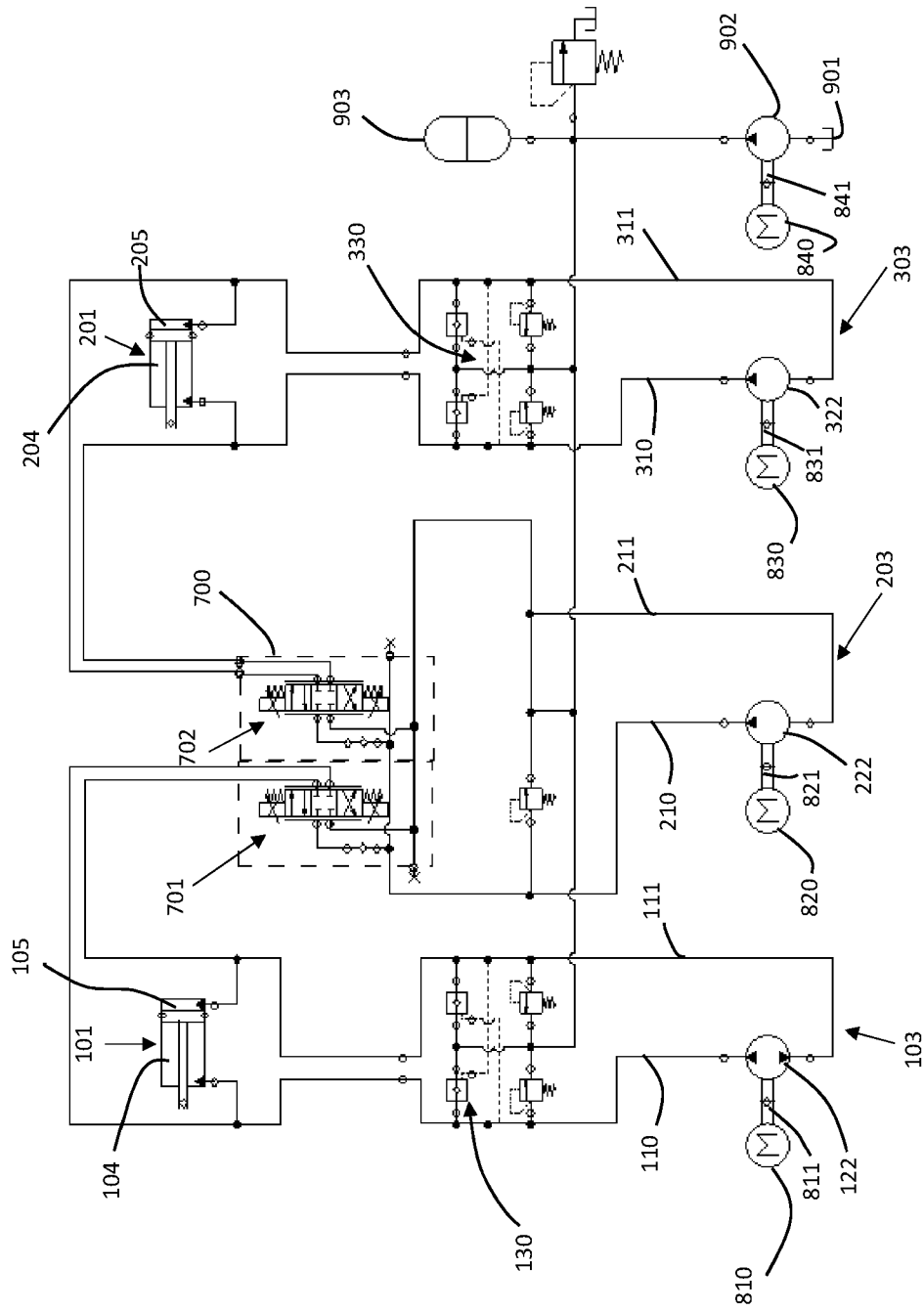


Figure 1d

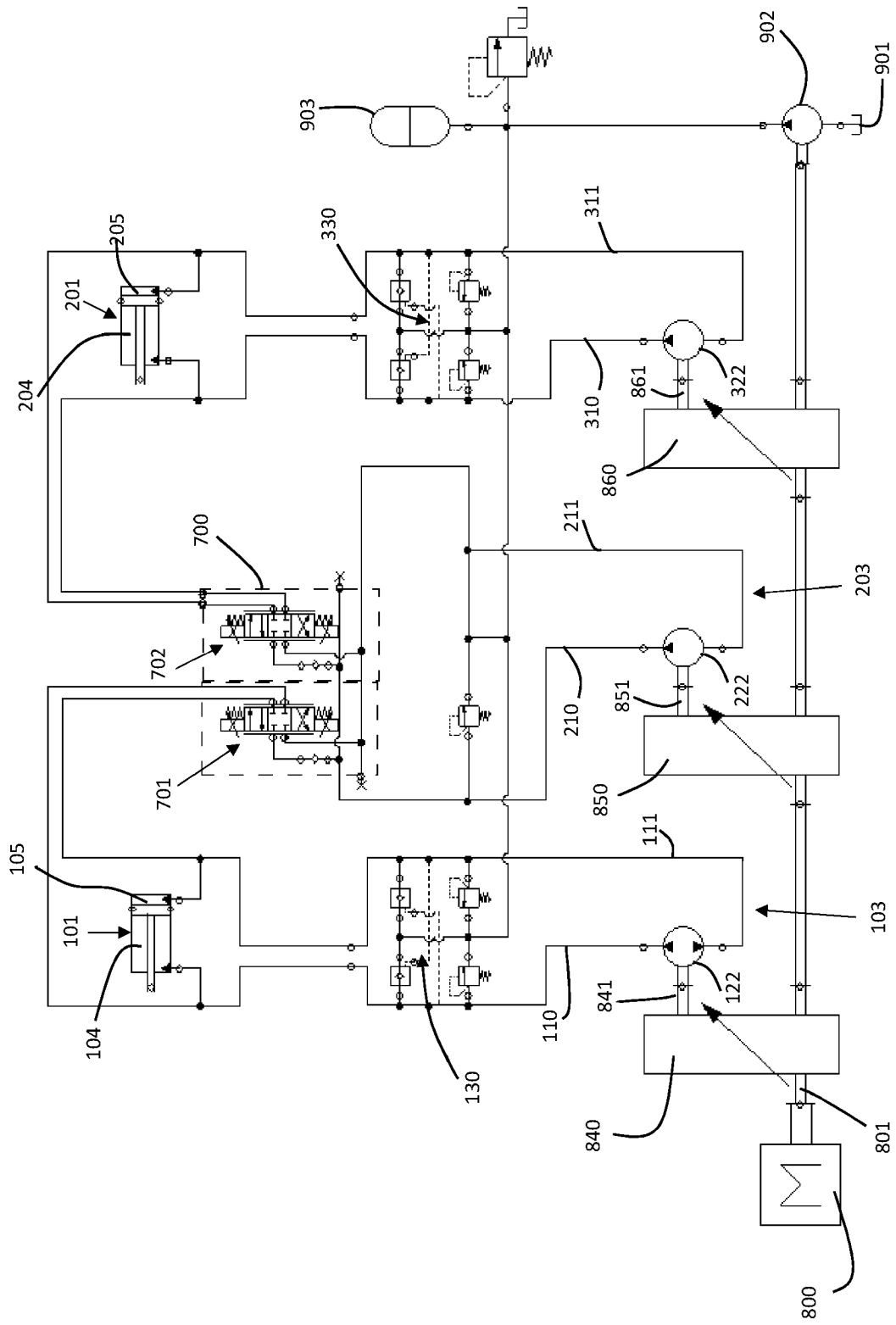


Figure 1e

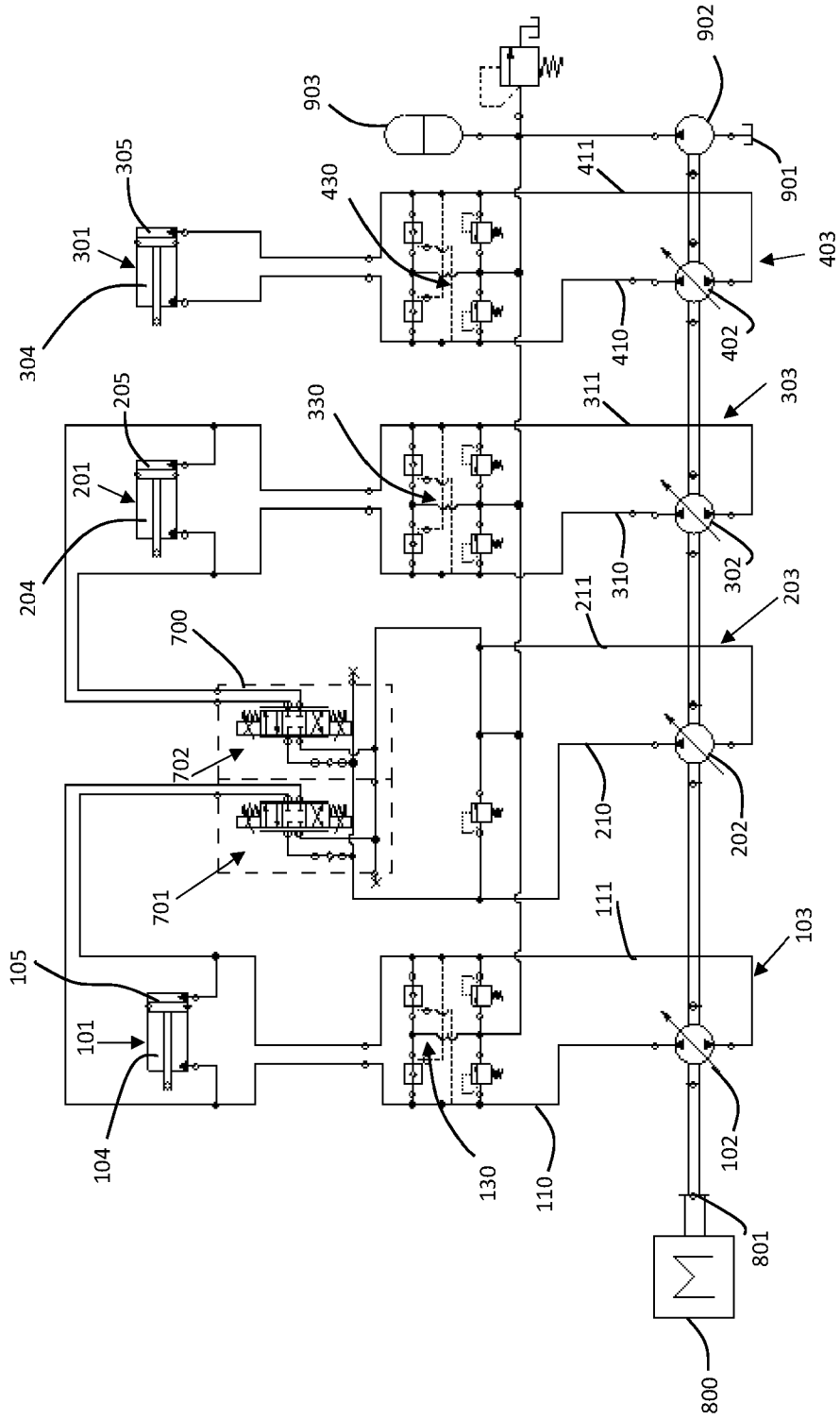


Figure 2

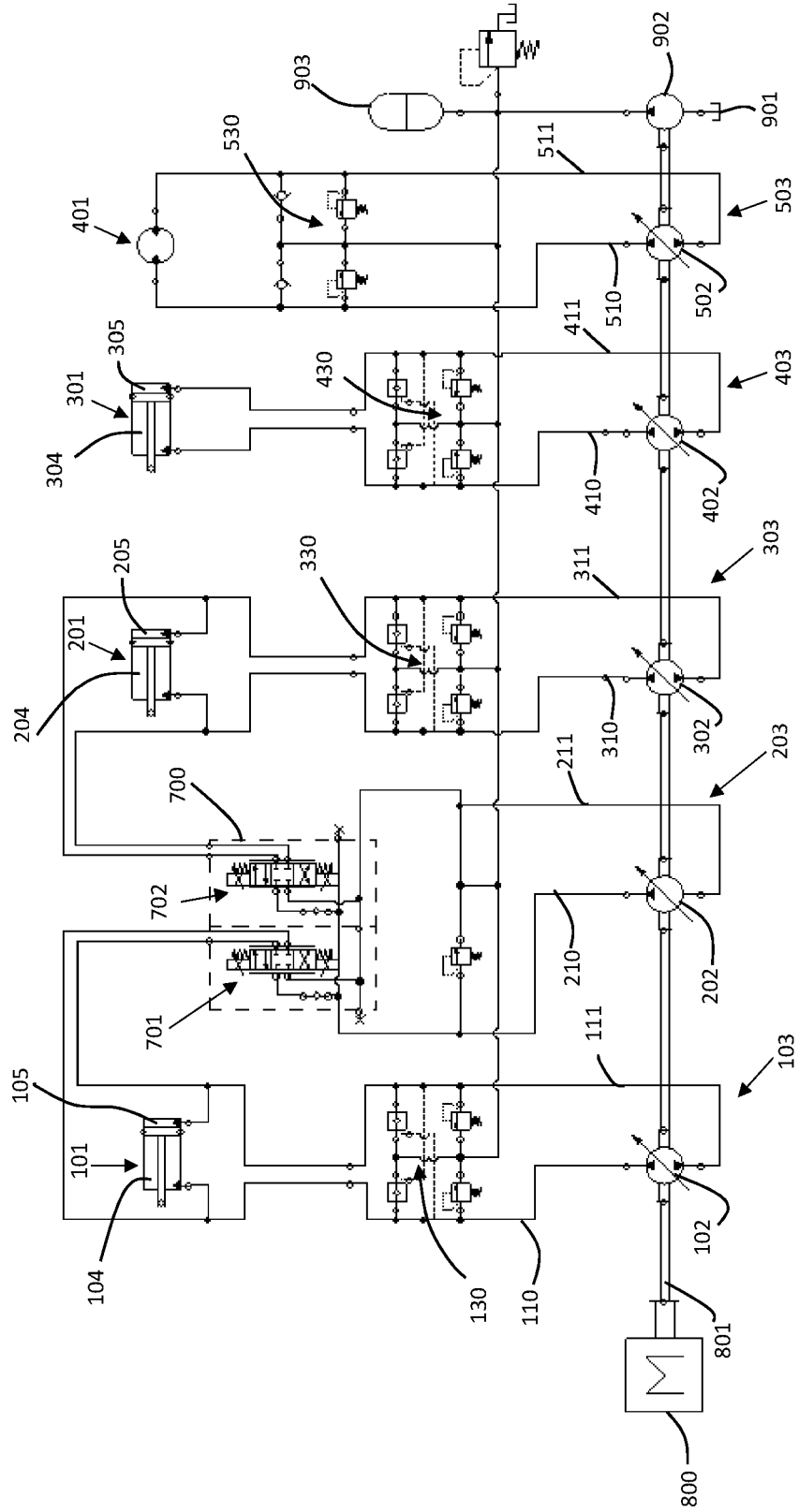


Figure 3



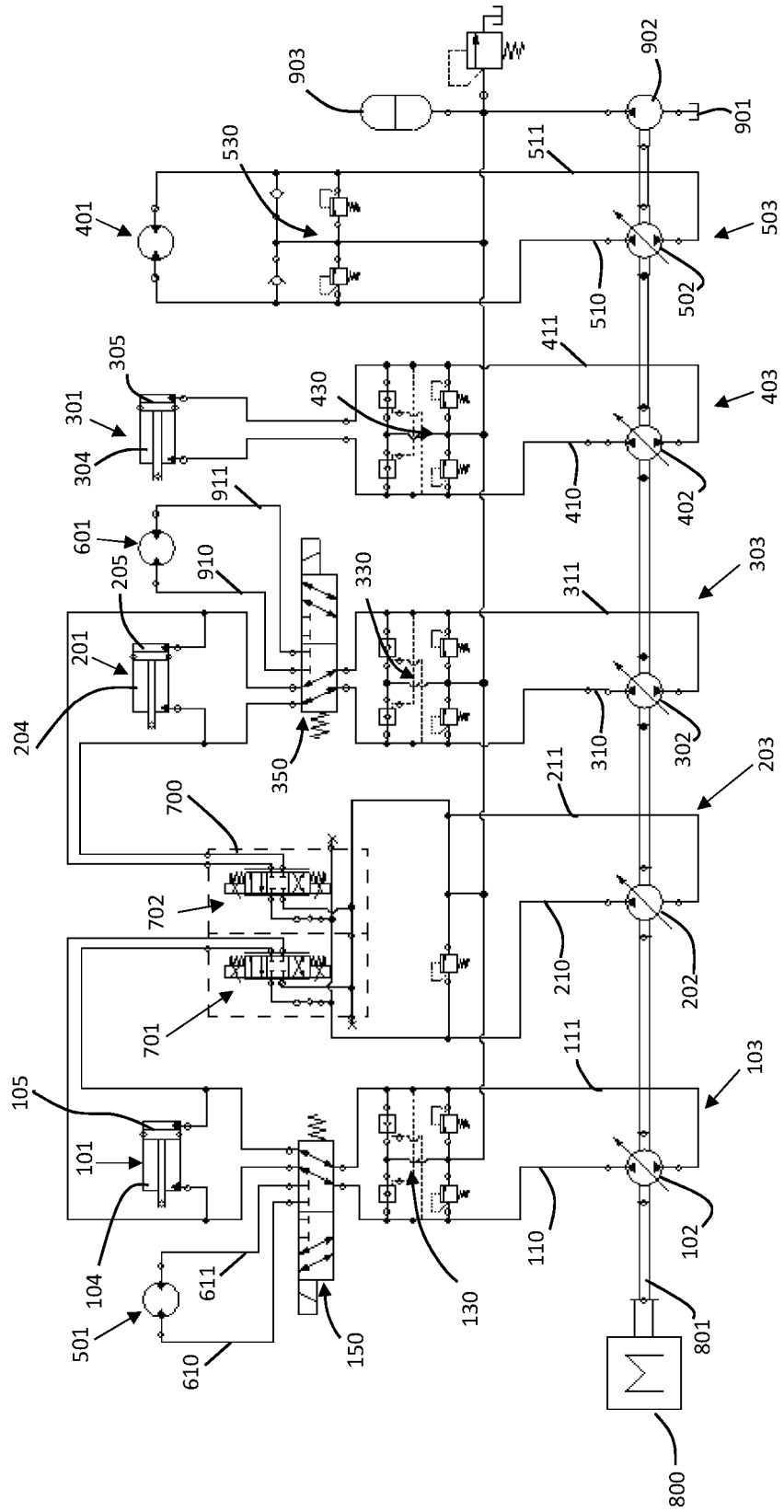


Figure 4

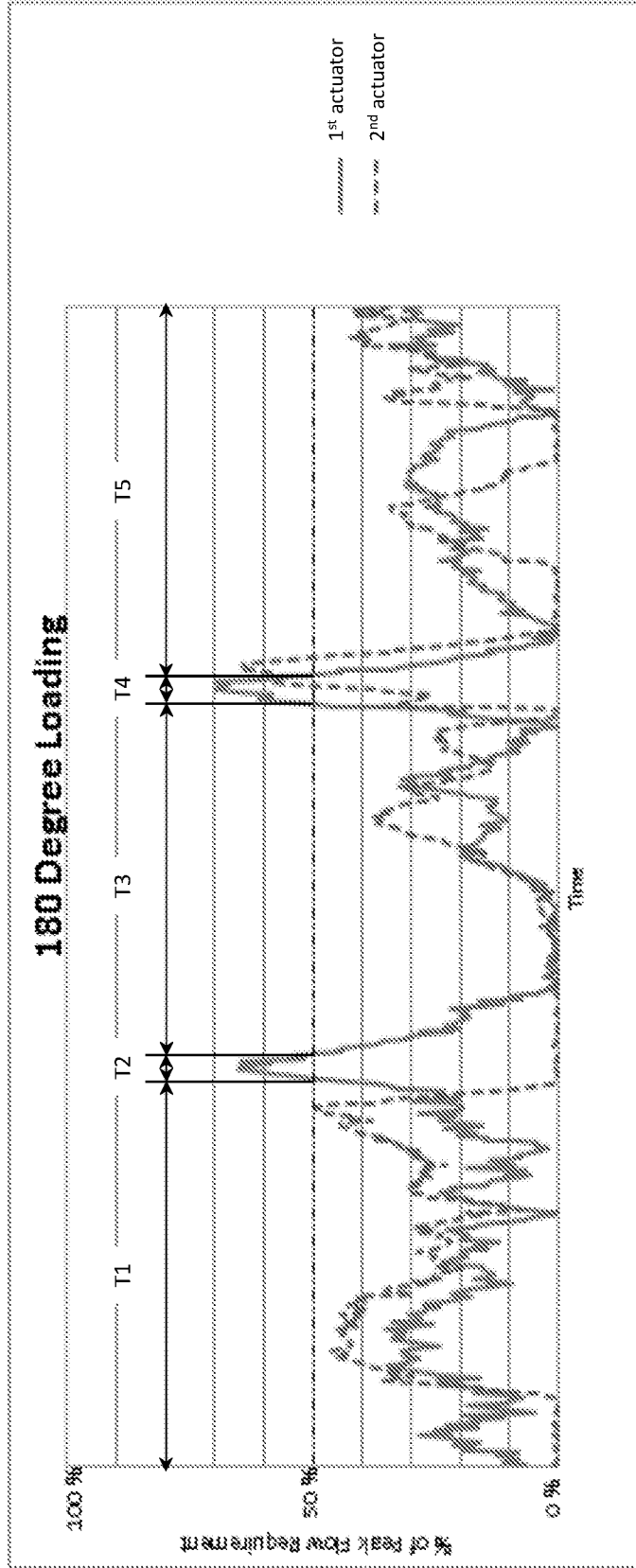


Figure 5

## Hydraulic Systems for Construction Machinery

The present invention relates to hydraulic systems, particularly hydraulic systems for construction machinery such as excavators. The invention further relates to construction machinery comprising the hydraulic system.

A variety of different hydraulic systems for construction machinery are known in the art. The hydraulic systems comprise several hydraulic actuators receiving a supply of pressurised fluid for actuating moveable members of the machinery, such as swing drives, booms, dippers, buckets, travel motors and other moveable parts of the respective construction machinery. In traditional hydraulic systems, depending on the size of construction machinery, one or more largely sized displacement pump/s is/are used to supply pressurised hydraulic fluid to all of the actuators of the respective machinery. To this end, the hydraulic displacement pump/s is/are each connected to several actuators by means of directional control valves, which connect the outlet port of the pump/s to all of the hydraulic actuators. The output flow of the hydraulic pump/s is therefore distributed between several actuators by means of proportional control valves. These so-called metering systems cause throttling of the flow through the control valves and are known to waste energy as a consequence.

In more recent developments, an alternative type of hydraulic system, which is known as a displacement controlled system or a meterless hydraulic system, was investigated in view of increased energy efficiency. Displacement controlled hydraulic systems comprise a plurality of hydraulic pumps, each of which is connected to a single actuator. The hydraulic pumps of displacement control systems are usually variable displacement pumps to selectively adjust the flow of pressurised fluid provided by the pump to its respective actuator. For example, to move an actuator at high speed, the flow of the respective pump is increased, while the flow is decreased if slower actuation of the actuator is required. Displacement controlled hydraulic systems are known to be more energy efficient than metering systems because the amount of flow directed to the actuators is controlled through variation of the pump output flow rather than restricting flow with proportional metering valves. In other words, the pumps of a displacement controlled hydraulic system are regulated to only discharge hydraulic fluid at a flow rate and pressure

28 10 21

necessary to move the actuators at the desired speed and force, and therefore do not incur energy losses through throttling of the fluid flow or reducing the pressure.

While displacement controlled hydraulic systems show significant improvements in energy efficiency, it was found that they are not commercially viable for utilisation in construction machinery, such as excavators. This is because known displacement controlled systems usually require the individual displacement pumps to be of large size in order to move the actuators at the desired speed (in excavators this speed is determined by the so-called cycle time needed to fully extend and retract an actuator in air). Implementing a plurality of largely sized pumps (one per actuator), however, significantly increases the manufacturing cost of displacement controlled system. Moreover, it is a known problem that large hydraulic pumps exhibit poor energy efficiency, when being operated at a reduced output flow rate, that is, if actuators are moved at slower speeds.

In view of the above, it is an object of the present invention to provide a hydraulic system that exhibits high fuel efficiency under high and low load/speed conditions. It is a further object of the invention to reduce manufacturing costs and improve energy efficiency compared to conventional displacement controlled hydraulic systems.

In a first embodiment, the invention relates a hydraulic system, comprising a first actuator and a first pump fluidly connected or connectable to the first actuator via a first circuit and adapted to drive the first actuator. The system further comprises a second pump connectable to the first actuator via a first control valve and a second actuator. A third pump is fluidly connected or connectable to the second actuator via a second circuit and adapted to drive the second actuator, wherein the second pump is connectable to the second actuator via a second control valve, and wherein the second pump is selectively and simultaneously connectable to the first and second actuators.

In simple terms, the hydraulic system of the present invention is a combination of a displacement controlled hydraulic system and a metering system. In more detail, the first and second circuits may be adapted as displacement controlled actuator circuits, which each include a variable displacement pump for actuating the first and second

actuators at different speeds/flow rates. The second pump, on the other hand, can be used assist actuation of the first actuator via a first control valve and/or assist actuation of the second actuator. This is particularly the case under high speed conditions, that is, when shorter cycle times for activation of the first and/or second actuators are required. It will be appreciated by the skilled practitioner that the actuation speed of one or more actuators of a construction machine is determined by the so called "cycle time", which relates to the time needed to fully expand and retract a respective hydraulic actuator in air. According to the present invention, the shortest cycle time, which will be referred to as the minimal cycle time, is achieved by combining the flow of the first and second pumps. It is a customer expectation that a machine is capable of achieving the minimal cycle time and this is a key metric used to judge the performance of construction machinery. Yet, it was found that in most duty cycles, the minimal cycle time only needs to be achieved occasionally and that an average duty cycle (i.e. for average digging work cycles) requires relatively low actuation speeds on average.

In view of the above the particular arrangement of the present invention permits for the first pump and third pumps to be sized smaller so as to be able to move the first and second actuators under normal/average speed conditions. Average speed requirements are ultimately determined via the demand of the operator of the machinery, during a particular duty cycle. If the first and/or second actuators are required to move quicker under certain circumstances, the fluid flow from the first pump and/or third pump can be assisted by a top up fluid flow from the second pump. Smaller sized pumps will reduce the cost of the hydraulic system when compared with traditional displacement controlled hydraulic systems that utilise large variable displacement pumps. Furthermore, it was found that using a plurality of smaller pumps will increase the efficiency of the entire hydraulic system. It should be understood that construction machinery may be provided with a plurality of different actuators, each of which could be supplied with flow from two or more different pumps to achieve the minimal cycle time, as will be described in more detail below.

In another embodiment, the first circuit is a closed loop circuit. The first circuit may be connected to a charge pump, which maintains the system at a slightly elevated fluid pressure, to prevent cavitation.

In a further embodiment, the second circuit is a closed loop circuit. In this case, the second circuit may be connected to a charge pump. Alternatively, the second circuit may be an open loop circuit, in which case the second pump draws hydraulic fluid directly from a fluid reservoir rather than being supplied with pressurised fluid from the charge pump.

According to another embodiment, the second pump is a variable displacement pump. This has the advantage that the top up fluid flow from the second pump can be adjusted to the precise requirements of the first and/or second actuators. Alternatively or additionally, the second pump may be a fixed displacement pump which is connected to the first actuator and/or to the second actuator via proportional control valves which can be used to adjust the flow of the fluid supplied from the fixed displacement second pump to the first and/or second actuator.

The first pump is directly connected/connectable to the first actuator, wherein the first control valve is part of a valve assembly and constructed as a first proportional control valve adapted to variably restrict a fluid flow from the second pump provided to the first actuator. In this specification, the term "directly connected" refers to an arrangement in which the pump is connected to the actuator directly via fluid lines that do not comprise proportional or reducer valves (throttles) that would introduce artificial flow restrictions, unlike metering circuits that require one or more proportional valves to distribute the fluid flow of the pump. In other words, the direct connection refers to a connection, which does not result in energy losses of the fluid flow, apart from unavoidable losses within the fluid lines and/or valves which are required for safety purposes such as hose burst check valves, load holding valves or on/off valves, which do not intentionally add additional flow metering to the circuit. Consequently, the first actuator will always receive substantially all of the output flow provided by the first pump. Due to the direct connection of the first pump with the first actuator, the first circuit can be described as a displacement controlled circuit. In contrast to this, the second pump is preferably connectable to the first actuator via a first proportional control valve (metering valve), which is adapted to only supply a predetermined portion of the second fluid flow to the first actuator. Consequently, the fluid circuit created by the second pump that is connected to the first actuator via

a metering/proportional valve, can be described as a metering circuit. As will be described in more detail below, the remaining portion of the second fluid flow, which is not used to support the flow of the first pump, may be applied to the second actuator simultaneously. As such, it is feasible for the second pump to assist the first pump in moving the first actuator, while simultaneously assisting movement of the second actuator.

The first proportional control valve is a directional, proportional spool valve. The first proportional spool valve is preferably a 4/3 spool valve. The 4/3 spool valve comprises four fluid ports and 3 position. A first fluid port may be connected to the high pressure port (or pump flow) of the first pump, whereas a second fluid port maybe connected to the low pressure port (or flow return) of the first pump. A third fluid port may be connected to a first chamber of the first actuator, whereas a fourth fluid port may be connected to a second chamber of the first actuator. In a first position, the 4/3 spool valve is closed and none of the fluid ports are connected. In a second position, the first and a fourth fluid port as well as a second and a third fluid port are connected. Accordingly, in the second position, the high pressure port of the first pump may be connected to the second chamber, while the low pressure port is connected to the first chamber of the first actuator, for extending the latter. In a third position, the first and third fluid ports as well as the second and fourth fluid ports are connected, to retract the first actuator. In this case, the second pump can be constructed as a uni-directional pump, as the 4/3 spool valve can be used to connect the high pressure/flow port and the low pressure/flow port of the unidirectional pump to the desired high/low pressure/flow inlet of the first actuator.

Similar to the first circuit, the third pump in the second circuit is directly connected or connectable to the second actuator, wherein the second control valve comprises a proportional control valve adapted to variably restrict a fluid flow from the second pump provided to the second actuator. Again, the term "directly" refers to the fact that the second circuit is a displacement controlled circuit, and hence has a third pump that is connected to the second actuator without any flow reducing components, such as proportional/metering valves. The second proportional control valve is a directional, proportional spool valve, preferably a standard 4/3 spool valve.

The first pump is configured as a bi-directional variable displacement pump and the second pump is configured as a uni-directional pump. According to this arrangement, the first pump is connected to the first actuator by a closed loop circuit and configured as a bi-directional pump to supply either of the actuator inlets selectively with pressurised hydraulic fluid. The second pump is connectable to both the first and second actuator via a directional control valve, and thus does not require a bi-directional pump. When using a uni-directional pump as the second pump, the top up circuit may either be constructed as an open or closed loop circuit.

According to another embodiment, the first pump comprises a first pump port connected or selectively connectable to a first chamber of the first actuator and a second pump port connected or selectively connectable to a second chamber of the first actuator. As the first pump is a bi-directional pump, both the first and second ports can be either be used as high or low pressure port. As such, when the first port of the first pump is a high pressure port, the first chamber of the first actuator is connected to a high pressure side of the pump, whereas the second port is then a low pressure port, hence connecting the second chamber of the actuator with a low pressure side of the pump. The opposite is the case, if the direction of the pump is reversed, such that the second port is the high pressure port. Consequently, supply of high pressure fluid from the first pump can be supplied to the first and/or second chamber of the first actuator. In another embodiment, load holding valves could be added between the ports of the pump and the chambers of the actuator. It should be understood that these load holding valves would not introduce a metering function. Accordingly, the first pump would still be "directly connected" to the first actuator.

In another embodiment, the second pump comprises a first port selectively connectable to the first or second chamber of the first actuator via the first control valve and a second port selectively connectable to the first or second chamber of the first actuator via the first control valve. The second pump of this embodiment is connectable to both chambers of the first actuator by means of the first control valve, which may be constructed as a standard 4/3 valve. As mentioned previously, this embodiment enables the second pump to be constructed as a uni-directional pump.



In another embodiment, the second circuit is constructed substantially identical to the first circuit and comprises a third pump with a first port connected or selectively connectable to a first chamber of the second actuator and a second port connected or selectively connectable to a second chamber of the second actuator. The first and second ports of the second pump may be selectively connectable to the first or second chamber of the second actuator via the second control valve.

In another embodiment, the first and second pumps and third pumps are connected to a prime mover by a common drive mechanism, such as a common drive shaft. Fourth and fifth pumps may be connected to the same prime mover via a second common drive shaft. The two drive shafts maybe connected to a gearing/variable ratio mechanism at the output of the prime mover in such a way that the first and second common drive shafts can be rotated at the same or different rotational speed. Accordingly, the first, second and third pumps are preferably driven at the same rotational input speed by means of the common drive shaft but may still provide different outlet flows. For example, the first, second and third pumps may be variable displacement swash-plate pumps, which may adjust their respective output flow rate independent of the rotational speed of the common drive shaft. Of course, this arrangement will render the hydraulic system of the present invention more compact and cost effective as only a single prime mover is required. As mentioned previously, the fourth and fifths pumps and potentially further pumps may preferably also be connected to the single prime mover via a second common drive shaft. It is also feasible to connect all of the pumps to a single common drive shaft. The invention is, however, not limited to a single prime mover driving the pumps via one or more common drive shafts. The skilled practitioner will appreciate that the pumps could be driven by one or more prime mover/s. The prime mover/s may be a fuel engine or an electric motor, either of which may be connected to the pump/s via a variable gear/ratio mechanism. There may be one prime mover per pump or one prime mover for all of the pumps.

According to another embodiment, the prime mover may be a single speed motor. Even if the motor is a single speed motor, it is feasible to drive the various pumps of the present system at different speeds by means of variable gear/ratio mechanisms. Accordingly, when using a single speed motor, each or some of the pumps maybe

connected to the motor via a common or separate variable drive mechanism/s. Alternatively, the prime mover may be an internal combustion engine, such as a diesel engine.

In another embodiment, the first pump is sized such that the maximum output flow rate of the first pump equals 25% to 75%, preferably 40% to 60% more preferably 45% to 55% of a peak flow rate necessary to drive the first actuator at a predetermined minimal cycle time. In other words, the first pump may be sized to provide a maximum flow rate sufficient to move the first actuator under regular speed requirements, which equal 25% to 75% of the speed/flow requirements to achieve the minimal cycle time, predetermined by the construction machinery manufacturer. In particular, the "minimal cycle time", relates to the shortest time needed to fully expand and retract a respective hydraulic actuator. For example, if the first actuator is a hydraulic ram used to lift the boom of an excavator, then the first pump may be sized to provide a maximum fluid flow rate that equates 25% to 75% of the flow rate required to lift and retract the boom at the a predetermined maximum speed, that is, 25% to 75% of the flow rate required to perform a full actuation cycle of the boom within the minimal cycle time. It should be noted that the cycle time is measured in air, i.e. when the boom does not have to work against any resistance other than gravity. In one exemplary embodiment, the predetermined minimal cycle time could be set to be about 5 seconds. In this example, the first pump would be sized such that the maximum flow rate provided by the first pump would be sufficient to achieve a longer cycle time of about 7.5 to 20 seconds. If an operator wishes to obtain the faster, minimal cycle time for actuating the boom, the maximum output flow rate of the first pump will not be sufficient to move the first actuator at the desired speed (i.e. to achieve the predetermined minimal cycle time) and hence assistance from the second pump will be required. It will be appreciated that the second pump is then sized complimentary to the first pump, such that a combination of the first and second pumps is sufficient to achieve the predetermined minimal cycle time. Of course, the invention is not restricted to the particular example of cycle times stated hereinbefore. In this regard, it should be appreciated that different cycle times, and hence different actuation speeds, apply to different actuators of construction machinery. For example, while the boom actuator of an excavator may need to

achieve a fastest/minimal (i.e. second) cycle time of 6 seconds, the minimal cycle time for a dipper actuator may be 4 seconds and 2.5 seconds for a bucket actuator.

Of course, the skilled person will appreciate the general requirement for the respective construction machinery to fulfil certain minimal cycle times, which are mainly determined by the customers demand. As such, the skilled practitioner is able to calculate the required maximum fluid flow rate value, which needs to be provided to move an actuator at a speed sufficient to achieve said minimal cycle time. The first pump will then be sized to exhibit a fluid flow that relates to 25% to 75% of the aforementioned maximum fluid flow rate value. It was found that sizing the first pump this way will result in substantially improved energy efficiency.

The hydraulic system of the present invention is restricted to working under normal/average speed conditions if only the first pump is used to supply the first actuator. However, the system is also configured to achieve the faster "minimal" cycle time by supplying the first actuator with pressurised fluid from the first and the second pump. That is, the hydraulic system of the present invention is also adapted to provide a second, higher fluid flow rate by combining the high pressure outlets of the first and second pumps. In contrast to this, commonly known displacement controlled hydraulic systems comprise heavily oversized displacement pumps for each actuator, which are capable of achieving the minimal cycle time independently, without assistance from other pumps. However, under regular speed conditions commonly known displacement pumps work at about 50% of their maximum outlet flow. Smaller pumps, according to this embodiment, that work at about 90% of their maximum outlet flow during normal working conditions are not only less expensive but work more efficiently.

In another embodiment, the hydraulic system comprises a controller connected to the first control valve and adapted to control the first control valve to selectively connect the second pump to the first circuit, if the maximum fluid flow output rate of the first pump is not sufficient to move the first actuator at high speed, that is, at shorter cycle times. In this embodiment, the controller may be connected to a sensor device connected to an operator interface. In particular, the sensor device may be connected to an input device, such as a joystick, used by the operator to

control movement of the first actuator. The desired actuation speed may be a function of the joystick position. It will be appreciated that according to one example, the desired speed may increase with the amount of displacement of the joystick. If the displacement sensed by the sensor device indicates a desired actuation speed/cycle time that exceeds the maximum fluid flow capability of the first pump, the controller will adjust the first control valve such that all or part of the second fluid flow from the second pump is diverted to the first actuator.

The first control comprises a proportional control valve. The proportional control valve may be connected to the controller such that the controller can adjust the proportional control valve such that the portion of the second fluid flow, which is directed to support the first pump in moving the first actuator, is sufficient to obtain the desired speed sensed by the sensor device. The controller may adjust the proportional control valve such that only a necessary amount of the top up fluid flow is supplied to the first circuit. The remaining parts of the top up fluid flow can simultaneously be used to move the second actuator.

In another embodiment, the third pump is sized such that the maximum output flow rate of the third pump equals 25% to 75%, preferably 40% to 60% more preferably 45% to 55% of a peak flow rate necessary to drive the second actuator at a predetermined minimal cycle time.

In another aspect, the second pump may be fluidly connectable to the second actuator via a second control valve to support the third pump in moving the second actuator at higher speed, to obtain faster cycle times as set out hereinbefore with respect to the first actuator. The valve assembly of this embodiment, comprising the first and second control valve, may be configured such that the second pump is fluidly connectable to the first and second actuator simultaneously or in sequence.

The aforementioned controller may also be adapted to control the second control valve to selectively connect the second pump to the second fluid circuit, if the maximum fluid output flow of the third pump is not sufficient to move the second actuator at high speed, i.e. at a predetermined minimal cycle time for the second actuator.

According to another embodiment, the first pump is sized to exhibit a maximum output flow, which is 50% to 150%, preferably 75% to 125%, more preferably 95% to 105%, of the maximum output flow of the second pump. Preferably, the third pump is also sized to exhibit a maximum output flow, which is 50% to 150%, preferably 75% to 125%, more preferably 95% to 105%, of the maximum output flow of the second pump. According to this embodiment, the first, second and third pumps are sized in a similar manner. As such, the first and second actuators can be moved with a maximum flow, which equates approximately twice the maximum output flow of the first or third pump respectively. Consequently, the faster, second cycle time (i.e. the minimal cycle time) can be reduced to 50% of the first cycle time. In the aforementioned example, the cycle time of the first actuator could thus be reduced from 10 seconds to 5 seconds, by combining the flow of the first and second pump in operating the first actuator.

In a particularly advantageous embodiment, the first, second and third pumps are identically sized, which reduces the cost of the present hydraulic system even further.

In another embodiment, the hydraulic system further comprises a third actuator and a fourth pump connectable to the fourth actuator via a third fluid circuit and adapted to drive the third actuator. The fourth pump is preferably directly connected to the third actuator. The third actuator may be a linear actuator, such as a hydraulic cylinder for moving an excavator bucket.

In another embodiment, the hydraulic system further comprises a fourth actuator and a fifth pump connectable to the fourth actuator via a fourth fluid circuit and adapted to drive the fourth actuator. The fifth pump is preferably directly connected to the fourth actuator. The fourth actuator may be rotary actuator, and in particular a hydraulic motor for slewing parts of a construction machine.

In another embodiment, the system further comprises a fifth actuator, wherein the first pump is selectively connectable to the fifth actuator. Preferably, the first pump is directly connectable to the fifth actuator, that is, via valves, which do not restrict the

fluid flow provided by the first pump. The valves can be constructed as a single diverter valve or a plurality of on/off valves.

In another embodiment, the system further comprises a sixth actuator, wherein the third pump is selectively connectable to the sixth actuator. The third pump is preferably directly connectable to the sixth actuator by means of valves, which do not restrict the flow provided by the third pump. The valves can be constructed as a single diverter valve or a plurality of on/off valves.

It should be understood that the aforementioned arrangement of the fifth and sixth actuator enable the operator to activate all of the six actuators simultaneously with only four pumps. For instance, while the first and third pumps might be used to activate the fifth and sixth actuator for tracking of the construction machine (e.g. excavator), the second pump may be utilised to drive the first and/or second actuator, via the first and/or second control valve. In an excavator, this would enable tracking of the machine at the same time as moving the dig end.

The present invention further relates to a construction machine comprising the hydraulic system described herein before.

Embodiments of the invention will now be described, by way of example only, with reference to the accompanying figure, in which:-

FIGURE 1a shows a schematic of a hydraulic system according to a first embodiment of the present invention;

FIGURE 1b shows a schematic of a hydraulic system according to a second embodiment of the present invention;

FIGURE 1c shows a schematic of a hydraulic system according to a third embodiment of the present invention;

FIGURE 1d shows a schematic of a hydraulic system according to a fourth embodiment of the present invention;

FIGURE 1e shows a schematic of a hydraulic system according to a fifth embodiment of the present invention;

FIGURE 2 shows a schematic of a hydraulic system according to a sixth embodiment of the present invention;

FIGURE 3 shows a schematic of a hydraulic system according to a seventh embodiment of the present invention;

FIGURE 4 shows a schematic of a hydraulic system according to an eighth embodiment of the present invention; and

FIGURE 5 shows the flow rate requirements of the first and second actuator during a typical duty cycle.

Figure 1a shows a schematic of a hydraulic system according to a first embodiment of the present invention. By way of example, this first embodiment of the first hydraulic system will be described below in connection with an earth moving device, such as an excavator. However, it should be understood that the hydraulic system shown in Figure 1a is not restricted to this application and is suitable for a variety of different machinery.

The hydraulic system comprises a first actuator 101 which is connected to a first pump 102 via a first circuit 103. The first actuator may be a linear actuator, such as a hydraulic cylinder. The first circuit 103 of Figure 1a is depicted as a closed loop circuit, containing the first pump 102 connectable to the first actuator 101. The first pump 102 is connectable to the first actuator 101 via first and second fluid lines 110, 111.

The first pump 102 is shown as a bi-directional, variable displacement pump, which is connectable to a first chamber 104 of the first actuator 101 via the first fluid line 110. A second outlet port of the first pump 102 is connected to a second chamber 105 of the first actuator 101 via second fluid line 111. Since the first pump 102 is a

bi-directional pump, pressurised fluid may be provided to the first chamber 104 via fluid line 110 or, alternatively, to chamber 105 via second fluid line 111. By changing the displacement of the first pump 102, the first actuator 101 may be operated at different speeds.

Figure 1a further shows a second pump 202, which is connectable to the first actuator 101 in a top up fluid circuit 203. The second pump 202 is selectively connectable to the first actuator 101 by means of a first control valve 701. The second pump 202 is further selectively connectable to a second actuator 201 by means of a second control valve 702. In particular, the first and second control valves 701, 702 are part of a valve arrangement 700, as depicted in Figure 1a. Both control valves 701 and 702 are constructed as solenoid actuated proportional spool valves. In more detail, both of the spool valves of the control valves 701 and 702 are 4/3 directional spool valve, which are biased towards their closed position. The control valves 701 and 702 could be separate units or built into a common valve block.

The second pump 202 is a uni-directional variable displacement pump, which is connectable via the second control valve 702 to the third actuator 201.

The second pump 202 is connectable to the first pump 102 by means of the first control valve 701. In detail, the second pump 202 is disconnected from the first actuator 101, when the first control valve 701 is in its rest position. In the first position of the first control valve 701 (downwards in Figure 1a), the high pressure port of the second pump 202 is connected with the second chamber 105 of the first actuator 101 and the low pressure port of the second pump 202 is connected to the first chamber 104 of the first actuator 101. This first position of the first control valve 701 can be used to assist the first pump 102 with extending the first actuator 101. When the first control valve 701 is in its second position (upwards in Figure 1a), the high pressure port of second pump 202 is connected to the first chamber 104 of the first actuator 101 and the low pressure port of the second pump 202 is connected to the second chamber 105 of the first actuator 101, thus assisting the first pump 102 with retracting the first actuator. It will be appreciated that the first and second pumps 102, 202 as well as the first control valve 701 are controlled in such a way that the



high pressure port of the first pump 102 and the high pressure port of the second pump 202 are always connected to the same chamber of the first actuator 101. Of course, the same applies to the low pressure ports of the first and second pumps 101, 202, which will also be connected to the same chamber.

The valve arrangement 700 is connected to a controller (not shown), which will regulate positioning of the first and second control valves 701 and 702 in response to demands for actuation speed of the first and second actuators 101, 201. Under normal/average conditions, the first pump 102 will independently provide the first actuator 101 with pressurised fluid in a displacement controlled manner. As such, the high pressure flow of the first pump 102 will be connected to the second chamber 105 if the piston rod of the first actuator 101 (linear actuator, such as hydraulic cylinder) shall be extended out of the cylinder housing (to the left in Figure 1a). In order to retract the linear actuator, the pumping direction of the first pump 102 is reversed such that the high pressure port of the first pump 102 is connected to the first chamber 104 and low pressure port is connected to the second chamber 105 of the first actuator 101. If the maximum fluid output flow of the first pump 102 is not sufficient to extend the first actuator 101 at the desired speed, the controller may transfer the first control valve 701 into its first position (downwards in Figure 1a), such that the high pressure outlet of the second pump 202 is connected to the second chamber 105 in order to assist the first pump 102 in extending the ram of the first actuator 101. If the maximum fluid output flow of the first pump 102 is not sufficient to retract the first actuator 101 at the desired speed, the controller may transfer the first control valve 701 into its second position (upwards in Figure 1a), such that the high pressure outlet of the second pump 202 is connected to the first chamber 104 in order to assist the first pump 102 in retracting the ram of the first actuator 101.

The first and second control valves 701 and 702 may be proportional spool valves such that the fluid flow/pressure supplied by the second pump 202 to the first and second actuators 101 and 201 can be distributed according to demand. That is, if only a small amount of additional flow/pressure is required to extend the first actuator 101 at the desired speed, the controller will adjust valve 701 such that only a small part of the second fluid flow supplied by the second pump 202 is diverted to the first

or second chamber 104, 105 of the first actuator 101. The remaining flow provided by the second pump 202 may therefore be used to drive/assist actuation of the second actuator 201 simultaneously.

Similar to the first actuator 101, the second actuator 201 shown in Figure 1a is again depicted as a linear actuator (particularly a hydraulic cylinder). The second actuator 201 may be used to move the dipper or arm of an excavator. The second actuator 201 is connected to a third pump 302 in a closed loop circuit 303. The third circuit 303 is substantially identical to the first circuit 103 and corresponding parts are labelled with reference numbers corresponding to the first circuit and increased by "200". Similar to the first circuit 103, the second pump 202 can be connected to the third circuit 303 via the second control valve 702 of the valve arrangement 700. As such, the second pump 202 can also be used to assist the movement of the second actuator 201, if the third pump 302 is not sufficient under high speed conditions, i.e. to achieve a predetermined minimal cycle time for the second actuator 201.

In the embodiment shown in Fig. 1a, the first and second pumps 102, 202 are driven by a common drive shaft 801, which connects each of the pumps 102, 202 to a single prime mover, shown as drive motor 800, such as a combustion engine or electric motor. The drive motor 800 is also connected to a charge pump 902 via the common drive shaft 801, as will be described in more detail below. The invention is not limited to this particular drive arrangement. For example, any prime mover could be used to drive the pumps and the pumps maybe connected to a plurality of prime movers via a plurality of drive shafts, examples of which are described below.

A second embodiment of the present hydraulic system is shown in the schematic drawing depicted in Figure 1b. Parts of the second embodiment, which are identical to the first embodiment in the drawing of Figure 1a are labelled with identical reference numbers. As will be appreciated, the second embodiment according to Figure 1b only differs from the embodiment of Figure 1a in that the valve arrangement 710 comprises first and second control valves 711, 712, which are constructed as bridge valves. Each of the bridge control valves 711, 712 comprises four independently controllable metering valves 711a, 711b, 711c, 711d, 712a, 712b, 712c, 712d. Each of the independent metering valves 711a, 711b, 711c, 711d,

712a, 712b, 712c, 712d is constructed as a normally closed 2/2 proportional solenoid valve. The independent metering valves 711a, 711b, 711c, 711d, 712a, 712b, 712c, 712d can be poppet or spool valves or any other kind of metering valve the skilled person would see fit. If the second pump 202 is used to assist the first pump 102 in driving the first actuator 101 to extend the piston rod, the controller moves the first metering valve 711a into its second position (towards the right in Figure 1b) to connect the high pressure outlet of pump 202 with the chamber 105 of the first actuator 101, via the first fluid line 210. At the same time, the controller opens independent solenoid valve 711d such that the first chamber 104 of the first actuator 101 is connected to the low pressure port of the second pump 202, via the second fluid line 211. If, on the other hand, the second pump 202 is used to retract the piston of the first actuator 101, the high pressure fluid port of pump 202 is connected to the first chamber 104, while the low pressure fluid port is connected to the second chamber 105. To this end, the controller opens independent valves 711c and 711b, while valves 711a and 711d remain closed.

The function of the second bridge control valve 712 of the valve arrangement 710 is substantially identical to the function of the first bridge control valve 711. Of course, in contrast to the first bridge control valve 711, the second bridge control valve 712 selectively connects the second pump 202 to the second actuator 201. It will be appreciated that the valve arrangements 710 of the second embodiment shown in Figure 1b allows for individual metering of the high pressure and low pressure fluid lines of the second circuit 203. For example, the first bridge control valve 711 allows for the high pressure fluid flow of the second pump 202 to be metered via independent metering valve 711a when extending the first actuator 101, while fluid being pushed out of the first chamber 104 of the first actuator 101 can be connected to the low pressure port of the second pump, without any metering along valve 711d. That is, the bridging valve arrangement of the second embodiment shown in Figure 1b allows for differential metering of the fluid flows in the first and second fluid lines 210, 211.

A third embodiment of the present hydraulic system is shown in Figure 1c. Parts of the third embodiment, which are identical to parts of the first embodiment according to Figure 1a are labelled with identical reference signs. The third embodiment

according to Figure 1c shows another valve arrangement 720, which differs from the valve arrangements 700 and 710 shown in Figures 1a and 1b. The valve arrangement 720 shown in Figure 1c has first and second control valves 721, 722, each of which include first and second independent metering spool valves 721a, 721b, 722a and 722b. Similar to the second embodiment of Figure 1b, the independent metering valves 721a and 721b can be used to meter the fluid flow in the first and second fluid lines 210, 211, between the second pump 202 and the first actuator 101, separately. Similarly, the first and second spool valves 722a, 722b of the second control valve 722 can be used to independently meter the fluid flow between the first and second fluid flow lines 210, 211 and the chamber 204, 205 of the second actuator 201.

As mentioned previously, the first and second pumps 102, 202 can be driven by any kind of prime mover such as an electric or fuel motor 800, which is connected to each of the pumps via a common connector shaft 801. In a fourth embodiment of the present invention, shown in Figure 1d, each of the pumps 102, 202, 302 and 902 is connected to a separate prime mover 810, 820, 830 and 840. In a particular embodiment of Figure 1d, the prime movers 810, 820, 830 and 840 are connected to their respective pump 122, 222, 322, 902 via connector shafts 811, 821, 831. The prime movers or motors 810, 820, 830, 840 are preferably adapted to drive the connector shaft 811, 821, 831 and 841 at varying revolution speeds, thereby varying the output flow rate of their respective pumps 122, 222, 322, 902. It will be appreciated that the first, second and third pumps 122, 222, 322 of this fourth embodiment may thus be fixed displacement pumps, as the output flow rate is controllable by varying the revolution speed of the individual connector shafts 811, 821, 831 via prime movers or motors 810, 820, 830. Alternatively, the motors 810, 820, 830, 840 may be single speed motors and comprise an adjustable gearing mechanism, which connects the output of the motor 810, 820, 830, 840 with the connector shafts 811, 821, 831, 841 so as to drive the connector shafts 811, 821, 831, 841 at varying revolution speeds.

According to a fifth embodiment shown in Figure 1e, the hydraulic system again comprises a single prime mover or motor 800 adapted to drive a common shaft 801, similar to the first embodiment of Figure 1a. Again, identical parts of the fifth

embodiment are labelled with identical reference numbers. In contrast to the first embodiment, the fifth embodiment of Figure 1e shows variable ratio mechanisms 840, 850, 860 arranged between the common drive shaft 801 and the first, second and third pump 122, 222, 322 respectively. The variable ratio mechanism 840 connects a drive shaft 841 of the first pump 122 to the common drive shaft 801 of the motor 800. A second variable ratio mechanism 850 connects a second drive shaft 851 of the second pump 222 to the common shaft 801. A third variable ratio mechanism 860 connects a third drive shaft 861 of the third pump 322 to the common shaft 801. The variable ratio mechanisms 840, 850 and 860 are adapted to convert the revolution speed of the common drive shaft 801 into a revolution speed of the first, second and third drive shaft 841, 851, 861 desired to drive the first, second or third pumps 122, 222, 322 respectively. As such, the variable ratio mechanisms 840, 850, 860 can have any commonly available form, such as gearing, belt or chain mechanisms. Similar to the fourth embodiment of Figure 1d, it is thus not required to provide variable displacement pumps, such as swash plate pumps, and hence the pumps 122, 222, 322 are illustrated as fixed displacement pumps. Of course, it will be appreciated that variable displacement pumps could still be implemented as the first and second pumps.

A typical duty cycle of the first and second actuators 101, 201 is shown in Figure 5. In particular, Figure 5 shows a duty cycle of an excavator performing a 180 degree loading process. In this example, the first actuator is a boom actuator, whereas the second actuator is an arm/dipper actuator of the excavator. The chart shows the flow requirements of the first and second actuators 101, 201 at different times during the 180 degree loading duty cycle. The solid line represents the flow provided to the first actuator 101, whereas the dashed line refers to the flow provided to the second actuator 201. It will be appreciated by the skilled person that different flow rates are required at different times of the duty cycle. In this particular example, the flow rates required by the first actuator (solid line in Figure 5) shows two distinct peaks, while for most of the duty cycle, the flow requirements are relatively low. A very similar behaviour is shown for the second actuator (dashed line in Figure 6), which only comprises a single distinct peak.

In particular, the chart of Figure 5 shows a percentage of the peak flow required by the first and second actuators 101, 201 at any point during the 180 degree loading duty cycle. It should be understood that the 100% horizontal line refers to a peak flow that can be provided to the first or second actuators respectively by combining the fluid flows of the first and second or third and second pumps respectively. As such, the 100% relates to the peak flow rate required to achieve the minimal cycle time as defined hereinbefore.

Evidently, the first and second actuators 101, 201 only require less than 50% of the peak flow rate during most of the duty cycle shown in Figure 5. As mentioned previously, the first and third pumps 102, 302 can be sized such that their maximum output flow equals 25 to 75%, more preferably 45 to 55%, of the peak flow rate necessary to drive the first actuator at said minimal cycle time. If, as an example only, the maximum fluid output rate of the first and third pump 102, 302 equals 50% of the peak flow rate required to actuate the first and second actuators 101, 201 at a speed sufficient to obtain the minimal cycle time, then any fluid flow requirement below the 50% horizontal line shown in Figure 5 can be provided by only using the first or third pump 102, 302.

With particular reference to the graph of the first actuator (solid line), this means that during time intervals T1, T3, and T5 shown in Figure 5, the first actuator can be supplied exclusively with fluid flow from the first pump 102, without the need of extra fluid flow from the second pump 202. Only during time intervals T2 and T4, that is when the first actuator is moved at higher speeds (i.e. higher flow rates and shorter cycle times are required), is assistance needed from the second pump 202. In other words, the fluid flow of the first pump 102 is assisted by fluid flow from the second pump 202 only during intervals T2 and T4. It should be understood that the duty cycle shown in Figure 5 only refers to a typical 180 degree loading cycle, and thus other duty cycles may have substantially higher or lower flow requirements. However, it has generally been found that peak flow in the respective actuators is only rarely requested by the operator, and thus most of the duty cycle is performed at flow rates relating to 25 to 75% of the peak flow. Accordingly, sizing the first and third pumps to produce a maximum output flow, which relates to 25 to 75% of the peak flow was found to increase the energy efficiency of the system significantly.

A sixth embodiment is shown in Figure 2. Parts of the sixth embodiment, which are identical to parts of the first embodiment according to Figure 1a are labelled with identical reference signs. The sixth embodiment shows an additional, third actuator 301, which is connected to a fourth pump 402 via a fourth fluid circuit 403. The third actuator is depicted as a further linear actuator, such as a hydraulic cylinder for actuation of an excavator bucket. Similar to the first actuator 101, the third actuator comprises first and second chambers 304 and 305, which are connected to separate ports of the bi-directional fourth pump 402. It will be appreciated that the illustrated fourth fluid circuit is preferably self-sufficient, that is fourth pump 402 is sufficiently sized to drive the third actuator (e.g. excavator bucket) at any speed that may be required by the operator. However, the skilled reader will understand that the third circuit 303, too, could be connected to the second, top-up pump 202, for example via a third control valve, similar to the first and second control valves 701, 702.

While the sixth embodiment of Figure 2 shows a motor 800 and spool valves 701, 702 equivalent to Figure 1a, it will be appreciated that the alternative valve arrangements and prime movers shown in Figures 1b to 1e could also be utilised in the hydraulic system shown in Figure 2.

A seventh embodiment of the present invention is shown in Figure 3. Figure 3 mostly corresponds to the sixth embodiment shown in Figure 2 and corresponding parts are labelled with identical reference signs.

The hydraulic system shown in Figure 3 further comprises a fourth actuator 401, which is connected to a fifth variable displacement pump 502 in a fourth closed loop circuit 503. The fourth actuator 401 may be a rotary actuator, such as a slew motor that can be used to slew an excavator about a vertical axis. The fifth pump 502 of this embodiment is a bi-directional variable displacement pump which is connected to first and second inlet ports of the fourth actuator 401 via first and second fluid lines 510, 511. As can be derived from Figure 3, the fourth circuit 503 is not connected to any of the first to fourth circuits 103, 203, 303 and 403. However, it is generally feasible to arrange the second pump 202 of the second circuit 203 connectable to the fourth actuator 401 via valve arrangement 700.

As depicted in an eighth embodiment in Fig. 4, the first and third pumps 102, 302 can further be connectable to fifth and sixth actuators 501, 601. In more detail, the first pump 102 can be connected to inlet ports of the fifth actuator 501 via third and fourth fluid lines 610, 611. The connection between the first pump 102 and the fifth actuator 501 may be shut off by diverter valve 150, when the first actuator 101 is in use. Similarly, the diverter valve 150 may be used to shut off the connection between the first pump 102 and the first actuator 101, when the first pump 102 is used to drive the fifth actuator 501. The fifth actuator 501 may be a rotary actuator, which is used as a travel motor for one of the tracks of an excavator (i.e. left track). Accordingly, the first pump 102 is not only configured to supply the first actuator 101 with pressurised fluids, but can also to supply the fifth actuator 501, sequentially, to drive the left track of the excavator.

When the first pump 102 is connected to the fifth actuator 501 via the diverter valve 150 (state not shown), the first actuator 101 is shut off from the first pump 102. Yet, it is still feasible to drive the first actuator 101 via the second pump 202 when the first pump 102 is used to drive the fifth actuator 501. As such, the system of Figure 4 can be used to drive the fifth actuator 501 by means of first pump 102 and, at the same time, activate the linear first actuator 101 by means of the second pump 202, which is connected to the first actuator 101 via the first control valve 701.

The third pump 302 is, in turn, connectable to the sixth actuator 601 via third and fourth fluid lines 910, 911 and diverter valve 350. Accordingly, the third pump 302 can be used to sequentially provide the second actuator 201 and the sixth actuator 601 with pressurised fluid. The sixth actuator 601 is configured as a rotary actuator, such as a travel motor for driving the remaining track of an excavator (i.e. right track). Similar to the first actuator 101, the second actuator 201 can be actuated at the same time as the sixth actuator 601 by connecting the second pump 202 to the second actuator 201.

In conclusion, when tracking the excavator via the fifth and sixth actuators 501, 601, the first and second pumps 102, 302 of the eighth embodiment shown in Figure 4 are exclusively used for tracking purposes. If the first and second actuators 101, 201



shall be used during tracking, the respective fluid flow is exclusively provided by second pump 202 via the control valves 701, 702 of the valve arrangement 700.

In the embodiment shown in Figs. 1a, 1b, 1c, 2, 3 and 4, the first, second, third, fourth and fifth pumps 102, 202, 302, 402, 502 are driven by a common drive shaft 801 which connects each of the pumps 102, 202, 302, 402, 502 to a single prime mover or drive motor 800, such as a combustion engine or electric motor. The drive motor 800 is also connected to a charge pump 902 via the common drive shaft 801. As mentioned previously in connection with Figs. 1d and 1e, the invention is not limited to this particular drive arrangement. For example, any prime mover could be used to drive the pumps and the pumps maybe connected to a plurality of prime movers via a plurality of drive shafts, as shown in Fig. 1d. Alternatively, the pumps could be connected to a common drive shaft via variable ratio mechanisms as depicted in Fig. 1e.

The charge pump 902 is configured to maintain the system pressure of the hydraulic system by supplying pressurised fluid from a hydraulic reservoir 901 to the fluid circuits. To this end, each of the fluid circuits comprises an anti-cavitation arrangement 130, 230, 330, 430, 530, with check valves that allow the charge pump 902 to maintain a slightly elevated pressure. Each of the anti-cavitation systems 130, 230, 330, 430, 530 further comprises pressure relief valves to avoid high pressure damages during operation of the respective fluid circuits.

The invention is not restricted to the particular embodiments described with reference to the embodiment shown in the attached illustration. In particular, the third, fourth and fifth pumps 102, 202, 302, 402, 502 may be fixed or variable displacement, uni- or bi-directional and/or reversible/non-reversible pumps. Similarly the first, second, third, fourth, fifth and sixth actuators 101, 201, 301, 401, 501, 601 are not restricted to the particular applications shown but may be any type of actuator suitable for moving respective parts of a construction machine.

## CLAIMS

1. A hydraulic system comprising:

- a first actuator;
- a first pump fluidly connected or connectable to the first actuator via a first circuit and adapted to drive the first actuator;
- a second pump connectable to the first actuator via a first control valve;
- a second actuator;
- a third pump fluidly connected or connectable to the second actuator via a second circuit and adapted to drive the second actuator,

wherein the second pump is connectable to the second actuator via a second control valve, and wherein the second pump is selectively and simultaneously connectable to the first and second actuators;

wherein the first pump is directly connected or connectable to the first actuator, and wherein the first control valve comprises a directional, proportional spool valve adapted to variably restrict a fluid flow from the second pump provided to the first actuator;

wherein the third pump is directly connected or connectable to the second actuator, and wherein the second control valve comprises a directional, proportional spool valve adapted to variably restrict a fluid flow from the second pump provided to the second actuator; and

wherein the first pump is configured as a bidirectional variable displacement pump and the second pump is configured as a unidirectional pump.

2. The hydraulic system of claim 1, wherein the first circuit is a closed loop circuit.

3. The hydraulic system of claim 1 or 2, wherein the second circuit is a closed loop circuit.

4. The hydraulic system of any of claims 1 to 3, wherein the first pump is a variable displacement pump and/or wherein the second pump is a variable displacement pump.
5. The hydraulic system of any preceding claim, wherein the first directional, proportional spool valve is a 4/3 spool valve.
6. The hydraulic system of any preceding claim, wherein the second directional, proportional spool valve is a 4/3 spool valve.
7. The hydraulic system of any preceding claim, wherein the first pump comprises a first port connected or selectively connectable to a first chamber of the first actuator and a second port connected or selectively connectable to a second chamber of the first actuator.
8. The hydraulic system of claim 7, wherein the second pump comprises a first port selectively connectable to the first or second chamber of the first actuator via the first control valve and a second port of the third pump is selectively connectable to the first or second chamber of the first actuator via the first control valve.
9. The hydraulic system of any of claims 1 to 8, wherein the third pump is configured as a bidirectional variable displacement pump and the second pump is configured as a unidirectional pump, and wherein the second control valve is a directional control valve.
10. The hydraulic system of claim 9, wherein the third pump comprises a first port connected or selectively connectable to a first chamber of the second actuator and a second port connected or selectively connectable to a second chamber of the second actuator.
11. The hydraulic system of claim 10, wherein a first port of the second pump is selectively connectable to the first or second chamber of the second actuator via the second control valve and a second port of the third pump is selectively connectable to the first or second chamber of the second actuator via the second control valve.

12. The hydraulic system of any of claims 1 to 11, wherein the first, second and third pumps are connected to a single drive motor via a common drive shaft.

13. The hydraulic system of any of claims 1 to 12, wherein the first pump is sized such that a maximum output flow rate of the first pump equals 25% to 75%, preferably 40% to 60%, more preferably 45% to 55%, of a peak flow rate necessary to drive the first actuator at a predetermined minimal cycle time.

14. The hydraulic system of claim 13, wherein the hydraulic system comprises a controller connected to the first control valve and adapted to control the first control valve to selectively connect the second pump to the first actuator, if the maximum fluid output flow of the first pump is not sufficient to move the first actuator at a speed necessary to obtain the minimal cycle time for the first actuator.

15. The hydraulic system of any of claims 13 to 14, wherein the third pump is sized such that a maximum output flow rate of the third pump equals 25% to 75%, preferably 40% to 60%, more preferably 45% to 55%, of a peak flow rate necessary to drive the second actuator at a predetermined minimal cycle time.

16. The hydraulic system of claim 15, wherein the hydraulic system comprises a controller connected to the second control valve and adapted to control the second control valve to selectively connect the second pump to the second actuator, if the maximum fluid output flow of the third pump is not sufficient to move the second actuator at a speed necessary to obtain the minimal cycle time for the second actuator.

17. The hydraulic system of any of claims 1 to 16, wherein the first pump is sized to exhibit a maximum output flow which is 50% to 150%, preferably 75% to 125%, more preferably 95% to 105%, of a maximum output flow of the second pump.

18. The hydraulic system of any of claims 1 to 17, wherein the third pump is sized to exhibit a maximum output flow which is 50% to 150%, preferably 75% to 125%, more preferably 95% to 105%, of a maximum output flow of the second pump.

19. The hydraulic system of one of claims 1 to 18, wherein the first actuator is a linear actuator.
20. The hydraulic system of claim 19, wherein the first actuator is a hydraulic cylinder for displacement of an excavator boom.
21. The hydraulic system of one of claims 1 to 20, wherein the second actuator is a linear actuator.
22. The hydraulic system of claim 21, wherein the second actuator is a hydraulic cylinder for displacement of an excavator arm.
23. The hydraulic system of one of claims 1 to 22, wherein the system further comprises a third actuator connected or connectable to a fourth pump via a third circuit, wherein the third actuator is a linear actuator.
24. The hydraulic system of claim 23, wherein the third actuator is a hydraulic cylinder for displacement of an excavator bucket.
25. The hydraulic system of any of claims 1 to 24, further comprising a fourth actuator and a fifth pump connectable to the fourth actuator via a fourth circuit and adapted to drive the fourth actuator.
26. The hydraulic system of claim 25, wherein the fourth actuator is a rotary actuator.
27. The hydraulic system of claims 25 or 26, wherein the fourth actuator is a hydraulic motor for slewing a parts of a constructions machine.
28. The hydraulic system of any of claims 1 to 27, wherein the system further comprises a fifth actuator, wherein the first pump is selectively connectable to the fifth actuator.

29. The hydraulic system of any of claims 1 to 28, wherein the system further comprises a sixth actuator, wherein the third pump is selectively connectable to the sixth actuator.

30. A construction machinery, comprising the hydraulic system of any of claims 1 to 29.