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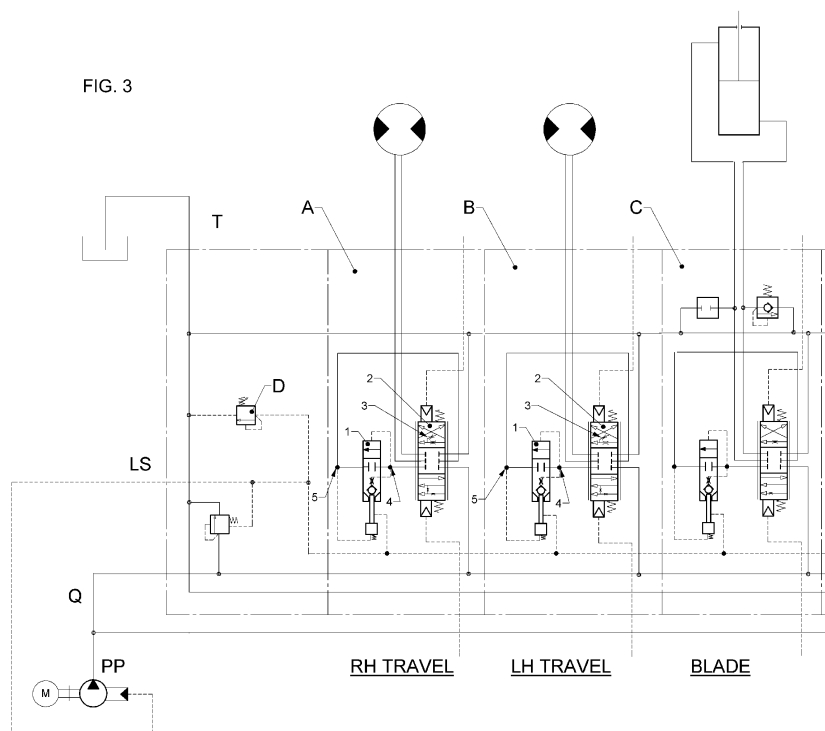
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(54) **Method for limiting the maximum power required by the hydraulic system of an earth-moving machine and directional control valve operating said method**

(57) Method for limiting the maximum power required by the hydraulic system of an earth-moving machine, controlled by a sectional flow-sharing having multiple sections (A, B, C), each section comprising a local compensator (1) and a spool (2), at least one section (A or B) or both (A and B) of which requiring the maximum flow and at least one additional section (C) intended to actuate a high pressure use. It is provided for bypassing the local

compensators (1) and the section(s) (A, B) that require the maximum flow in the moment in which it/they is/are actuated and concurrently completely relative to the third section (C) at high pressure, imparting a delivery pressure lower than that required by the section (C) and calculated so that the power required is less than or equal to the power that can be delivered by the motor M or less than the power at which the torque limiter intervenes, with the effect that the flow does not undergo reductions.



Description

APPLICATION FIELD OF THE INVENTION

[0001] The present finding is directed to the earth-moving machines field, in particular of the excavators controlled by a hydraulic system comprising at least one sectional flow-sharing directional control valve having multiple elements or sections (each element or section being provided with a spool and a compensator), at least one relief valve, at least one pump, and at least one motor capable of providing the power required.

[0002] More precisely, the present finding finds application in a well-specified and particular functioning configuration of the machine, i.e., that involving the complete and simultaneous actuation of at least one section of the directional control valve requiring all the flow that can be delivered by the pump and of at least another high pressure section.

[0003] An example of that is the case where two sections of the sectional directional control valve, termed "travel" in the jargon, control the excavator translation while the third one controls a cylinder: when both said travel sections, which usually require together the maximum flow, are completely and concurrently actuated, and the third section is in turn actuated, for example, at the relief valve calibration with the cylinder at the end of its stroke, the required may come to exceed the one that can be delivered by the motor, which consequently turns off.

STATE OF THE ART

[0004] In order to avoid exceeding the power that can be delivered by the motor, the current LS pumps are often provided with a torque limiter that is calibrated so that, if the required power, that is nothing else than the product of the flow and the pressure, exceeds that that required can be delivered by the motor, it intervenes by reducing the power.

[0005] To reduce the required power, the torque limiter has to reduce at least one of the involved parameters (pressure and/or flow), and particular it intervenes on the flow by automatically reducing, as a function of the pressure, the maximum inclination possible of the pump plate and then the maximum flow.

[0006] In so doing, the required power falls back in the limits of the one that can be delivered by the motor, and the behaviour of the directional control valve, that continues to be consistent with the flow-sharing functional concepts, is at the maximum pressure (coincident to that of the relief valve added to the flow losses); and the maximum flow that is delivered continues to go to the travel sections, however, it being reduced because of the torque limiter intervention on the pump, the travel sections slow down.

[0007] The result is that upon actuating a third high pressure element, the machine translation slows down,

to then accelerate again when it is released.

[0008] This machine behaviour, even not being a malfunctioning, but the logical consequence of the system functioning, is not desired by the operator; besides, however, it is neither acceptable that the system requires a power higher than the one that can be delivered by the motor.

EXPOSITION AND ADVANTAGES OF THE FINDING

[0009] It is the object of the present discovery to obviate the above-cited drawbacks, i.e., the slowdown and acceleration of the machine translation following the actuation of a further section to that or those already completely actuated and requiring the maximum flow that can be delivered, by reducing the power required from the motor intervening on the pressure instead of on the flow, as is typical of the prior art.

[0010] The present invention reduces the maximum pressure to a value such that, when multiplied by the maximum flow, the required power is always lower than that can be delivered by the motor, so that the section at the maximum flow (for example, the two travel sections previously mentioned) is not slowed down in case of the actuation of a third section at the end of the stroke of the cylinder.

[0011] The pressure increase upon the actuation of a third high pressure section is generated, as better illustrated in the detailed descriptive part, by the local compensators of the sections that require the maximum flow (for example, the two travel sections), which throttle the passage (and so the flow) towards the travels because they are subjected to the Load Sensing signal coming from the actuated third member.

[0012] By bypassing the local compensators of such sections, the pressure would not increase, the torque limiter would not intervene, and therefore the travel sections would continue to work at the maximum speed also during the actuation of a further section at the end of the stroke; however, in this manner, the travel sections would not act as flow-sharing with the other ones anymore.

[0013] Instead, by reducing the maximum pressure in each functioning condition of the machine, the operations requiring high pressures would result to be thereby penalised, such as, for example, the excavating operation of the excavator; therefore, the optimum would be to reduce such maximum pressure only when the two travel sections are completely and concurrently actuate, to then make it to return to the relief valve value when the two travel sections are no more completely actuated. The present invention, in the three implementation solutions thereof, looks for a functioning compromise, always in a flow-sharing logic, to achieve the best functionality of the machine.

[0014] According to a first functioning logic of the finding, which finds application in the first implementation solution, it is made sure that the compensators "inhibition" only occurs with the section(s) at the maximum flow

being actuated at the end of their stroke, while the compensator works properly in intermediate positions.

[0015] Therefore, if the maximum pressure that can be delivered by the local compensators of the sections at the maximum flow (it is reminded that the system is of the flow-sharing type, with LS functioning) is calculated so that the product of the flow and the pressure does not make the torque limiter to intervene, then upon actuating the third section as indicated above, the pump flow does not decrease, therefore the section will have the whole maximum flow of the pump, thus avoiding the problem described above.

[0016] Said pressure limitation on the local compensators ends as the complete actuation of the sections, such as, for example, the travel sections, at the maximum flow stops, thus allowing the system to reach the calibration maximum pressure of the valve.

[0017] According to a further functioning logic of the finding, that finds application in the second and third implementation solutions, the pressure limitation is also active in the case of an only partial requirement of flow and not only at the maximum flow requirement, with the consequence that partial negative effects could possibly occur, consisting in possible flow increases to the use compared to the desired one, that are anyway compensated by a considerable constructive simplification of the directional control valve, as described below.

[0018] Said objects and advantages are all achieved by the method for limiting the maximum power required by the hydraulic system of an earth-moving machine and by the directional control valve operating said method, which is the object of the present finding, characterized in what has been provided for in the claims reported below.

BRIEF DESCRIPTION OF THE FIGURES

[0019] This and other characteristics will result more highlighted by the following description of some embodiments illustrated, by way of non-limiting example, in the annexed drawings.

- Fig. 1: partial diagram of a mini excavator system comprising a sectional flow-sharing directional control valve, a pump, motors, a cylinder, and a relief valve, which is typical of the prior art;
- Fig. 2: section of an element of the flow-sharing directional control valve with the maximum pressure limiting system that is the object of the present finding (first implementation solution);
- Fig. 2A: an enlargement of the recess obtained on the spool 2 (first implementation solution);
- Fig. 3: hydraulic diagram of a mini excavator system comprising the flow-sharing directional control valve with the maximum pressure limiting system that is the object of the present finding (first implementation solution);
- Fig. 4: section of an element of the flow-sharing di-

rectional control valve with the maximum pressure limiting system that is the object of the present finding (second implementation solution);

- Fig. 5: hydraulic diagram of a mini excavator system comprising the flow-sharing directional control valve with the maximum pressure limiting system that is the object of the present finding (second implementation solution);
- Fig. 6: section of an element of the flow-sharing directional control valve with the maximum pressure limiting system that is the object of the present finding (third implementation solution);
- Fig. 7: hydraulic diagram of a mini excavator system comprising the flow-sharing directional control valve with the maximum pressure limiting system that is the object of the present finding (third implementation solution);
- Fig. 8: hydraulic diagram with distribution of the flows and pressures of the flow-sharing directional control valve that is typical of the prior art, with the 2 travel sections completely and concurrently actuated;
- Fig. 9: hydraulic diagram with distribution of the flows and pressures of the flow-sharing directional control valve that is typical of the prior art, upon actuating the third section with cylinder at the end of the stroke;
- Fig. 10: hydraulic diagram with distribution of the flows and pressures of the flow-sharing directional control valve that is typical of the prior art, under conditions of complete flow that is sent to the travels and delivery pressure equal to the relief valve calibration pressure plus the pump limit;
- Fig. 11: hydraulic diagram with distribution of the flows and pressures of the system with the flow-sharing directional control valve with the maximum pressure limiting system that is the object of the present finding (first implementation solution);
- Fig. 12: hydraulic diagram with distribution the flows and pressures of the system with the flow-sharing directional control valve with the maximum pressure limiting system that is the object of the present finding (second implementation solution);
- Fig. 13: hydraulic diagram with distribution the flows and pressures of the system with the flow-sharing directional control valve with the maximum pressure limiting system that is the object of the present finding (third implementation solution).

DESCRIPTION OF THE FINDING AND NUMERICAL REFERENCES

[0020] With particular reference to Figs. 1, 8, and 9, the functioning characteristics of an excavator system with the maximum power limiting system that is typical of the prior art and the problems related thereto are illustrated.

[0021] The system illustrated in Fig. 1 comprises a sectional flow-sharing directional control valve having 3 sections (A, B, and C), a relief valve D, a load sensing pump

PP, and a motor M; each of the sections A, B, or C of the system flow-sharing valve comprises a spool 2 and a compensator 1.

[0022] It is assumed that the two A and B sections of the valve control the excavator travels, that is, the actuation of the feeding means, and that the third section C controls a cylinder; the sections A and B are conventionally sized so that, when the two travels are completely and concurrently actuated, they require the maximum flow Q that can be delivered by the pump PP, and in particular each of the sections A and B requires half of the maximum flow Q , i.e., $Q/2$.

[0023] This means that, by completely actuating the two travel sections A and B, the pump plate PP is inclined at most, thus providing the maximum flow $Q_{\max}$ possible; in such situation, the flows and pressures distribution is represented in the simplified diagram of Fig. 8.

[0024] The above-reported hypothesis, in its widest meaning, also provides for the actuation of a single section, A or B, completely actuated so as to require all the maximum flow Q that can be delivered by the pump PP.

[0025] Referring again to the example reported above, it is assumed that for the actuation of the travels, which are controlled by the sections A and B, 100 bars are needed, that the pump PP limit is of 20 bars, and that the flow losses through the fully open local compensator 1 are null (because the sections A and B are those at the highest pressure); therefore, there are 100 bars downstream the spools 2, 100 bars in the LS line, and 120 bars from the pump PP to the spools 2 (delivery pressure).

[0026] When also a third section C is actuated, concurrently to the sections A and B, and that the cylinder controlled by it is at the end of its stroke, a transient is present, in which the new flows and pressures distribution is reported in the simplified diagram of Fig. 9.

[0027] Since the cylinder controlled by the section C is at the end of its stroke, the same pressure as the delivery - 120 bars - arrives to the signal LS, therefore to the pump PP, without anyway generating alterations to the state of the pump PP itself, as it is already at its maximum.

[0028] Therefore, in this situation of maximum flow, the pressure increase is not due to the signal LS increase to the pump PP, but to the fact that this signal arrives to the local compensators 1 of the sections A and B.

[0029] The local compensators 1 of the sections A and B then intervene, according to the known flow-sharing logic, throttling the flow to the travels, and in doing so, it is they that increase the delivery pressure until arriving to the end situation, in which all the flow continues to go to the travels, which therefore do not slow down, but with a delivery pressure that is equal to the relief valve D calibration pressure plus the pump PP limit, according to the flows and pressures distribution highlighted in the simplified diagram of Fig. 10.

[0030] Assumed that the relief valve is calibrated at 250 bars, therefore there are, in the sections A and B, 270 bars from the pump PP to the spools 2 (delivery

pressure), 250 bars downstream the spools 2, and 250 bars in the LS line.

[0031] This behaviour will then continue to be controlled according to the flow-sharing logic.

[0032] Since the machines are usually not equipped with first motors M (generally endothermic) capable of meeting the requirement of maximum power $P_{\max}$ that there is in the case of the requirement of the maximum flow $Q_{\max}$ at the maximum pressure $p_{\max}$, that is the relief valve D calibration one, it results that the motor M would turn off.

[0033] To obviate this problem, the prior art usually uses LS pumps provided with a torque limiter that is calibrated so that, if the required power $P = Q \times p$ exceeds the one that is generable by the motor M, it intervenes by reducing the required power, i.e., by reducing at least one of the two involved parameters; in particular, the pump torque limiter intervenes on the flow Q by automatically reducing, as a function of the pressure, the maximum inclination possible of the pump plate, therefore the maximum flow $Q_{\max}$.

[0034] The result, as previously described, is that upon actuating a third element C at high pressure, the machine translation slows down, to then accelerate again when it is released.

DISCLOSURE OF THE INVENTION

[0035] The present finding solves the above-mentioned slowdown/acceleration problem of the machine translation by intervening on the other power factor, i.e., on the pressure p ; such result is obtained by bypassing the local compensators of the section(s) that require the maximum flow (for example, of the travel sections) in the instant in which it/they is/are completely and concurrently actuated relative to a third section with cylinder at the end of its stroke; said bypassing occurs by imparting a delivery pressure lower than that imparted by the relief valve D calibration.

[0036] The present finding can be applied both to hydraulic systems in which the LS pump is provided with a torque limiter, and to systems in which the pump is not provided with it.

[0037] The above-mentioned delivery pressure is calculated so that the power required is:

Less than or equal to the power that can be delivered by the motor M, if the system pump PP is not provided with a torque limiter;

or, alternatively,
less than the power at which the torque limiter is tripped, if the system pump PP is provided with a torque limiter.

[0038] Assuming that, at the maximum flow, the torque limiter operates upon reaching 190 bars, the local compensators intervention should have to be calibrated so that they create a delivery pressure not higher than 180

bars.

[0039] In this manner, upon actuating the third section C, the pump PP flow does not decrease, therefore the travels do not slow down (i.e., the sections at the maximum flow continue to operate in such configuration), thus avoiding the previously described problem.

[0040] Three possible embodiments of the invention in order to limit the maximum pressure are described herein below.

FIRST EXEMPLARY EMBODIMENT

[0041] With particular reference to Figs. 2, 3, and 11 the first constructive solution of the maximum pressure limiting system that is the object of the present finding is described.

[0042] A first method for limiting the maximum pressure in the case of completely actuated travels A and B together with a third section C is to make so that the spools 2 of the sections A and B, at the end of their stroke, open a passage 3 between the two areas upstream 4 and downstream 5 the compensator 1 thereof, so as to bypass the same.

[0043] Said passage 3 is a recess that is obtained on the spool 2 of section A, B, as illustrated in detail in Fig. 2A.

[0044] Said passage 3 has to be such that, at the delivery pressure of 180 bars calculated before, all the maximum flow Q ($Q/2$ per travel) passes through it, and not through the compensator 1, that is practically shut out. By doing so, when the spools 2 of the travels A and B are completely actuated, and a third section C is actuated, the maximum flow Q (which, in the example, is divided in $Q/2$ per part) continues to go entirely to the travels A and B, the delivery pressure does not reach the calibration pressure value of the relief valve D, but only the set pressure value (the above-mentioned 180 bars).

[0045] At this pressure value, the torque limiter is tripped, therefore the travels do not slow down, thus obviating the problem reported above.

[0046] As illustrated in the simplified diagram of Fig. 11, the flows and pressures distribution is as follows: 180 bars from the pump PP to the spools 2 (delivery pressure), 160 bars downstream the spools 2, and 180 bars in the LS line.

SECOND EXEMPLARY EMBODIMENT

[0047] With particular reference to Figs. 4, 5, and 12, the second constructive solution of the maximum pressure limiting system that is the object of the present finding is illustrated.

[0048] A second method for limiting the maximum pressure consists in opening a passage 2E, practically, a hole, between the two areas upstream 4 and downstream 5 the compensator 1 directly in the section A and B.

[0049] Said passage 2E is calculated so that, at the

delivery pressure of 180 bars calculated before, all the flow Q ($Q/2$ per travel) passes through it, thus bypassing the compensator 1.

[0050] As illustrated in the simplified diagram of Fig. 12, the flows and pressures distribution is as follows: 180 bars from the pump PP to the spools 2 (delivery pressure), 160 bars downstream the spools 2, and 180 bars in the LS line.

[0051] With the through hole 2E obtained directly in the sections A and B, the pressure limiting is, however, active also in the case of only partial requirement of the flow, and not only at the maximum requirement, with the consequence that, when the travels sections A, B are partially actuated, flow increases to the use compared to that desired can occur, which translate in acceleration phenomena.

[0052] However, on the other hand, the solution cost is lower, it being constructively easier.

THIRD EXEMPLARY EMBODIMENT

[0053] With particular reference to Figs. 6, 7, and 13, the third constructive solution of the maximum pressure limiting system that is the object of the present finding is illustrated.

[0054] A third method for limiting the maximum pressure consists in using a compensator 1A that is designed so that, at the end of the stroke, it leaves a passage 2G open, through which all the flow Q passes ($Q/2$ per travel) at the previously calculated pressure of 180 bars.

[0055] Said passage 2G is obtained by limiting the compensator stroke so as not to let it completely close, or through a recess obtained on the same compensator.

[0056] Also in this case, as in the second constructive solution, the limitation of the maximum pressure is active also in the case of only partial requirement of flow, and not only at the maximum requirement, therefore in intermediate positions of the spool 2 of the travels A and B, acceleration phenomena can occur; however, such disadvantages are compensated by the simplicity of the solution.

[0057] This functionality is always valid when the power required P, given by the maximum flow absorbed by the travels for the maximum pressure delivered by a third use C with the cylinder at the end of its stroke, exceeds the maximum power that can be delivered by the motor M, independently from the presence or not of the torque limiter.

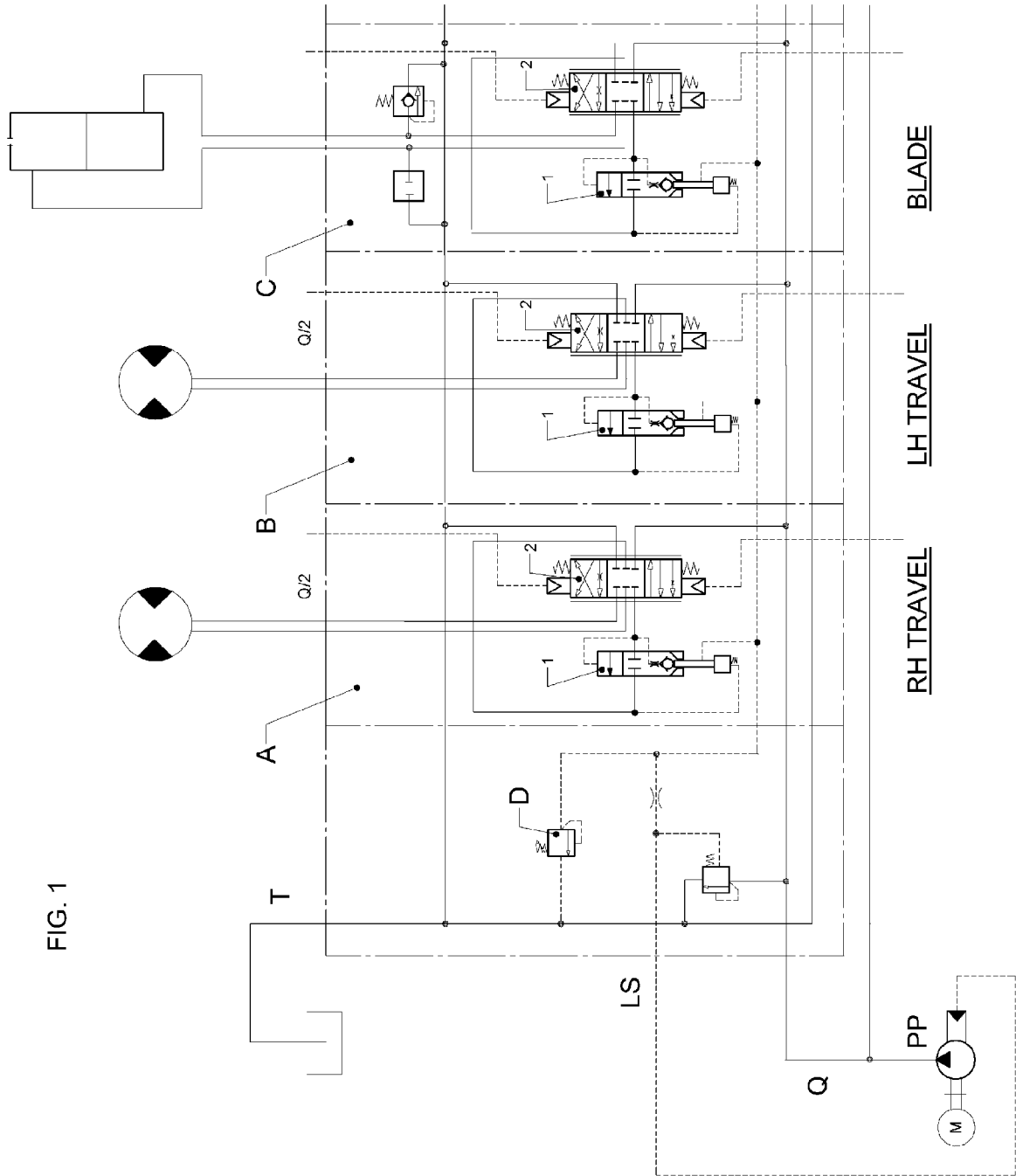
[0058] The optional absence of the torque limiter does not rise the motor turning off problem, since this is avoided by limiting the maximum pressure as described before.

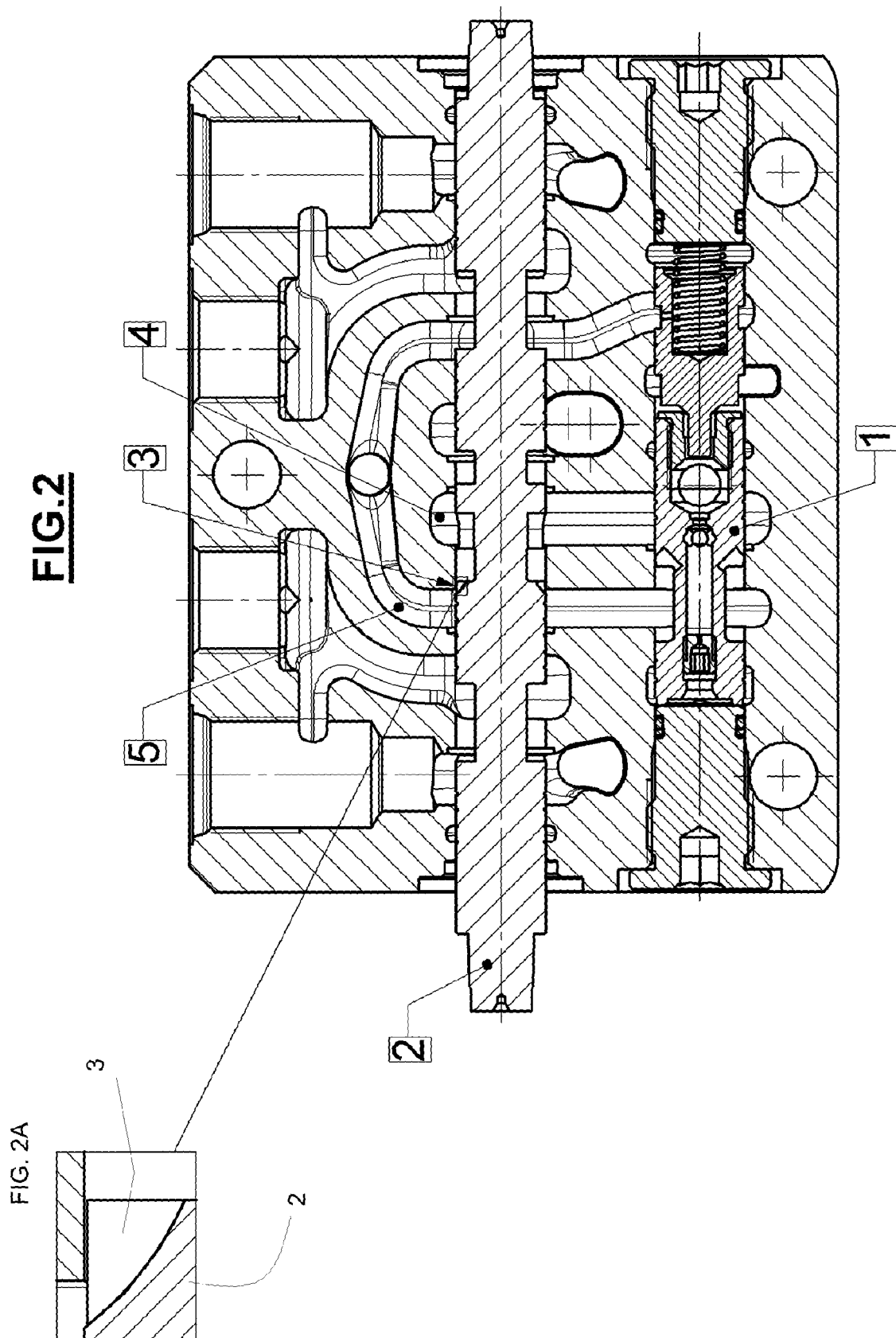
[0059] The same applies also if the pump PP is not LS, and a compensator is then comprised on the flow-sharing directional control valve side.

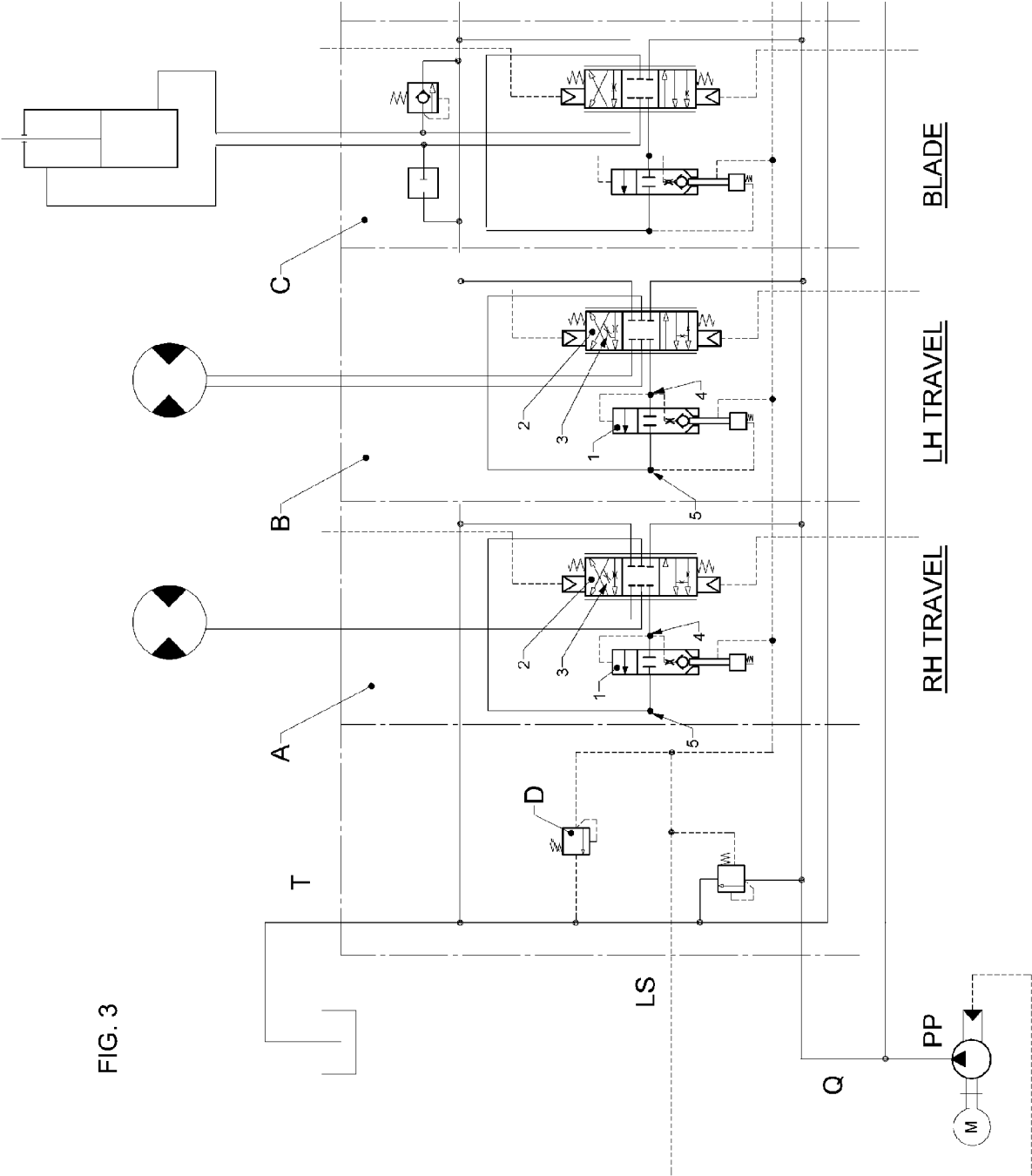
Claims

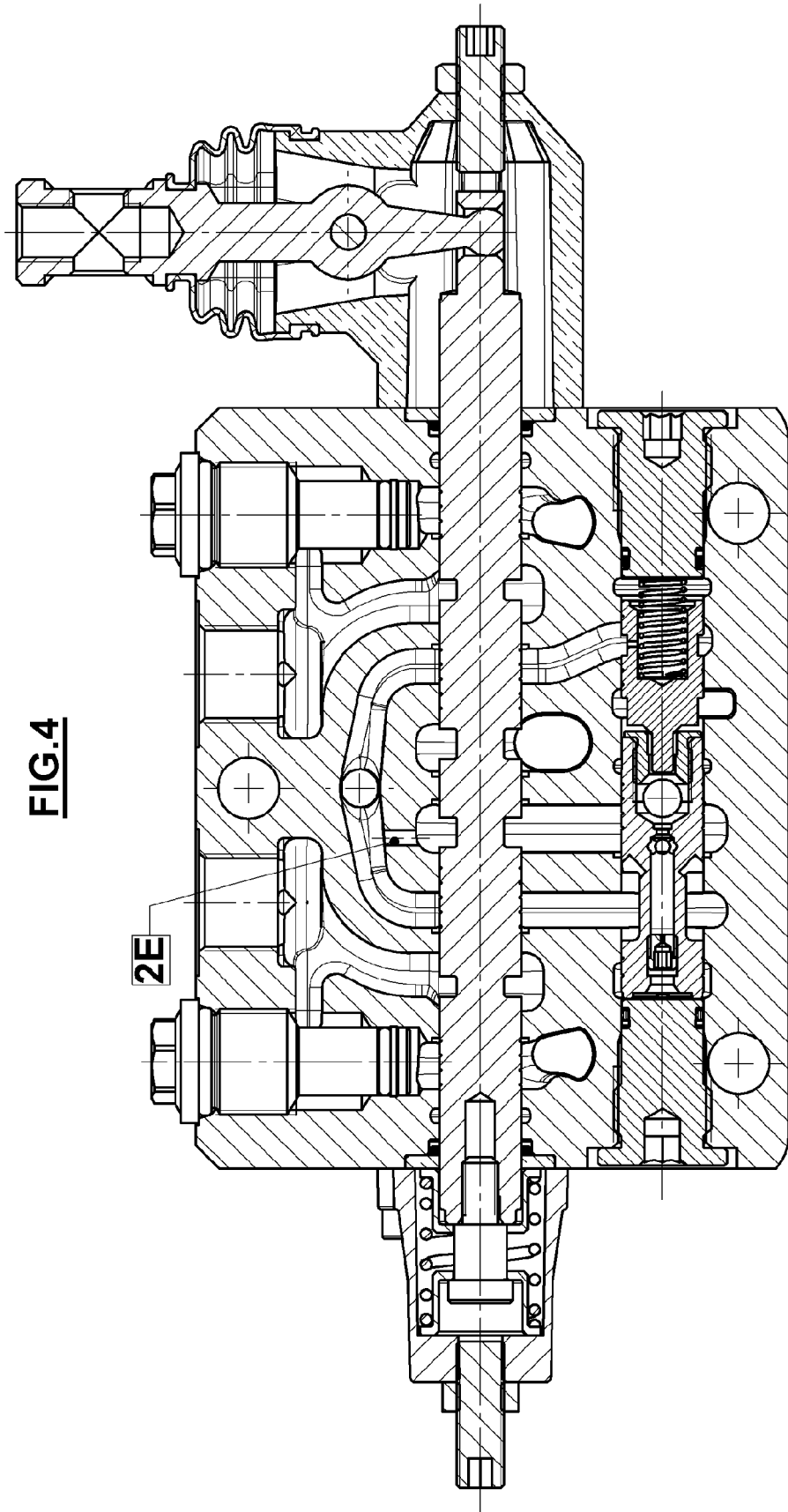
1. A method for limiting the maximum power required by the hydraulic system of an earth-moving machine, such hydraulic system being composed of a sectional flow-sharing directional control valve having multiple sections (A, B, C), each section comprising a local compensator (1) and a spool (2), with one section (A) or (B) or both the sections (A) and (B) requiring the maximum flow and at least one additional section (C) designed for actuating a work function of the machine, a pump (PP) with or without torque limiter, a motor (M) and a relief valve (D), **characterized in that** it includes the step of bypassing the local compensators (1) of the section and/or sections (A, B) requiring the maximum flow when they are fully actuated at the same time as the third section (C), to impart a lower delivery pressure than that of the relief valve (D), and calculated so that the power required (the product of flow and pressure) is:
 - less than or equal to the power that can be delivered by the motor (M), if the pump (PP) has no torque limiter or
 - less than the power at which the torque limiter is tripped, if the pump (PP) has one whereby the flow is not reduced.
2. The method as claimed in claim 1, **characterized in that** it includes the step of bypassing the local compensator (1) of the section and/or sections (A, B) requiring the maximum flow, by opening a passage (3) between the two areas upstream (4) and downstream (5) from the compensator (1); said passage (3) being directly opened by the spool/s (2) of the section/s (A, B) at the end of their stroke; said passage (3) being calculated so that, with the delivery pressure appropriately calculated as claimed in claim 1, all the maximum flow passes through it and not through the compensator (1).
3. The method as claimed in claim 1, **characterized in that** it includes the step of bypassing the local compensator (1) of the section and/or sections (A, B) requiring the maximum flow, by opening a passage (2E) between the two areas upstream (4) and downstream (5) from the local compensator (1), directly machined into the sections (A, B) requiring the maximum flow; said passage (2E) being calculated so that, with the delivery pressure appropriately calculated as claimed in claim 1, all the maximum flow passes through it and bypasses the compensator (1).
4. The method as claimed in claim 1, **characterized in that** it includes the step of bypassing the local compensator (1) of the section and/or sections (A, B) requiring the maximum flow, by opening a passage (2G) that is left open by the local compensator (1) itself at the end of its stroke, said passage (2G) being calculated so that, with the delivery pressure appropriately calculated as claimed in claim 1, all the flow passes through it and bypasses the compensator (1).
5. A sectional flow-sharing directional control valve having multiple sections (A, B, C), each section comprising a local compensator (1) and a spool (2), a pump (PP) with or without a torque limiter, a motor (M) and a relief valve (D), **characterized in that** the section/s (A, B) designed to operate at the maximum flow open a passage (3, 2E, 2G) to bypass the local compensator (1) if a third section (C) is actuated; said passage (3, 2E, 2G) being calculated so that, with the delivery pressure appropriately calculated as claimed in claim 1, all the flow passes through such passage (3, 2E, 2G).
6. The directional control valve as claimed in claim 5, **characterized in that** said passage (3) is directly opened by a recess formed on the spool (2) at the end of its stroke, between the two areas upstream (4) and downstream (5) from the local compensator (1).
7. The directional control valve as claimed in claim 5, **characterized in that** said passage (2E) is a hole formed between the two areas upstream (4) and downstream (5) from the local compensator (1), directly machined in the section/s (A, B) requiring the maximum flow.
8. The directional control valve as claimed in claim 5, **characterized in that** said passage (2G) is opened directly by the local compensator (1) itself at the end of its stroke.

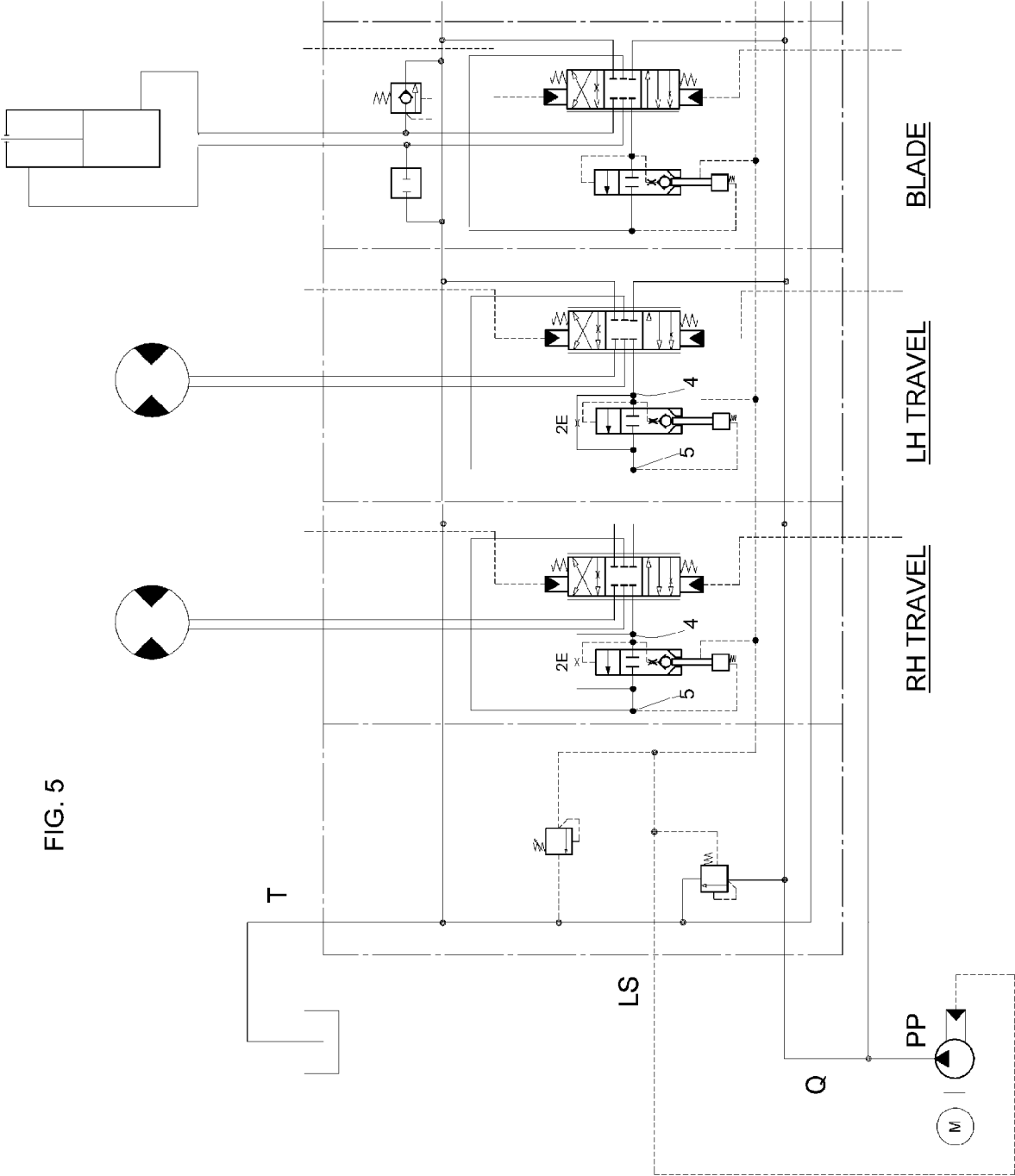
FIG. 1

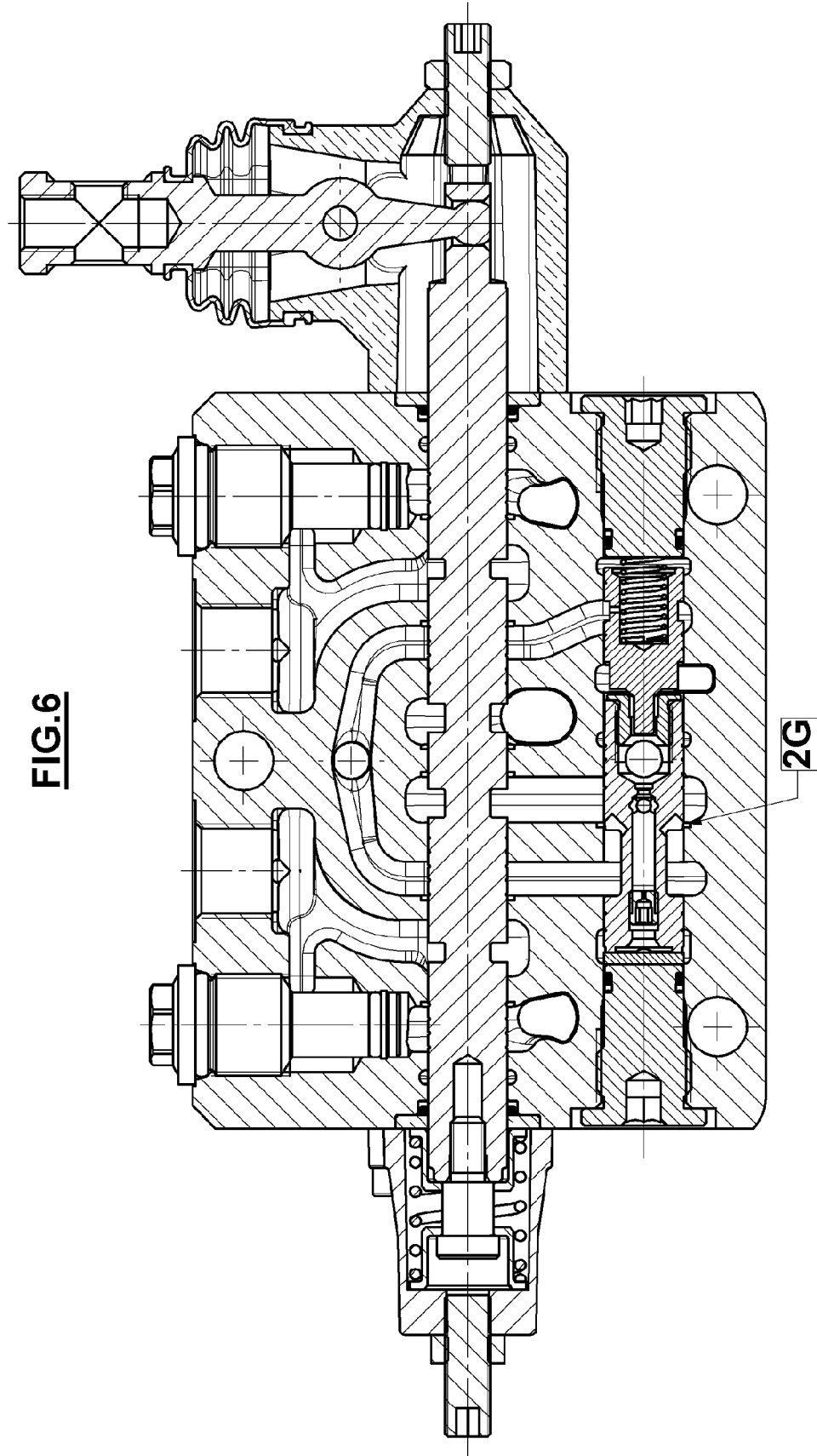












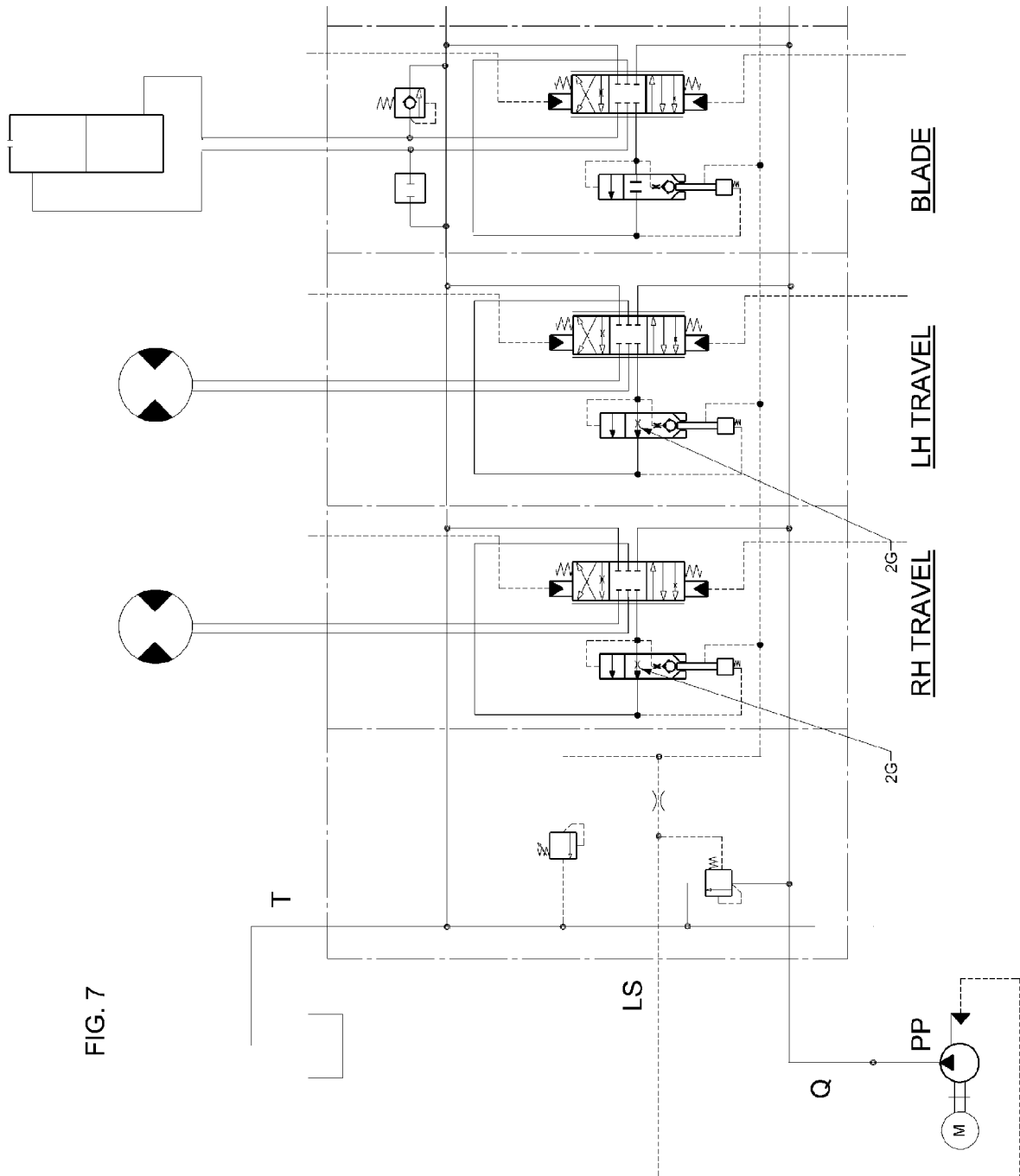
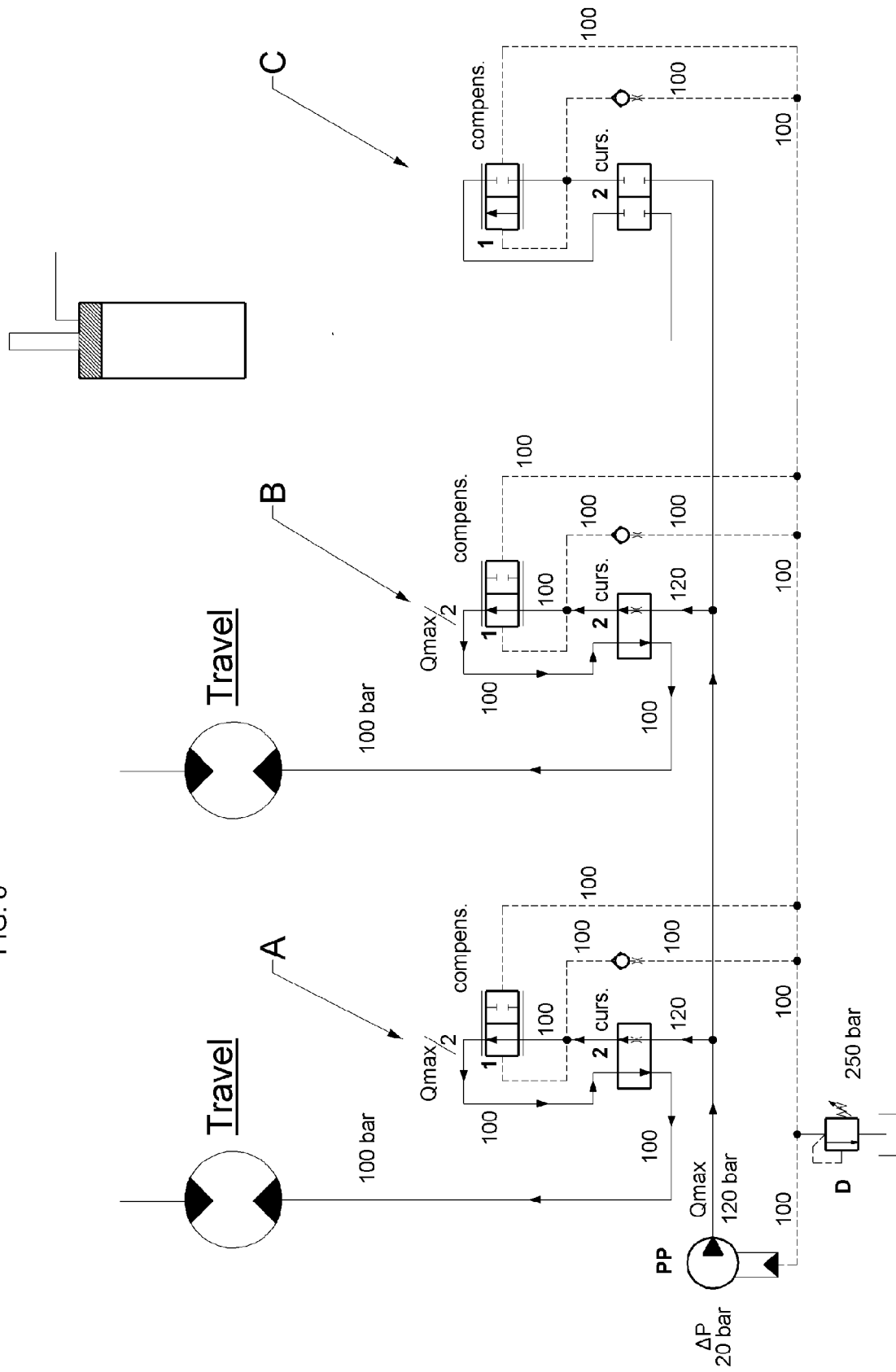
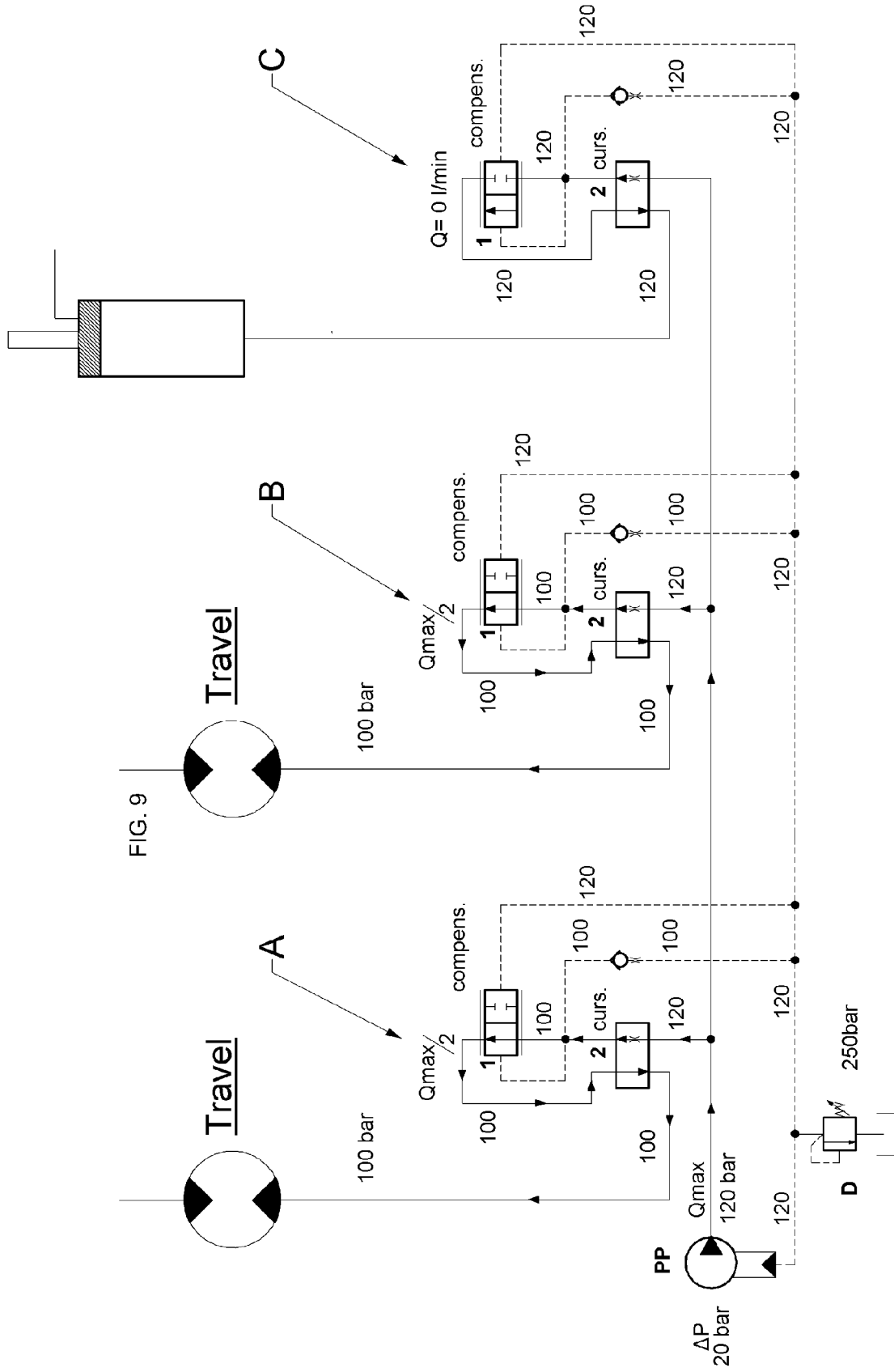


FIG. 7

FIG. 8





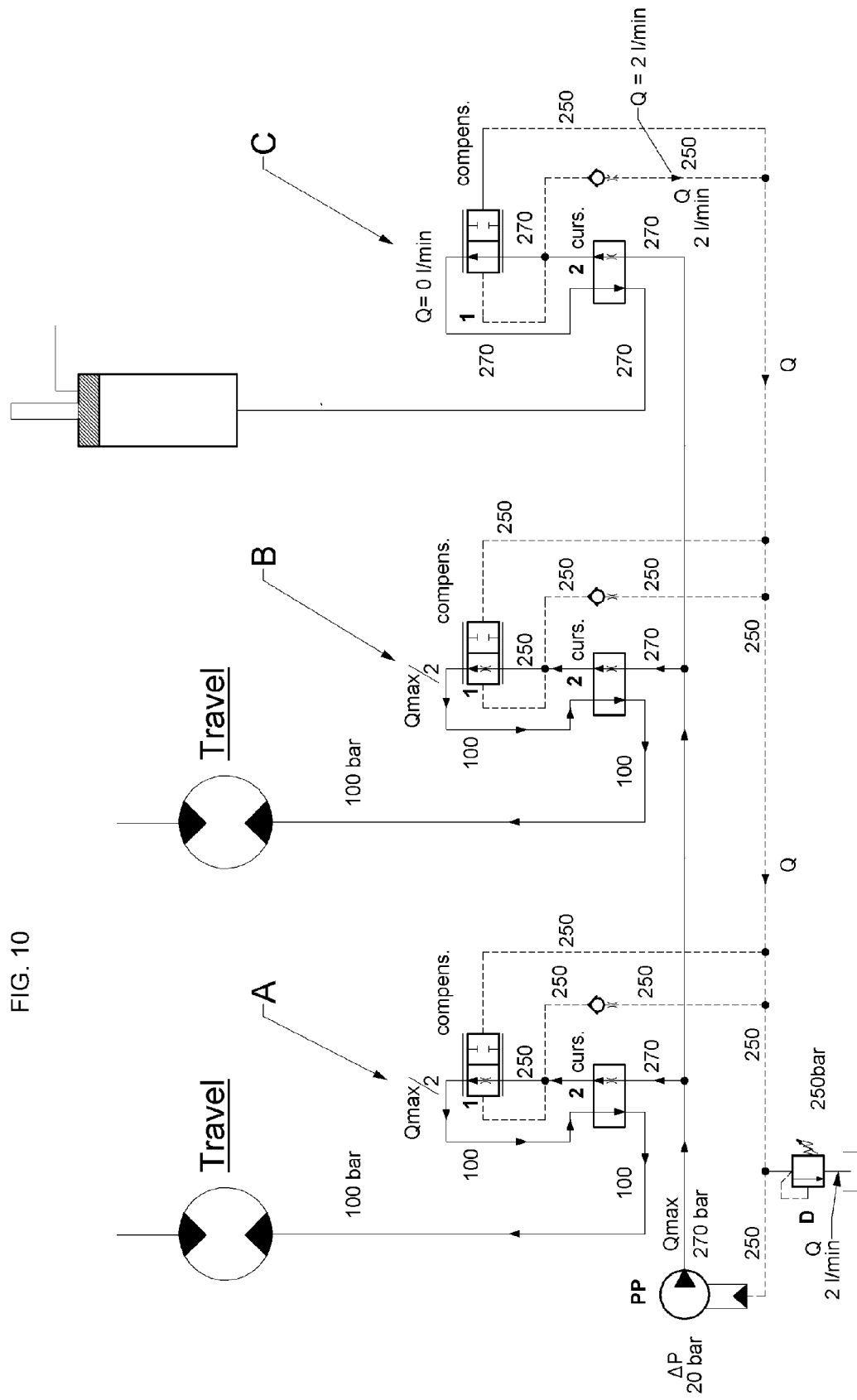


FIG. 11

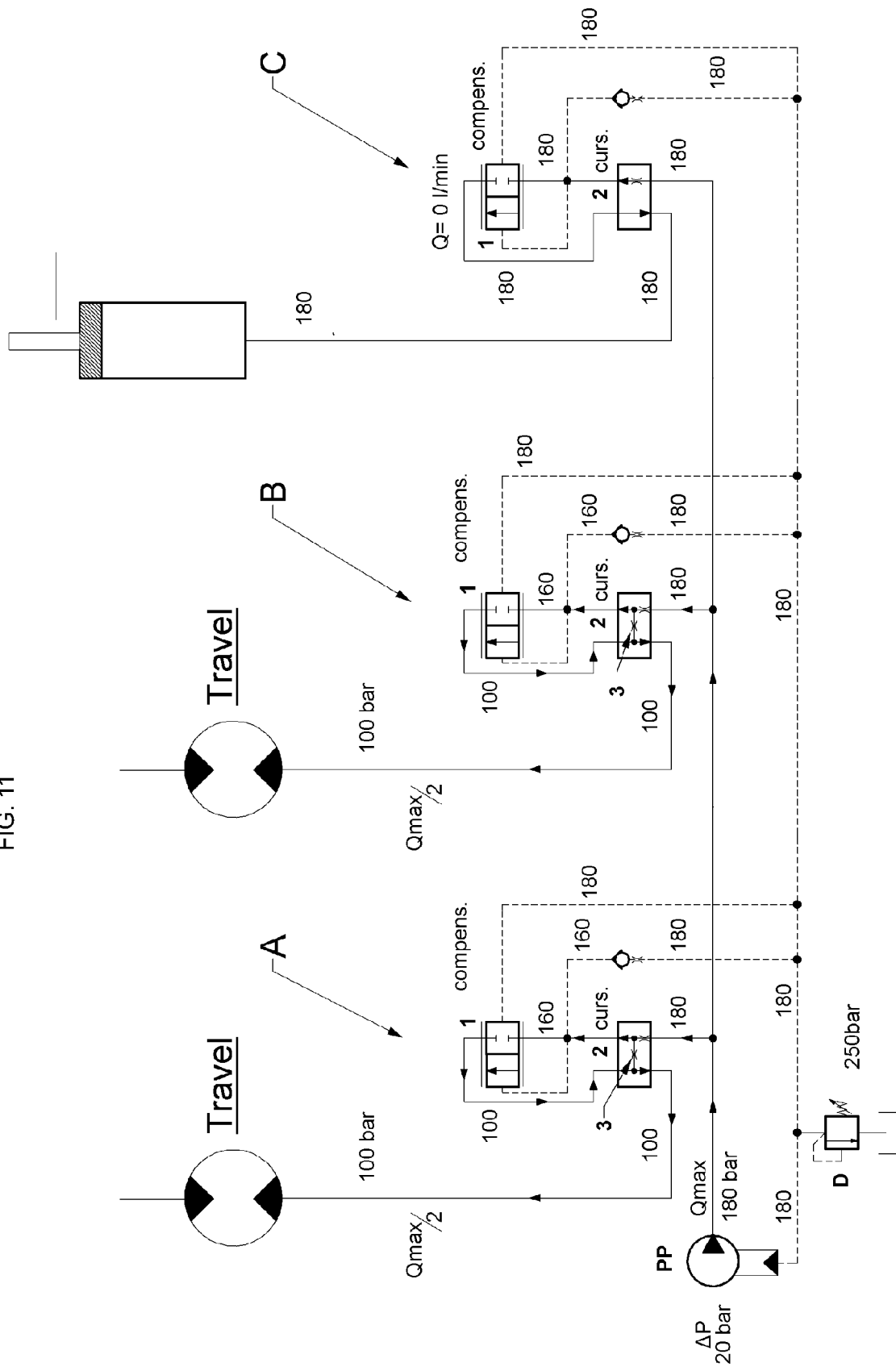


FIG. 12

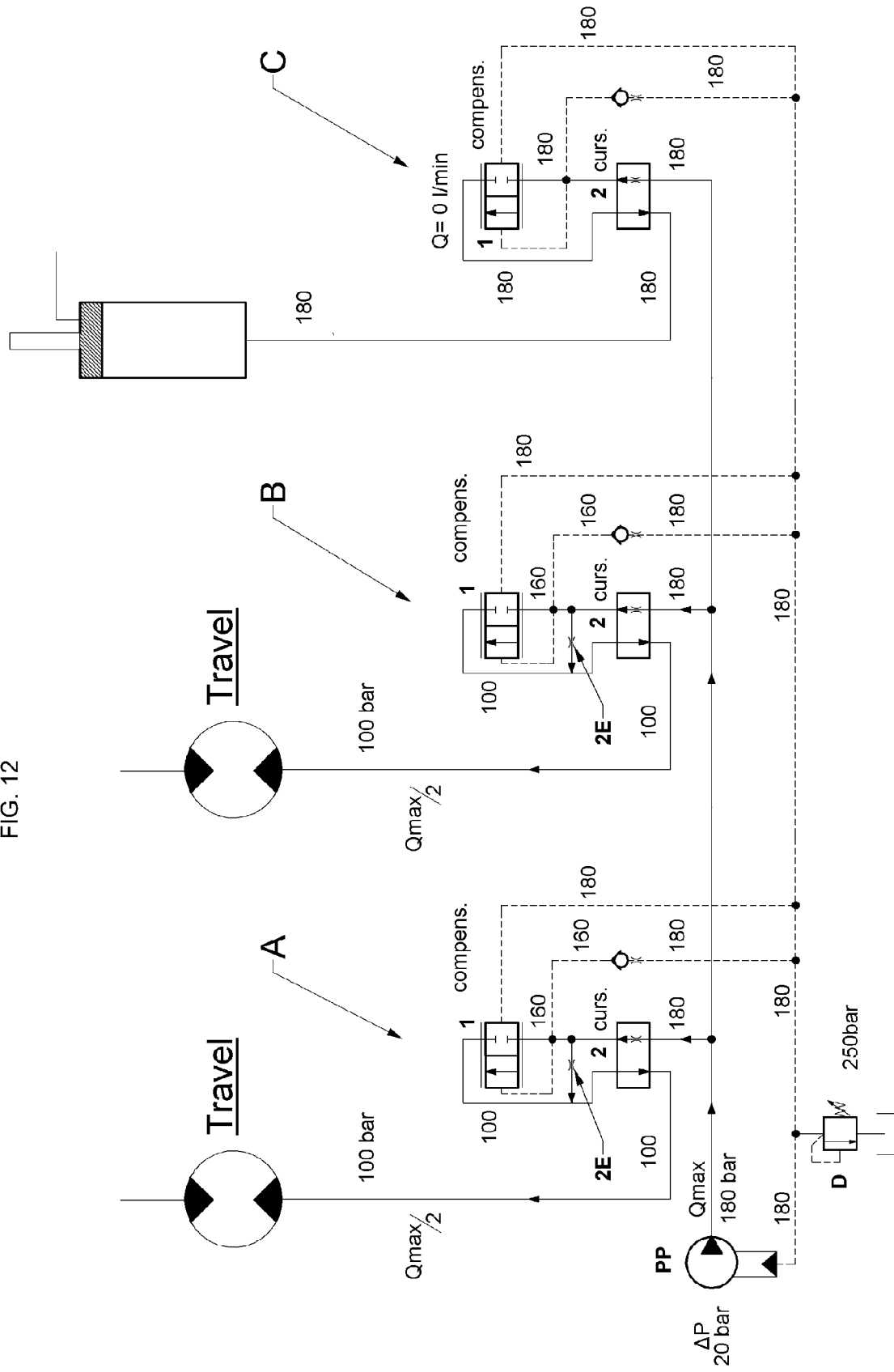
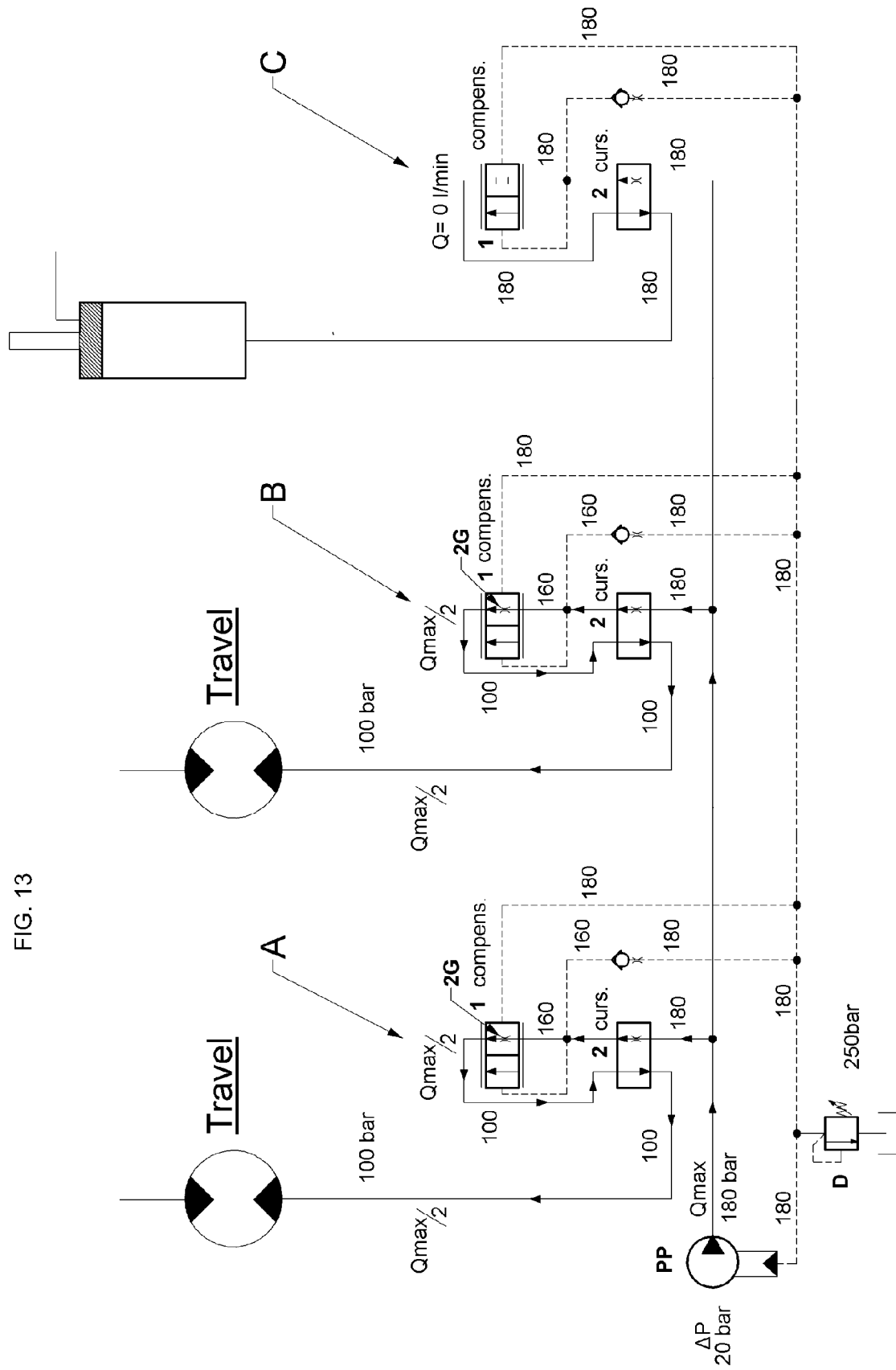


FIG. 13





EUROPEAN SEARCH REPORT

Application Number
EP 09 17 4983

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Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	CLASSIFICATION OF THE APPLICATION (IPC)
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The present search report has been drawn up for all claims			
Place of search The Hague		Date of completion of the search 29 December 2009	Examiner Krikorian, Olivier
CATEGORY OF CITED DOCUMENTS X : particularly relevant if taken alone Y : particularly relevant if combined with another document of the same category A : technological background O : non-written disclosure P : intermediate document T : theory or principle underlying the invention E : earlier patent document, but published on, or after the filing date D : document cited in the application L : document cited for other reasons & : member of the same patent family, corresponding document			

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EPO FORM 1503 03.82 (P04C01)

**ANNEX TO THE EUROPEAN SEARCH REPORT
ON EUROPEAN PATENT APPLICATION NO.**

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29-12-2009

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