

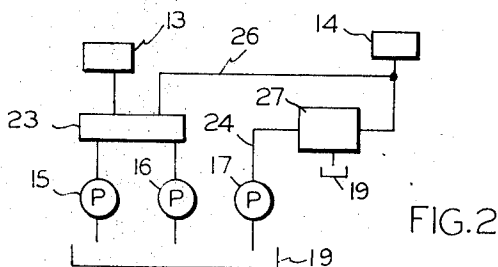
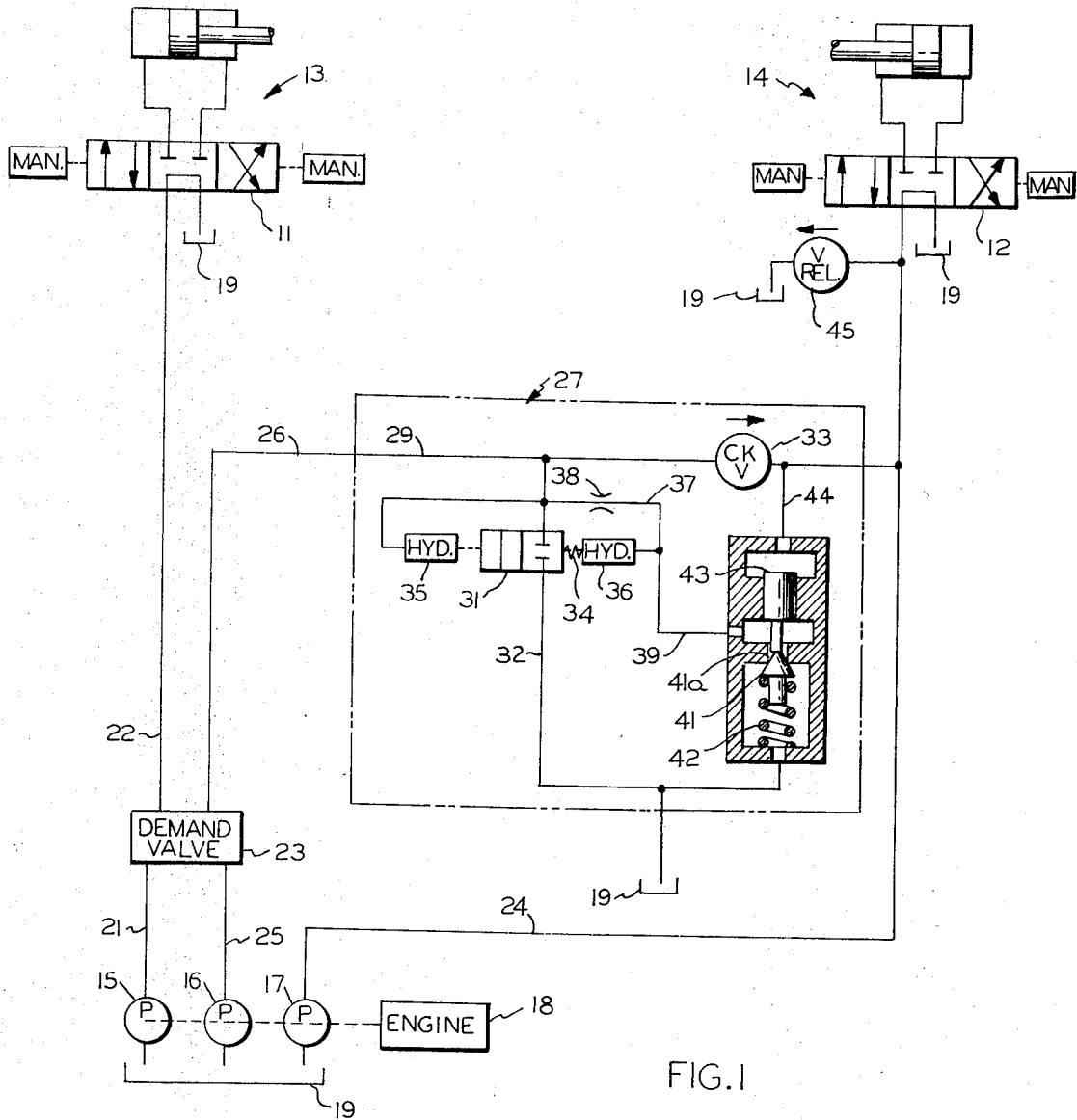
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THREE-PUMP HYDRAULIC SYSTEM INCORPORATING AN UNLOADER

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THREE-PUMP HYDRAULIC SYSTEM INCORPORATING AN UNLOADER
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ABSTRACT OF THE DISCLOSURE
An integrated hydraulic system for a vehicle including steering and implement circuits and three supply pumps. Two of the pumps are connected, respectively, with the two circuits, and a demand valve splits the output of the third pump between the circuits as needed to maintain a substantially constant rate of flow to the steering circuit. An unloader diverts to tank either the output of the third pump or the implement output of the demand valve when implement pressure exceeds a predetermined value, and delivers said output to the implement circuit when that pressure reduces to a predetermined lower value.

BACKGROUND AND SUMMARY OF THE INVENTION
U.S. Pat. 3,355,994, granted Dec. 5, 1967, discloses an integrated hydraulic system for construction vehicles, such as articulated loaders, which includes a pair of pumps for supplying oil exclusively to the steering and implement circuits, respectively, and a flow dividing or demand valve for splitting the output of a third pump between the two circuits as needed to maintain a substantially constant rate of flow to the steering circuit. Although this system affords real advantages, its use on some articulated loaders has been accompanied by excessive heating of the hydraulic fluid. Some people have attributed this difficulty to the steering circuit evidently because the flow rates usually are higher than in the corresponding circuits for non-articulated vehicles, and also because metering of these flows commonly is accompanied by large pressure drops. However, our study of the problem compels a different conclusion and provides a sound basis for obtaining relief.

Commonly, the work cycle of an articulated loader is of twenty seconds duration and is divided into the following four phases: (1) the time spent working in a pile of material filling the bucket, (2) the time required to move from the pile to a truck, (3) the time spent at the truck dumping the material, and (4) the time consumed returning to the pile. Our investigation indicates that, on a typical loader whose engine has low and high idle speeds of 600 and 2200 r.p.m., respectively, the operating conditions of the hydraulic system during a representative work cycle are as follows:

Work cycle phase	Eng. speed, r.p.m.	Time, sec.	Implement		Steering	
			Pressure, p.s.i.	Supply, g.p.m.	Pressure, p.s.i.	Supply, g.p.m.
In pile	1,700	5	2,000	72	100	26
	2,200	2	1,000	94	100	33
	2,000	2	100	85	2,000	30
Pile-to-truck	2,100	2	100	90	100	32
	700	1	500	16	500	25
	1,200	1	500	44	500	26
At truck	2,200	1	500	94	500	33
	2,200	2	100	94	100	33
	2,100	2	100	90	500	32
Truck-to-pile	2,100	2	100	90	100	32

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This table shows that the steering circuit consumes only about 20% of the total hydraulic energy expended during the work cycle, and thus demonstrates clearly that the steering circuit is not the cause of the heating problem. Just as importantly, the table indicates that the major portion of the implement energy is spent while the loader is at the pile and the bucket is being filled. Since the bucket manipulations required for the filling operation require maximum pressure, but only very small rates of flow, it follows that most of the large volume of oil delivered to the implement circuit in this phase of the cycle is diverted to tank across the circuit's relief valve. This, then, is the real cause of the overheating problem.
15 The object of this invention is to provide an improved version of the three-pump system of Pat. 3,355,994 which reduces the amount of energy converted into heat during the "In Pile" phase of the work cycle. According to the invention, the system is provided with an unloader which, in response to attainment of a predetermined load pressure in the implement circuit, diverts to tank at a low pressure drop either the output of the implement pump or the implement output of the demand valve. The choice between these alternatives depends upon which output is larger. The pressure setting of the unloader is below the setting of the implement relief valve and above the pressure required to lift a loaded bucket. Thus, during the "In Pile" phase, the volume of oil diverted to tank across the relief valve will be reduced considerably, and energy will be conserved. On the other hand, during the "Pile-To-Truck" phase, in which the loaded bucket is raised, the implement circuit will be supplied at the usual maximum rate so that the bucket-raising operation is not prolonged.
25 **BRIEF DESCRIPTION OF THE DRAWINGS**
The preferred embodiment of the invention is described herein with reference to the accompanying drawing in which:
30 **FIG. 1** is a schematic diagram of an improved system in which the unloader is interposed in the implement output conduit of the demand valve.
FIG. 2 is a simplified schematic diagram of an improved system in which the unloader is interposed in the output conduit of the implement pump.
35 **DESCRIPTION OF PREFERRED EMBODIMENT**
The improved system shown in FIG. 1 is arranged to supply fluid to the open center directional control valves 11 and 12 of the steering and implement circuits 13 and 14, respectively. Although the illustrated implement circuit 14 has only a single valving unit 12, it will be understood that the actual circuit includes a valving unit for each implement function. The system includes three fixed displacement supply pumps 15-17, normally of the gear type, which are driven by the vehicle's propulsion
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engine 18 and which draw fluid from the common reservoir tank 19 to which the valves 11 and 12 exhaust. The output of pump 15 is delivered continuously to control valve 11 through conduits 21 and 22 and the demand valve 23, and the output of pump 17 is delivered continuously to implement valve 12 through conduit 24. Pump 16, on the other hand, is connected with demand valve 23 through conduit 25, and, depending upon the speed of engine 18, its output is delivered to either or both of the valves 11 and 12 through conduit 22 or conduit 26. The demand valve 23 can take any one of various known forms, but, in any case, it serves to maintain a substantially constant total rate of flow to valve 11 throughout a major portion of the speed range of engine 18. For example, if the low and high idle speeds of engine 18 are 600 and 2200 r.p.m., respectively, valve 23 may be designed to maintain a constant steering supply rate at speeds between 700 and 1700 r.p.m. At higher speeds, switch pump 16 is isolated from the steering circuit by valve 23, and the total output of the pump is delivered to conduit 26.

In the FIG. 1 system, switch pump 16 is larger than implement pump 17, so the unloader 27 provided by the invention is interposed in the conduit 26 leading from demand valve 23 to implement circuit 14. Although various types of unloaders can be used, the device 27 chosen for illustration comprises a through passage 29 leading from conduit 26 to implement circuit 14, an unloading valve 31 which is adapted to open and close an exhaust passage 32 interconnecting passage 29 and tank 19, and a check valve 33 which is interposed in passage 29 and is oriented to prevent escape of oil from circuit 14 at times when valve 31 is in unloading position. Unloading valve 31 is biased to the illustrated loading position by a spring 34 and is actuated by a pair of opposed pressure motors 35 and 36. Motor 35, which urges valve 31 to unloading position, is connected with passage 29 and responds continuously to the pressure upstream of check valve 33, whereas motor 36 is subjected to pressures developed by a piloting circuit which responds to the load pressure in implement circuit 14. The piloting circuit includes a pilot passage 37 which leads to motor 36 from passage 29 and contains a flow restrictor 38, and a second pilot passage 39 which leads from motor 36 to tank 19 and is controlled by pilot valve 41. The pilot valve is biased closed by a spring 42 and is opened by the pressure of the fluid which is transmitted to it through pilot passage 39 and which acts upon its nose 41a. Since this pressure is essentially the same as the load pressure in implement circuit 14, it will be evident that pilot valve 41 opens when implement pressure reaches a predetermined level. And, since the pilot passage 37 is restricted, it also should be evident that opening of valve 41 will unbalance the pressures to which motors 35 and 36 are subjected and enable motor 35 to shift valve 31 to unloading position. After valve 41 has been opened, it is held open by a piston 43 which is subjected to the load pressure in implement circuit 14 through a passage 44. The cross sectional area of piston 43 is larger than the effective area of valve nose 41a, so the pressure at which valve 41 closes will be materially lower than the pressure at which the valve opens. This differential is necessary in order to make unloader 27 stable.

In order for unloader 27 to be effective as an energy conservator, valve 31 must shift to unloading position at a load pressure below the setting of the implement relief valve 45. On the other hand, since rapid raising of the loader bucket requires the outputs of both of the pumps 16 and 17, valve 31 must shift to loading position at a pressure above that needed to lift the bucket. In a typical case wherein relief valve 45 is set for 2,000 p.s.i. and the maximum pressure required for lifting operations is 1,000 p.s.i., spring 42 and the differential between the effective areas of piston 43 and valve nose 41a are so selected that pilot valve 41 opens at a pressure of 1,800 p.s.i. and closes at a pressure of 1,500 p.s.i.

When the FIG. 1 system is in service and the loader commences working in a pile of material, the load pressure in implement circuit 14 will rise above the 1,800 p.s.i. setting of unloader 27, and pilot valve 41 will open. As mentioned earlier, this action reduces the pressure at motor 36 relatively to the pressure at motor 35 and enables the latter to shift valve 31 to unloading position. Therefore, the implement output of demand valve 23 will be diverted to tank 19 through passage 32 at a relatively low pressure drop. Since, during this bucket-filling operation, the speed of engine 18 usually is relatively high, most if not all of the output of switch pump 16 will follow this unloading path. This reduces drastically the volume of oil which escapes to tank 19 through relief valve 45, and consequently conserves energy.

When the bucket-filling operation has been completed, the load pressure in implement circuit 14 will decrease below 1,500 p.s.i., and spring 42 will overpower piston 43 and close pilot valve 41. Closure of pilot passage 39 effects equalization of the pressures at motors 35 and 36, so spring 34 now shifts valve 31 back to the illustrated loading position. As a result, the implement output of demand valve 23 now passes to implement circuit 14 via conduit 26, passage 29 and check valve 33. Therefore, as the loader withdraws from the pile, the operator may raise the bucket at a rapid speed utilizing the full outputs of both of the pumps 16 and 17. And, in the usual case, unloader 27 will remain in this loading condition throughout the remainder of the work cycle.

Although, in most installations switch pump 16 is larger than implement pump 17, sometimes the relative capacities of the pumps are reversed. In these cases, the implement output conduit 26 of demand valve 23 is connected directly to the implement circuit, and unloader 27 is interposed in the output conduit 24 of implement pump 17. This arrangement is illustrated in FIG. 2.

The unloader 27 can be mounted at various places on the vehicle, but the preferred location is at the implement valve 12. This arrangement permits the two valves to share a common tank line, and allows the unloader to serve either of the pumps 16 and 17 merely by interchanging conduit connections.

We claim:

1. In an integrated hydraulic system of the type including first and second work circuits (13, 14) connected to receive oil from first and second pumps (15, 17), respectively, and a demand valve (23) connected with the two circuits and adapted to split the output of a third pump (16) between them as required to maintain a substantially constant total rate of flow to the first circuit (13), the improvement which comprises an unloader (27) interposed either in the connection (26) between the demand valve (23) and the second work circuit (14) or in the connection (24) between the third pump (17) and the second work circuit, the unloader being responsive to the load pressure in the second work circuit (14) and serving to divert to a reservoir (19) the oil delivered to the associated connection (24 or 26) when the pressure rises above a first value and to deliver said oil to the second work circuit when the pressure reduces below a second, lower value.

2. The improvement defined in claim 1 in which:

- (a) the third pump (17) is in continuous communication with the second work circuit (14); and
- (b) the unloader (27) is interposed in the connection (26) between the demand valve (23) and the second work circuit.

3. The improvement defined in claim 1 in which:

- (a) the connection (26) between the demand valve (23) and the second circuit (14) is continuously open; and
- (b) the unloader (23) is interposed in the connection (24) between the third pump (17) and the second work circuit (14).

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4. The improvement defined in claim 1 in which the unloader (27) comprises:

- (a) an unloading valve (31) for opening and closing an exhaust path (32) leading from the associated connection (24 or 26) to the reservoir (19);
- (b) means including a first pressure motor (36) urging the unloading valve closed;
- (c) a second pressure motor (35) subject to the pressure in the associated connection (24 or 26) and urging the unloading valve open; and
- (d) pilot valve means (37-39, 41) for subjecting the first pressure motor (36) to the pressure in the associated connection (26) when the pressure in the second circuit (14) decreases below said second value, and for subjecting the first pressure motor (36) to a lower pressure when the pressure in the second circuit increases above said first value.

5. The improvement defined in claim 4 in which the unloader (27) also includes a check valve (33) interposed in the associated connection (24 or 26) between the junction with said exhaust path (32) and the second circuit (14) and oriented to prevent flow from the second circuit.

6. The improvement defined in claim 4 in which the pilot valve means comprises:

- (a) a pilot passage including a first portion (37) leading from the associated connection (24 or 26) to the first pressure motor (36) through a flow restriction (38), and a second portion (39) leading from said pressure motor (36) to the reservoir (19) and being normally closed by a pilot valve (41);

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- (b) means (41a) responsive to the pressure in said second portion (39) for opening the pilot valve (41) when the pressure rises above said first value; and
- (c) means (43) responsive to the pressure in the second circuit (14) for holding the pilot valve (41) open until the pressure reduces to said second value.

7. The improvement defined in claim 6 in which:

- (a) the unloader (27) includes a check valve (33) interposed in the associated connection (24 or 26) between the junction with said exhaust path (32) and the second circuit (14) and oriented to prevent flow from the second circuit; and
- (b) the pilot passage (37, 39) joins the associated connection at the upstream side of the check valve (33).

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