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Analysis of hydraulic drive circuit selection for a carrot collecting system

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Abstract:

Compared to conventional mechanical transmissions, hydraulic transmission has the disadvantage of lower energy efficiency. However, it is widely employed in various working machines such as construction machines, off-road vehicles, forest logging machines, and especially agricultural machinery, owing to its flexibility in design, geometric arrangement, and simplicity in operation. In the development project for carrot harvesting machinery conjugate, hydraulic transmission is the optimal solution for the design and implementation of the carrot collecting roller system. This project is oriented towards achieving a practical result where the system structure is simple and effective, utilising hydraulic components available on the market in Vietnam. Given the myriad of hydraulic transmission circuits available, this paper undertakes an analysis based on numerical simulations of the functionality and energy efficiency of potential system design options and proposes a new practical speed control circuit. This suggested circuit uses a pressure-compensated flow control valve arranged parallel to the hydraulic motor. The study is carried out in the Matlab/Simulink simulation environment, where the circuit models are constructed using hydraulic components with practical parameter settings sourced from component suppliers. The analysis results indicate that the proposed circuit strikes a balance between functionality and economic efficiency.

Keywords: carrot collecting machine, hydraulic transmission, speed regulating circuit, 3-way flow control valve.

Classification numbers: 2.1, 2.3, 3.1

1. Introduction

The soil separating and collecting machine is the concluding unit of the carrot harvesting machine conjugate, which is part of the research products of the “Research on the procedure and synchronisation of mechanised equipment in producing carrot in Vietnam” project.

The basic structure of the soil separating and collecting machine is a roller-type separating system that removes soil, grass, and trash from carrot roots and then collects the carrot roots into containers. The entire arrangement of the machine conjugate is illustrated in Fig. 1.

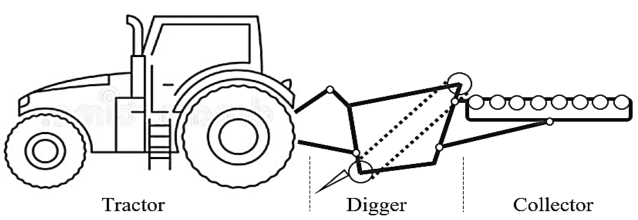

Fig. 1. Arrangement of the carrot harvesting machine conjugate.

Given the chosen configuration of the machine conjugate, the design of the transmission for the collecting machine over a long distance needs to be considered. Moreover, the collecting machine needs to be capable of lifting during the operational process; thus, flexibility in the transmission line is essential.

Fig. 2. Roller system of the carrot collecting machine.

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The structure of the roller system of the carrot collecting machine is depicted in Fig. 2. To maintain the functionality of the system, the revolution speed of the rollers must be adjustable based on the harvesting conditions such as soil moisture, soil hardness, and supply volume from the digger. This ensures efficient carrot separation from soil, grass, and trash. Additionally, the revolution speed must remain stable at a chosen value, essentially constant, regardless of the random variation in the actual load impacting the roller system due to the working conditions.

Given the mentioned structural and operational aspects, it's evident that using a mechanical transmission line for the collecting machine is not advantageous. The hydraulic transmission line, on the other hand, appears more suitable, utilising the hydraulic power supply from the tractor. Nonetheless, the operational cost of the machine conjugate during the cultivation process remains a significant concern. Given that the machine conjugate is designed based on a commercial tractor, which already possesses a fixed configuration with a constant hydraulic flow source, optimising hydraulic power supply for the collecting machine isn't feasible. The primary objective of a hydraulic transmission line design in this instance is to minimise power loss as much as possible. The remainder of this article delves into an analysis of potential hydraulic circuit options to determine the most effective design solution for efficient hydraulic transmission deployment.

2. Materials and methods

2.1. Analysis on hydraulic circuit options

From the foregoing discussion, the hydraulic transmission design needs to cater to the roller collecting machine's flexible operational needs while utilising a constant hydraulic flow source from the tractor. Furthermore, for realistic deployment and to curtail manufacturing costs, the design should be straightforward and utilise affordable components readily available in the market. Detailed operational requirements include: the revolution velocity of the roller system should be continuously adjustable; roller velocity should remain consistent at a specific level; power loss should be minimised; the design should be simple design and cost-effective for deployment.

2.2. Speed control using variable displacement motor

There are two fundamental methods to construct a speed control circuit: (1) Utilising variable displacement pumps; (2) Combining fixed displacement pumps with flow control valves [1]. In the first method, the hydraulic motors' speed can be modulated by adjusting the pumps' volumetric displacement, thereby regulating fluid flow from the pumps to the motors. When pumps' displacement remains fixed, analogous circuits can be achieved with variable displacement motors. Circuits based on this principle, exemplified in Fig. 3, ensure high hydraulic efficiency, minimising power loss and providing prompt response to control actions [2].

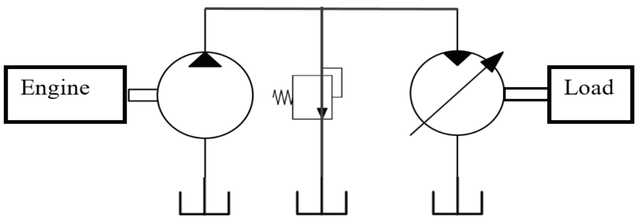

Fig. 3. Speed control circuit using a variable displacement motor.

However, speed regulation in the circuit depicted in Fig. 3 necessitates a closed-loop control system (illustrated in Fig. 4) to maintain the motor's constant angular velocity [3, 4]. This feedback mechanism involves feeding the output value (either load or motor angular velocity) back to the controller. This system's inherent complexity, especially considering the cost of variable displacement motors, sensors, and control equipment, makes it an expensive proposition.

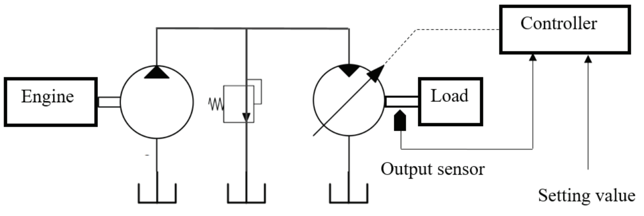

Fig. 4. Principle of the closed loop control system.

2.3. Speed control using hydraulic orifice

Orifices are commonplace in hydraulic component markets. By pairing fixed displacement pumps/motors with orifices, one can devise a speed control circuit. This method typically offers a more economical alternative for a speed control hydraulic circuit compared to methods relying on variable volumetric displacement pumps/motors [5]. The generally lower costs of fixed displacement pumps/motors, owing to their uncomplicated structure, account for this affordability.

Figure 5 showcases a speed control circuit iteration with an orifice in the main line. The design calls for a pressure relief valve, which caps the system's maximum pressure, effectively establishing an almost uniform pressure source while concurrently controlling the hydraulic fluid flow into the motors.

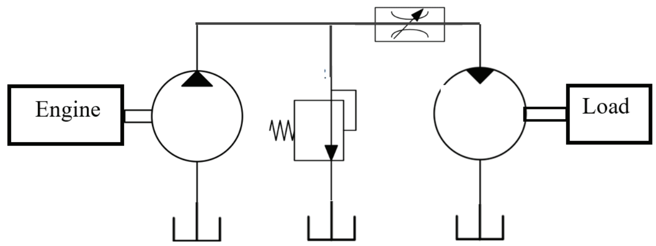

Fig. 5. Speed control circuit with orifice in main line.

While this configuration boasts simplicity and cost-effectiveness, it is marred by considerable power loss. This inefficiency stems from the continuous need for high system pressure irrespective of load demand. Two factors contribute to this power loss: (1) Speed control necessitates reaching the pressure relief valve's set value due to orifice adjustments; (2) The orifice induces pressure loss as given by:

$$P=\Delta P_{tl}+\Delta P_{t} \tag{1}$$

where P is the system pressure; ΔP_{tl} represents the drop pressure at the orifice; ΔP_{t} stands for the drop pressure at the load motor.

Under uncertain load conditions, the system pressure - dictated by the pressure relief valve - must be sufficiently high to ensure the system operates optimally. Large magnitude load fluctuations lead to further power loss. Another inherent flaw is the orifices' inability to stabilise motor speed. Theoretically, when the load alters, the pressure at the orifice's inlet shifts correspondingly, causing fluctuating fluid volumes returning to the tanks. Consequently, the fluid volume supplied to load motors becomes inconsistent, changing motor speed.

Practically, if the pressure relief valve exhibits a broad regulation spectrum, as demonstrated in Fig. 6, and load changes remain relatively minor, motor speed can be considered nearly constant with negligible error [6]. However, pinpointing the real load proves challenging, especially when trying to confine its variation to align with the pressure relief valve's specifications.

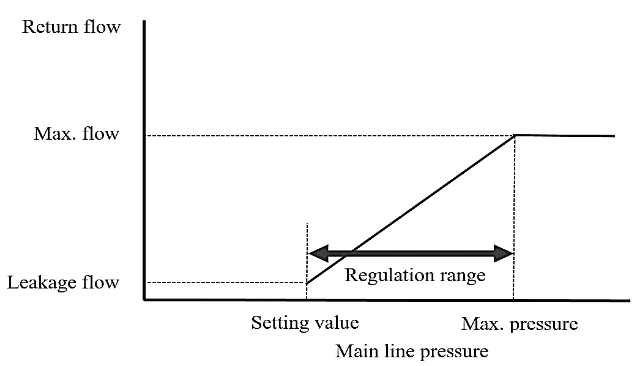

Fig. 6. Characteristics of the pressure relief valve.

To attenuate power loss, one could position the orifice in the return line [5, 7]. Circuits adopting this strategy are depicted in Fig. 7. These designs can adjust system pressure in line with the load, hence reducing power loss. However, this circuit cannot regulate speed since the return flow through the orifice depends on the main line pressure, leading to varied flow into the load motors. In this context, the pressure relief valve merely acts as a safety mechanism [7].

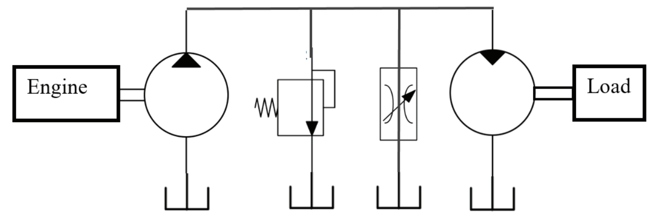

Fig. 7. Circuits using an orifice in the return line.

2.4. Proposed approaches using pressure compensated flow control valves

Pressure compensated flow control valves are a type of flow valve. They are designed to produce a set constant flow rate, largely independent of the load. The principal structure of a pressure compensated flow control valve with a variable flow rate is presented in Fig. 8.

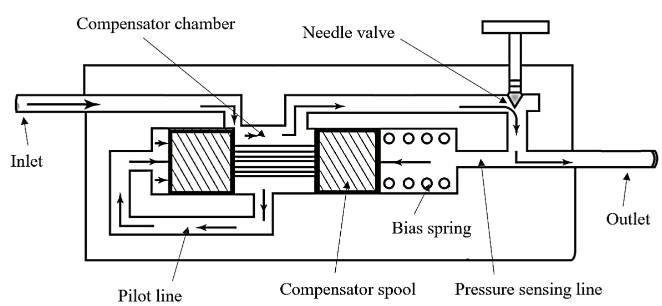

Fig. 8. Principle of pressure compensated flow control valves.

Using pressure compensated flow control valves in place of orifices in a speed control circuit offers an effective solution, especially in cases of fluctuating loads [5, 8]. Theoretically, these valves can sustain a constant flow rate, regardless of load. The flow rate value is derived from the differential pressure between the outlet and inlet of the valve, as follows:

$$Q=C.A.\sqrt{\frac{2.\Delta P_{tl}}{\rho}} \tag{2}$$

where C stands for the discharge coefficient; A represents the open area of needle valve; $\Delta P_{tl}=P_k-P_r$ defines the drop pressure through the valve with inlet pressure P_k and outlet pressure P_r; ρ denotes the hydraulic oil density.

Once other parameters are established, the flow rate relies solely on the differential pressure ΔP_{tl}. At equilibrium, the differential pressure across the needle valve satisfies the following equation:

$$P_k \times A_p - F_l - P_r \times A_p = 0 \tag{3}$$

or equivalently

$$P_k - P_r = \frac{F_l}{A_p} \tag{4}$$

where F_l is the force generated by the bias spring; A_p denotes the compensator spool area. As inferred, this differential pressure is always set by the bias spring force, ensuring the flow rate remains independent of the load pressure (P_r). In real scenarios, the flow rate is still alter when the load fluctuates significantly and rapidly [9-11].

Considering the effect of wide-ranging load variations on the flow rate through a pressure compensated flow control valve, the bias spring force component, F_l, in Eq. (4) is given by:

$$F_l = k\left(x_0 + \Delta x\right) \tag{5}$$

where k denotes the spring hardness; x_o is the initial compression of the spring; Δx represents the change of spring compression when load varies.

For minor load alterations (corresponding to a low Δx value), the variation in bias spring force is negligible. According to Eq. (4), the differential pressure remains fairly constant, resulting in a constant flow rate. However, with extensive load shifts, Δx must be taken into account. The discrepancy between the actual and ideal differential pressures can be calculated as follows:

$$\tilde{\Delta} P = \frac{k.\Delta x}{A_p} \tag{6}$$

This introduces errors in the flow rate through the valve. Moreover, research results in [11] reveal that while flow rate errors are minimal at low fluid flow velocities, they become significant at higher velocities.

Load change rates also considerably influence the performance of the flow control valve due to the valve mechanism's dynamics. Mechanical dynamics invariably introduces a transitional phase at the outset of the regulation process. The length of this phase varies depending on individual valve structural parameters. This dynamic mechanism is explained by a second-order dynamical equation; see [11, 12] for specifics. Research of D.T. Hieu, et al. (2009) [11] also indicates that the transitional phase's duration is affected by the rate of load change; a faster change rate elongates the transition, increasing flow rate discrepancies.

Pressure compensated flow control valves can be implemented in two configurations, akin to orifice-based setups. It's important to note that to achieve continuous motor speed control, the flow control valve must be of the variable flow rate variety, integrated with a needle valve (as illustrated in Fig. 8). In the primary configuration, with the flow control valve situated in the main line (refer to Fig. 9), the circuit results in substantial power loss. Nonetheless, the motor speed remains more stable, especially when compared to orifice-based circuits.

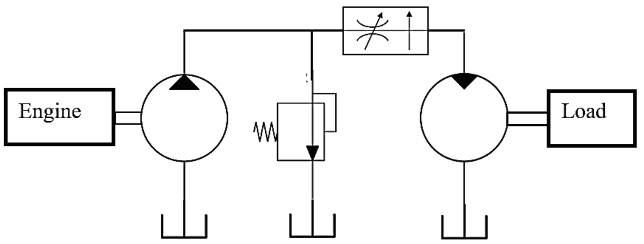

Fig. 9. Speed control circuit using a pressure compensated flow control valve in the main line.

In the second configuration, where flow control valves are positioned in the return line [5, 13], the circuit proves more efficient compared to the previously mentioned options (see Fig. 10). However, its capability to maintain a constant speed is compromised because the flow control valve, responsible only for the return flow [14], cannot detect changes in the main line's flow.

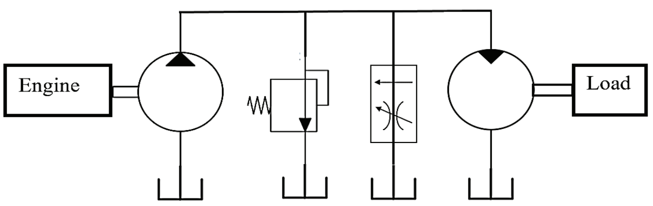

Fig. 10. Circuit using a pressure compensated flow control valve in the return line.

By substituting the two-way pressure compensated flow control valve with a three-way version in the main line (as depicted in Fig. 11), the resulting circuit's efficiency matches that of Fig. 10. Moreover, speed regulation improves as the valve directly controls the flow into the load motor.

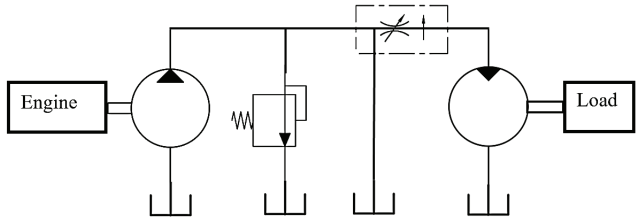

Fig. 11. Circuit using three-way pressure compensated flow control valves.

A circuit that utilises a flow control valve in the main line paired with an orifice in the return line (illustrated in Fig. 12) to modulate the motor speed has been implemented in agricultural and forestry machines [15]. This setup can sustain a steady motor speed. Nevertheless, the system's pressure must be at its peak, resulting in significant power loss similar to the circuit in Fig. 5.

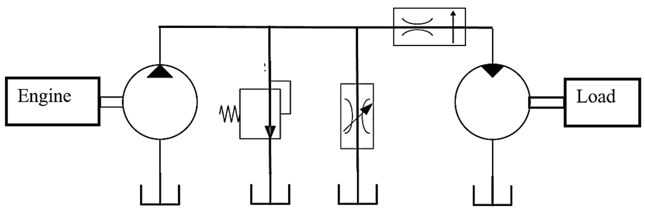

Fig. 12. Circuit using flow control valves and variable orifices.

For scenarios where the maximum load value is known, circuits with pressure compensated flow control valves in the main lines serve as optimal solutions for speed regulation. To mitigate power loss, a variable pressure relief valve can be integrated into the return line, maintaining the lowest pressure in line with the load demand. This configuration is illustrated in Fig. 13.

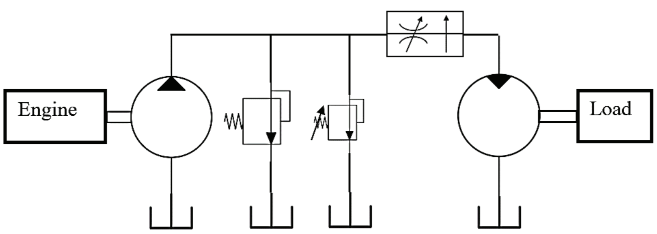

Fig. 13. Circuits using an auxiliary variable pressure relief valve.

From the speed control circuits analysed, those that employ: 1. A variable flow control valve in main line (Fig. 9); 2. A variable flow control valve in return line (Fig. 10); 3. A variable three-way flow control valve (Fig. 11); 4. A combination of a variable flow control valve and an auxiliary pressure relief valve (Fig. 13) appear to satisfy the system's operational requirements.

In the Fig. 9 circuit, the pressure relief valve acts solely as a safety feature. Notably, this valve is already integrated into the hydraulic system of the tractor. Thus, implementing this circuit demands only a fixed displacement motor and a common pressure compensated flow control valve, both readily available at affordable prices. Similarly, the Fig. 10 circuit comprises the same hydraulic components. Setting up the Fig. 11 circuit necessitates a three-way flow control valve to replace the two-way version. Recent designs for this type of hydraulic valve employ electro-mechanical actuator mechanisms, making them more costly than standard flow control valves. The final circuit in Fig. 13, besides requiring a flow control valve, also needs an auxiliary pressure relief valve - a component commonly available in the market.

2.5. Research approach

This article is not intended to systematically prove the characteristics of speed control circuits, which have been analysed in previous content. Instead, its focus is on application results, where the circuits are constructed using selected low-cost hydraulic components available on the market. The working capabilities of the proposed circuits are analysed using the professional hydraulic simulation toolbox, Simscape/Hydraulic, provided by Matlab/Simulink software. The simulation results help in deciding the most suitable circuit to deploy on real systems.

3. Results and discussion

The carrot harvesting machine system is powered by a Kubota L4508VN tractor. This tractor is equipped with an auxiliary hydraulic power system of 9 kW. The maximum flow rate of the hydraulic pump is 31.7 litres per minute, and the maximum hydraulic pressure is 17.7 MPa [16]. The collecting roller system requires an adjustable speed range from 300 to 500 rpm, depending on actual working conditions. Due to structural space limitations, the rollers are directly driven by the hydraulic motor without a reduction gearbox. Consequently, low-speed hydraulic motors are chosen for their high torque and efficiency.

According to the datasheet of available hydraulic motors on the market [17], motor BMR50, with a volumetric displacement of 51.7 ml/r, is suitable. Its speed ranges from 10 to 775 rpm, corresponding to the flow rate from 0.527 to 40.06 litres per minute. This motor, therefore, requires a flow rate from 15.5 to 25.85 litres per minute for the desired speed range. This flow rate matches the supply value from the source pump, ensuring continuous speed control. The motor's pressure, with an intermittent maximum of 17.5 MPa and a continuous maximum of 14 MPa, nearly meets the system's pressure requirements.

Based on preliminary calculations, the pressure-compensated flow control valve needs to accommodate a maximum flow rate of 16.2 litres per minute when placed in the return line (Fig. 10) and 28.85 litres per minute when located in the main line (Fig. 9). Based on the datasheet for two-way flow control valves [18], valve

FG-02-30-30 - adjustable for flow rates from 0.05 to 30 litres per minute with a maximum pressure of 21 MPa - is suitable. Similarly, a three-way pressure-compensated flow control valve, EFG-02-30, is chosen for the circuit in Fig. 11. This valve's flow rate ranges from 0.3 to 30 litres per minute with a maximum pressure of 20.6 MPa. For the pressure relief valve in the Fig. 13 circuit, valve DT-02-C-22 can be used. Its regulation pressure ranges from 3.5 to 14 MPa with a maximum flow rate of 16 litres per minute. The specifications of the selected valves are summarised in Table 1.

Table 1. Specification of the selected hydraulic valves.

Selected valve	Min. flow rate	Max. flow rate	Regulation pressure	Max. pressure
FG-02-30-30	0.05 l/min	30 l/min	-	21 MPa
EFG-02-30	0.3 l/min	30 l/min	-	20.6 MPa
DT-02-C-22	-	16 l/min	3.5 to 14 MPa	21 MPa

Using the machine system's design parameters and the selected circuit components, circuit analysis is performed with the Simscape/Hydraulic toolbox. This toolbox offers a realistic simulation environment with a library of common hydraulic components. These components factor in the effects of hydraulic losses in the circuits, making model implementation straightforward. Two admission criteria are specified: motor speed regulation capability; circuit energy efficiency.;

The energy losses in circuits is defined as follows:

$$\Delta E = \frac{E_v - E_r}{E_v}100 \tag{7}$$

where $E_v = \int_0^t n_p.M_p dt$ (8)

which denotes the input energy supplied from the tractor engine to the hydraulic pump during simulation time t. Here, the pump speed n_p is assumed to be constant and M_p stands for the torque at pump shaft. Similarly,

$$E_r = \int_0^t n_m.M_m dt \tag{9}$$

defines the output energy of the hydraulic motors. The two variables n_m, M_m are the speed and torque of hydraulic motor, respectively. Defining energy loss this way includes all system losses such as leakage in pumps/motors, valves, and losses in pipelines.

Technical specifications [18] for the chosen valves corresponding to each circuit are input into the parameter settings of relevant Simulink/Simscape models. The circuits are initially set with a fixed load torque of 40 Nm on the motor shaft, with the motor speed set at 400 rpm. To analyse the circuits under varying loads, a Gaussian random signal is added, with a frequency of 3 Hz, zero mean, and a standard deviation of 0.3. The resulting random load signal is displayed in Fig. 14.

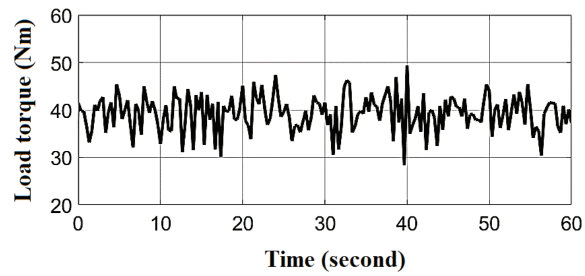

Fig. 14. Random load torque applied at the motor shaft.

As can be seen from Fig. 14, the load varies over a broad range from 30 to 50 Nm with rapid and random fluctuations. The test results for the four selected circuits are presented in Figs. 15-18.

The plots in Fig. 15 depict changes in motor speed under the impact of random variation in load for the four analysed circuits: the first circuit (Circ. 1) uses the FG-02-30-30 two-way pressure compensated flow control valve in return line (Fig. 10); the second one (Circ. 2) uses the EFG-02-30 three-way pressure compensated valve (Fig. 11); the third one (Circ. 3) exploits FG-02-30-30 valve in main line with an auxiliary pressure relief valve DT-02-C-22 in return line (Fig. 13); and the fourth one (Circ. 4) is the conventional speed control circuit using FG-02-30-30 valve in main line (Fig. 9). The errors in motor speed compared to the set value of 400 rpm are summarized in Table 1.

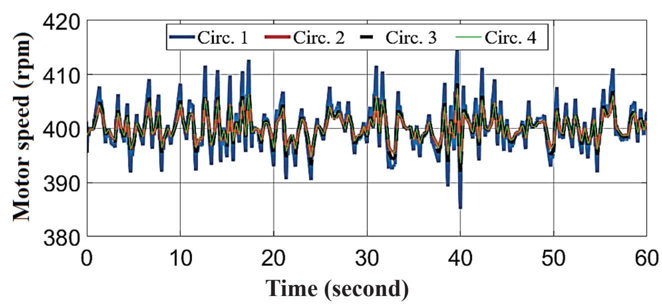

Fig. 15. Motor speed of the four analysed circuits.

Table 2. Summary of motor speed error.

Speed regulation error	Circ. 1	Circ. 2	Circ. 3	Circ. 4
Maximum absolute error (rpm)	15.20	7.45	8.55	7.48
Relative error (%)	3.8	1.9	2.1	1.9

From the results in Table 2, it's evident that the second and fourth circuits are the optimal solutions for motor speed regulation with a minor relative error of 1.9%. Meanwhile, the third circuit maintains the error within 2.1%, and the first circuit has a maximum error of 3.8%.

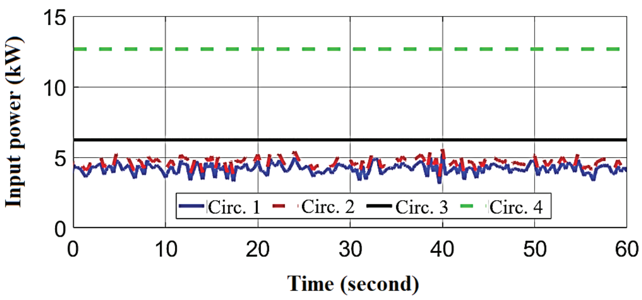

Fig. 16. Mechanical power consumption of the circuits.

Figure 16 illustrates the power sourced from the tractor engine to the circuits. The third and fourth circuits employ flow control valves in the main line. In the fourth circuit, system pressure consistently reaches its peak, leading to a maximum power consumption of 13 kW. The third circuit features an auxiliary pressure relief valve, thereby allowing for significant power conservation when compared to the fourth one. The first circuit, with its flow control valve in the return line, demands the least input power, followed by the second circuit, which employs a three-way flow control valve.

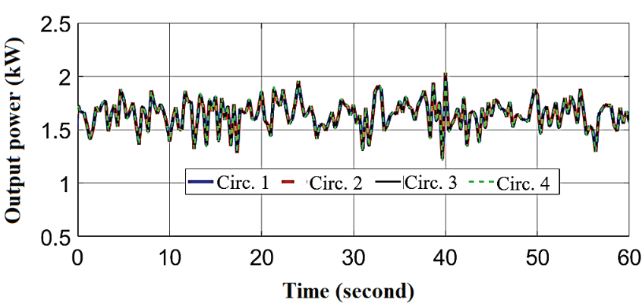

Fig. 17. Mechanical output power of the circuits.

Figure 17 showcases the mechanical power measured at the motor shaft. Given the minor differences in motor speed amongst the circuits, their output powers are relatively similar, hovering around 1.7 kW.

The energy efficiency of each circuit is determined by the ratio of input to output powers. As these figures fluctuate with the load, energy efficiency metrics adjust accordingly. Using Eqs. (7-9), the average values of efficiency are determined based on input and output energy computations. Fig. 18 provides insights into the percentage of energy losses for each circuit.

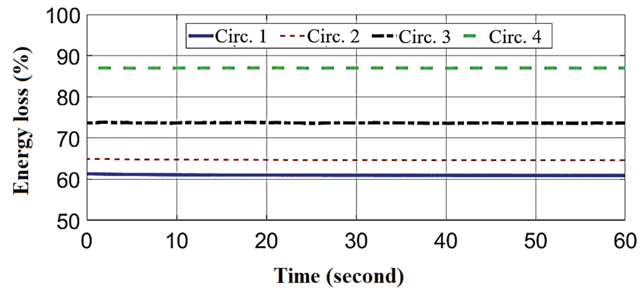

Fig. 18. Energy losses in the analysed circuits.

From Fig. 18, it's clear that the fourth circuit, with the FG-02-30-30 valve in the main line, incurs the highest energy loss at 88%. The third circuit, equipped with an auxiliary pressure relief valve, reduces energy loss to 73% for the given load. The second circuit, utilising the EFG-02-30 three-way flow control valve, sees a loss of 64%. Impressively, the first circuit, with the FG-02-30-30 in the return line, records the lowest energy loss at around 61%. Comparison criteria are presented in Table 3.

Table 3. Summary of comparison criteria.

Comparison criteria	Circ. 1	Circ. 2	Circ. 3	Circ. 4
Relative speed error (%)	3.8	1.9	2.1	1.9
Energy loss (%)	61	64	73	88
Component cost (estimated level)	1	3	2	1

Upon evaluating the comparison criteria, decisions regarding the most viable speed control circuit for practical application can be made. Clearly, the second and fourth circuits excel in speed regulation. However, the fourth circuit's high energy loss and the second circuit's elevated component cost render them less appealing. The third circuit, marked by significant energy loss, high cost, and large error, is also less than ideal. The first circuit emerges as the preferred option due to its cost-effectiveness, minimal energy loss, and a manageable error of 3.8% - well within an acceptable range.

4. Conclusions

This paper has delved into the functionality and efficiency of four distinct options for the speed regulation hydraulic circuit in a carrot collecting machine, specifically examining energy loss and implementation costs. Of the four circuits, Circs. 2-4 are conventional and frequently employed in hydraulic drive systems. Conversely, Circ. 1 represents a novel arrangement in which the pressure-compensated flow control valve is positioned in the return line, parallel to the hydraulic motor. With an aim towards genuine working conditions, these circuits were tested under fluctuating loads imposed on the hydraulic motor. Subsequently, their speed regulation functionality and energy efficiency were critically examined.

The simulation tests, conducted using real component parameters, offer a dependable analysis, underlining the proficiency and adequacy of each circuit in meeting system operational demands. It's clear from the findings that there isn't a singular, 'one-size-fits-all' solution for the speed control circuit; different contexts demand different compromises among the selection criteria.

From this standpoint, the first circuit - deploying the FG-02-30-30 pressure-compensated flow control valve in the return line (Fig. 10) - emerges as a favoured choice. Its strengths lie in its cost-effective implementation and diminished energy loss. These benefits translate into better fuel economy and a reduced burden on the cooling system, rendering it particularly advantageous for use in a carrot collecting machine.

CRediT author statement

Ngoc Danh Dang: Conceptualisation, Methodology, Software, Validation, Formal analysis, Investigation, Writing - Original draft preparation; Xuan Thiet Nguyen: Writing - Reviewing and Editing, Resources, Project administration.

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COMPETING INTERESTS

The authors declare that there is no conflict of interest regarding the publication of this article.

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