Investigation On The Influence of A Novel Redirector With By-Pass Damping On The Performance of Load Sensing Steering System

Yu-Qi Wang

Jilin University <https://orcid.org/0000-0001-6670-4295>

Xin-Hui Liu

Jinlin University

Jin-Shi Chen (✉ spreading@jlu.edu.cn)

Dong-Yang Huo

Jilin University

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Title page

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Yu-Qi Wang, born in 1993, is currently a PhD candidate at *School of Mechanical and Aerospace Engineering, Jilin University, China*. She received bachelor's degree from *Jilin University, China*, in 2016. Her research interests include fluid power transmission and control.

Tel: +86-188-44197058; E-mail: yuqi18@mails.jlu.edu.cn

Xin-Hui Liu, born in 1962, is currently a professor at *School of Mechanical and Aerospace Engineering, Jilin University, China*.

Jin-Shi Chen, born in 1983, is currently an associate professor at *School of Mechanical and Aerospace Engineering, Jilin University, China*.

E-mail: spreading@jlu.edu.cn

Dong-Yang Huo, born in 1996, is currently a master candidate at *School of Mechanical and Aerospace Engineering, Jilin University, China*.

Corresponding author: Jin-Shi Chen E-mail: spreading@jlu.edu.cn

ORIGINAL ARTICLE

Investigation on the Influence of a Novel Redirector with By-pass Damping on the Performance of Load Sensing Steering System

Yu-Qi Wang¹ • Xin-Hui Liu¹ • Jin-Shi Chen¹ • Dong-Yang Huo

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Abstract: The load sensing steering system of articulated loaders is prone to large pressure shock and oscillation during steering operations, affecting the stability of the steering system. To improve this phenomenon, an optimized structure of the redirector with by-pass damping is proposed, which adding orifices and throttle grooves to the original redirector. In the study, to control the steering load and working conditions, the steering load of the loader is replaced by using a pressure regulating valve. The simulation and experimental results shows that, compared with the traditional redirector, using the novel structure redirector with by-pass damping ensures good load sensing characteristics, the peak output pressure shock caused by the load unit step signal decreases from 65.04bar to 56.35bar, with a decrease rate of 13.4%, reducing the pressure oscillation of the hydraulic system. The pressure fluctuation time can be reduced from 2.09s to 1.6s, with a decrease rate of 23.4%, the output pressure oscillation decays faster, the smoothness of the steering operation is significantly improved.

Keywords: Articulated vehicle • Redirector • Load sensing steering system • Pressure oscillation

1 Introduction

The loader is mainly used for loading and unloading materials and shoveling work within a certain range [1-2]. During the operation, frequent steering operations are required. The articulated loader is widely used due to its compact structure and easy operation. Scholars have conducted a comprehensive study on the steering system of

articulated vehicles using numerical analysis and simulation methods [3-5]. More accurate steering performance and the main parameters affecting the performance were obtained. During steering operation of the existing articulated loader, when the driver turns the steering wheel, there are some unstable situations such as shocks and oscillations in the hydraulic system, resulting in poor stability of the machine when starting and stopping the steering. The higher the steering wheel speed, the more obvious the shaking or shimmy of the fuselage [6-7].

The optimization of steering system performance has also become a focus of scholars' research. Reference [8] introduced the steering method into the study of the steering system, combined the dynamic model with the physical model, and used the NSGA-II algorithm to optimize the steering system for further improve the performance of the system. A novel steering system has been developed to improve the driving performance during steering [9]. References [10-11] optimized the articulated frame to obtain a more compact articulated frame steering system, improving steering flexibility.

For the unstable situation of the hydraulic system, the introduction of a steering system with independent control of the inlet and outlet ports has improved the steering stability and steering sensitivity under high torque [12]. The load sensing pump [13], the proportional variable pump [14]-[17] and other variable displacement pump were used to replace the fixed displacement hydraulic pump for the single-rod hydraulic cylinder steering system [18] -[21] and the double-rod hydraulic cylinder steering system [22] to build a closed-loop steering system. The displacement of the pump was controlled by the rotation speed of the steering wheel, which can supply oil on demand, reduce the loss of the steering system and

✉ Jin-Shi Chen
spreading@jlu.edu.cn

¹ School of Mechanical and Aerospace Engineering, Jilin University, 130022, Jilin, China

improve the stability of the steering system. Caterpillar studied working and steering flow sharing hydraulic systems with the flow sharing valve [23]. The steering scheme of installing a by-pass proportional throttle valve at the outlet of a fixed displacement hydraulic pump was proposed in reference [24]. By-pass throttling reduces the outlet pressure of the hydraulic pump and the pressure shock of steering operating. Sany, Liugong and other companies mainly studied the improvement of the full-hydraulic steering system and load sensing hydraulic steering system with flow amplification valve [25-27]. EATON has done a lot of research on the structure optimization of the full hydraulic redirector, developed a full hydraulic redirector that can eliminate oil trapping [28]. Lanzhou University of Technology has done a lot of work on the development of the steering priority flow amplification valve and the structure optimization of the load sensing priority valve [29].

Based on the above research, scholars have researched and optimized hydraulic systems and components. Appropriately adjusted the oil supply pressure of the system, eliminated the oil trapped, which to some extent alleviated the pressure shock, oscillation and other instability of articulated vehicles during steering. However, under high speed condition, it is still difficult to control the pressure shock caused by the sudden flow changes, and the steering system will still have a large oscillation. For this reason, a redirector structure with by-pass damping is proposed.

The structure of this paper is as follows: Section 2 introduces the working principle of the redirector with by-pass damping. Section 3 deduces the relationship between the valve port area of the redirector and establishes a mathematical model. To research the relationship between the flow rate, the pressure and rotation speed of the redirector. In Section 4, the simulation model of the redirector with by-pass damping is established and verified the performance, the simulation results are obtained. Section 5 builds a simulation experimental platform and obtains experimental results of the by-pass damping redirector. Section 6 analyzes the dynamic characteristics and oscillation analysis of the novel redirector and the original redirector. Section 7 is the conclusion.

2 Working Principle

The research object of this paper is the coaxial flow rate amplification redirector, whose structure is shown in Figure 1 (a). It is mainly composed of three parts: housing, metering motor and rotary valve (including control spool

Figure 1 Schematic diagram of the redirector. (a) Structure of the redirector; (b) Structure of the rotary valve; (c) Principle diagram of the redirector.

and control sleeve). The control spool of the redirector is connected to the steering wheel, and the control spool rotates synchronously when the steering wheel is turned. There is a relative angular displacement between the control spool and the control sleeve. The input P opens after the control spool rotates a certain angle. The fluid enters through the input and pushes the metering motor to rotate. The metering motor is mainly used to ensure that the flow rate of oil delivered by the redirector is proportional to the steering wheel angle when the steering pump is working. If the steering pump is not working, the

motor directly sucks oil from the oil tank and supplies it to the steering cylinder when rotating the steering wheel.

The metering motor is mainly used to ensure that the flow rate of oil delivered by the redirector is proportional to the steering wheel angle when the steering pump is working; when the steering pump is not working, manually turn the steering wheel, the motor directly sucks oil from the oil tank and supplies it to the steering cylinder. The faster the rotation speed of the steering wheel, the greater the relative angular displacement of the control spool and control sleeve, the flow rate is greater, and the steering speed is faster. The rotary valve is mainly composed of a control spool with four groups of grooves and a control sleeve with ten groups of through holes. When the control spool rotates, the direction and flow rate of the oil are changed by the on-off and opening area of different grooves and through holes, as shown in Figure 1 (b).

Figure 1 (c) is the principle diagram of the redirector. When the steering wheel is not rotated, the redirector is in the medium position, and the port P is disconnected to ports R and L. The oil enters the channel C through ports P and LS and returns to the tank through the port T. The steering rod drives the control spool to rotate when the steering wheel rotates to the left. The oil enters the even-column H of the C channel through the P port and reaches the metering motor, then output from the odd-column H through the B channel and the L port to the steering cylinder. The port T is closed, the channel A connects the port LS, port R and the fuel tank, and the left steering is completed. When the steering wheel rotates at high speed, the steering rod drives the control spool to rotate a large angle, the oil enters the channel B through flow rate amplification holes and is directly output from parts L and R of the redirector to the steering cylinder to complete the steering. When the redirector is in the medium position, the by-pass damping hole is not connected to the port R or L. When the steering wheel rotates, the port R or L is connected to the port T through the by-pass throttle hole K. At this time, the pressure shock and oscillation caused by the displacement of the cylinder can be reduced by by-pass damping to achieve smooth steering.

3 Methodology and Model

3.1 Calculation of the Opening Area

It is necessary to calculate the opening area of each port and channel when researching the performance of the redirector. For the convenience of calculation, the

Figure 2 Expanded view of control spool and sleeve. 1. Inlet module; 2. Flow rate amplification module; 3. Metering motor module; 4. Load sensor module; 5. Working module; 6. Return module; 7. By-pass module.

Figure 3 Relation between the opening area of each module and the control spool displacement

assembly of the control spool and control sleeve along the axis is expanded, as shown in Figure 2. The channel portion of the control sleeve is indicated by solid lines, and the control spool portion is indicated by dotted lines. The through hole on the control spool is evenly distributed along the circumference, and 12 holes in each of P and H are distributed along the same center line. Holes H are numbered from left to right, odd-columns are aligned with the center line of holes L and even-columns are aligned with the center line of holes R. The position of by-pass damping holes K, holes LS and flow rate amplification holes are shown in the figure and the other through holes are T. There are 6 channels in each of A, B, C and D on the control sleeve distributing along the circumference. Channel A controls the on-off of L-T or R-T. Channel C controls the on-off of P-T and feedback of LS. Channel B is evenly distributed between channels A and C to control

the on-off of R-H or L-H and realize the performance of flow rate amplification. Channel D controls the on-off of L-K or R-K.

The oil flow path of the redirector is divided into 7 modules. The opening area of the inlet module P-C is

$$A_p = \begin{cases} 3r^2(\theta - \sin \theta) & (0 < x < 2r) \\ 6\pi r^2 & (2r < x) \end{cases}, \quad (1)$$

where $\theta = 2\arccos \frac{r-x}{r}$, r is diameter of the hole, θ is the angle corresponding to the opening area.

The opening area of the flow rate amplification module A-C is

$$A_{pA} = \begin{cases} 9r^2(\theta - \sin \theta) & (0 < x < x_1) \\ 6r^2(\theta - \sin \theta) + \left(\frac{3}{2}\theta + 1.02\right)r^2 & (x_1 < x < x_2) \\ 18\pi r^2 \left(\frac{3}{2}\theta + 1.02\right)r^2 - 6r^2(\theta - \sin \theta) & (x_2 < x) \\ +18\pi r^2 - 1.5r^2 \sin \theta - 3(r-x) & \end{cases}, \quad (2)$$

where x_1 , x_2 are displacements of two limit positions.

The opening area of the metering motor module when the oil goes through the inlet and outlet H of the metering motor is

$$A_H = \begin{cases} 1.5r^2(\theta - \sin \theta) + 3xl & (0 < x < 2r) \\ 3r(l + \pi r) & (2r < x) \end{cases}, \quad (3)$$

where l is the side length of motor hole.

The opening area of the working module H-R\L is

$$A_{R\L} = \begin{cases} 3r^2(\theta - \sin \theta) + 6xl & (0 < x < 5r) \\ 6r(2l + \pi r) & (5r < x) \end{cases}, \quad (4)$$

The opening area of the return module R\L-T is

$$A_T = \begin{cases} r^2(\theta - \sin \theta) + 2xl & (0 < x < a) \\ r^2(\theta - \sin \theta) + 6xl + 2r^2(\theta_1 - \sin \theta_1) & (a < x < 2r-a) \end{cases} \quad (5)$$

$$\text{where } \theta_1 = 2\arccos \frac{r-x+a}{r}$$

The opening area of the load sensor module when oil through LS is

$$A_{LS} = \begin{cases} 6\pi r^2 & (x < a) \\ 6\pi r^2 - \frac{3}{2}r^2(\theta - \sin \theta) & (a < x < a+l) \\ 3\pi r^2 - \frac{3}{2}r^2(\theta - \sin \theta) & (a+l < x < a+l+2r) \\ 0 & (a+l+2r < x) \end{cases}, \quad (6)$$

where a is the transverse distance of opening area.

The opening area of the by-pass module R\L-D is

$$A_D = \begin{cases} 3l(a-x) + \frac{3}{2}r^2(\theta - \sin \theta) & (0 < x < a) \\ 0 & (a < x) \end{cases}, \quad (7)$$

The opening area of the by-pass module D-K is

$$A_{D-K} = \begin{cases} \frac{1}{2}r^2(\theta - \sin \theta) & (0 < x < a) \\ \frac{1}{2}r^2(\theta - \sin \theta) + r^2(\theta_1 - \sin \theta_1) & (a < x < 2r) \\ \pi r^2 + r^2(\theta_1 - \sin \theta_1) & (2r < x < l) \\ \frac{1}{2}r^2(\theta_2 - \sin \theta_2) + r^2(\theta_1 - \sin \theta_1) & (l < x < 2r+a) \\ \frac{1}{2}r^2(\theta_2 - \sin \theta_2) + 2\pi r^2 & (2r+a < x < a+l) \\ \frac{1}{2}r^2(\theta_2 - \sin \theta_2) + r^2(\theta_3 - \sin \theta_3) & (a+l < x < l+2r) \end{cases} \quad (8)$$

$$\text{where } \theta_2 = 2\arccos \frac{x-r-l}{r}, \theta_3 = 2\arccos \frac{x-a-r-l}{r}$$

The opening area of the by-pass module K-T is

$$A_K = \begin{cases} \frac{3}{2}r^2(\varphi_1 - \sin \varphi_1) + \frac{3}{2}r^2(\varphi_2 - \sin \varphi_2) & (0 < x < 2r-l_2) \\ 3\pi r^2 + \frac{3}{2}r^2(\varphi_1 - \sin \varphi_1) & (2r-l_1 < x < l_1) \\ 3\pi r^2 & (l_1 < x < l_3) \\ \frac{3}{2}r^2(\varphi_3 - \sin \varphi_3) & (l_3 < x < 2r+l_3) \end{cases} \quad (9)$$

$$\begin{aligned}\varphi_1 &= 2 \arccos \frac{r+x-l_1}{r}, & \varphi_2 &= 2 \arccos \frac{r-x-l_2}{r}, \\ \varphi_3 &= 2 \arccos \frac{x-l_3-r}{r}.\end{aligned}$$

where l_1, l_2 are distances between two limit positions and two holes, l_3 is the distance between the two holes.

The relationship curve between the opening area of each module and the control spool displacement is shown in Figure 3.

3.2 Mathematical Model of the Redirector

For the purpose of further analyzing the dynamic characteristics of the redirector with by-pass damping structure, the mathematical model of the redirector is established. The following assumptions are accepted, ignore the pipeline loss, internal leakage, friction and hydraulic force of the redirector which have little effect on steering performance. According to the structure and working principle of the redirector, the schematic diagram of the equivalent damping is shown in Figure 4.

The combination of the metering motor seal oil cavity and the hole and channel structure of the rotary valve realizes the oil distribution relationship of the redirector. The flow rate continuity equation of the metering motor without considering leakage is

$$q_m = D_m \dot{\theta}_m, \quad (10)$$

where $\dot{\theta}_m$ is the angular velocity of motor, D_m is the displacement of motor.

During the working process of the redirector, it is mainly subjected to pressure uniformly distributed in the radial direction [30]. Since the control sleeve of the redirector is rigidly connected to the rotor of the motor, the force condition of the redirector is the same as that of the metering motor. According to the working principle of the redirector, the viscous friction and drag torque between the control sleeve and control spool of the rotary valve are main loads of the motor. The viscous friction can be ignored because it is much smaller than the drag torque. The force balance equation of the motor is

$$T_l = D_m (P_{H1} - P_{H2}), \quad (11)$$

where P_i is the pressure between each orifice in rotary valve.

The control spool and control sleeve of the redirector are expanded into a planar structure according to their axial

direction. The displacement of the control spool is

$$x_z = \frac{d}{2} \left(\int_0^t \omega_f dt - 2\pi \int_0^t \frac{q_m}{D_m} dt \right) = x_z(\omega_f), \quad (12)$$

where ω_f is the angular velocity of the rotary valve.

The differential pressure and flow rate of the redirector inlet are

$$\Delta P = P - P_{LS}, \quad (13)$$

$$q_1 = C_d A_1(x_z) \sqrt{\frac{2\Delta P}{\rho}}, \quad (14)$$

where C_d is the flow coefficient, ρ is density of oil.

According to the previous research and analysis of priority valves and redirectors [31-32], the differential pressure of the redirector orifice is basically a constant value. The flow rate through the inlet is only related to the opening area of the rotary valve, that is, it is related to the rotation speed of the control spool of the steering gear, and has nothing to do with the electromotor speed and steering load.

The flow rate of the redirector through each damping hole can be expressed

$$q_i = C_d A_i(x_z) \sqrt{\frac{2\Delta P_i}{\rho}}, \quad (15)$$

where A_i is the area of orifice, ΔP_i is the pressure drop through each orifice.

When the redirector is in the medium position, medium damping holes R_2, R_3 are working, other damping holes are in a negative opening state, and P_{LS} is low pressure.

$$q_z = q_2 = q_3, \quad (16)$$

When the redirector rotates, R_2, R_3 are closed. The flow rate of each damping hole is

$$q_z = q_7 = q_1 + q_6 = q_9 = q_{L-R} + q_8, \quad (17)$$

$$q_m = q_1 = q_4 = q_5, \quad (18)$$

As can be seen from Figure 4, the differential pressure of oil through each damping hole is: $\Delta P_2 = \Delta P$
 $\Delta P_3 = P_{LS} - P_T$ $\Delta P_4 = P_{LS} - P_{H1}$ $\Delta P_5 = P_{H2} - P_{FD}$,
, , ,

$$\begin{aligned}\Delta P_6 &= P - P_{FD} & \Delta P_7 &= P_{FD} - P_L & \Delta P_{L-R} &= \Delta P_8 = P_L - P_R \\ \Delta P_9 &= P_R - P_T, & & & & \end{aligned}$$

The flow rate-pressure relationship of the redirector can be transformed into

$$P = \frac{\rho}{2} \left(\frac{q_m}{C_d} \right)^2 \left(\frac{1}{(A_4(x_z))^2} + \frac{1}{(A_1(x_z))^2} + \frac{1}{(A_5(x_z))^2} \right) + \frac{T_l}{D_m} + \frac{\rho}{2} \left(\frac{q_m + q_6}{C_d} \right)^2 \left(\frac{1}{(A_7(x_z))^2} + \frac{1}{(A_9(x_z))^2} + \frac{1}{(A_8(x_z) + A_z)^2} \right) + P_T, \quad (19)$$

$$q_6^2 = \left(\left(\frac{A_6(x_z)}{A_1(x_z)} \right)^2 + \left(\frac{A_6(x_z)}{A_4(x_z)} \right)^2 + \left(\frac{A_6(x_z)}{A_5(x_z)} \right)^2 \right) q_m^2 + \frac{2T_l}{\rho D_m} (C_d)^2 (A_6(x_z))^2, \quad (20)$$

4 Simulation Method and Related Settings

4.1 Simulation Method

As shown in Figure 2, all the circumferential movements of the redirector are converted into translational movements. To simplify model structure, the metering motor is replaced by a meter cylinder with an inlet and an outlet replacing seven closed chambers and only six inlet holes are considered for translation in the same direction. In addition, the seven holes of the valve body corresponding to port H and the metering motor are equivalent to an inlet and a return port. The magnitude of the input force signal in the model represents the rotation speed of the steering wheel. The pressure regulating valve as equivalent load instead of the steering load. The model is established in accordance with the 7 oil circuit modules in Section 3.

Figure 5 shows the simulation model of the redirector.

4.2 Simulation General Setting

Table 1 Key parameters of the simulation model

Definition	Item	Value	Definition	Item	Value
Priority valve	d_{spool}	25mm	Redirector	d_{spool}	10mm
	d_{rod}	14mm		d_{rod}	5mm
	m	0.3kg		m_1	0.284kg
	F_{0k}	458.8N		m_2	0.301kg
Pump	k	28.3N/mm	Metering motor	m_0	0.6kg
	Q_t	80cc/rev		d_{piston}	79.6mm
	Signal	$F(x)$		ρ	850kg/m ³
		oil

Figure 4 Schematic diagram of the equivalent damping

Figure 5 Simulation model of the novel redirector.

The main parameters of the simulation model are given in Table 1. The hydraulic pressure acting on the control spool can be offset, so the acting area of the two relative valve blocks should be kept equal, and all parameters of the valve port area must be consistent. According to Equations (1)~(9), the opening area formed by the relative movement of the control spool and control sleeve is set. When the metering cylinder is used to replace the metering motor, the measuring area of the cylinder is calculated by the displacement of the motor.

$$d = \frac{4V}{\pi D^2}, \quad (21)$$

The speed signal of the steering wheel is converted to the speed of translation

$$v = 2\pi r n, \quad (22)$$

4.3 Validation of the Model

To verify whether the simulation model can well reflect the working principle and dynamic performance of the actual structure, the performance of the redirector with by-pass

Figure 6 Displacement of control spool and sleeve.

Figure 7 Relation between speed signal and ΔP , $\Delta P_{R/L-P}$.

damping under different working conditions is simulated and analyzed. Figure 6 is the displacement curve of the control spool and control sleeve under the certain rotating speed signal. It can be seen from the curve that the control spool begins to move immediately when the force signal is applied. The control sleeve does not move at the beginning due to the dead zone between the control spool and control sleeve. When the rotation speed increases and the port P is opened, the opening area and the displacement difference between the control spool and control sleeve increase. When the rotation speed is 0, the opening area is reduced to close and the control sleeve has a slight lag in stop movement relative to the control spool. The simulation results show that the control spool and control sleeve of the redirector ensure good follow-up performance when the steering wheel speed increases.

Figure 7 shows the differential pressure ΔP and the pressure loss $\Delta P_{R/L-P}$ of the redirector under the constant increase speed signal. It can be seen from curves that $\Delta P_{R/L-P}$ gradually increases with the increase of the redirector opening area, while ΔP remains at a constant value. The overflow area and flow rate of the redirector with the increase of the steering wheel rotating speed. However, $\Delta P_{R/L-P}$ increases with the overflow area increasing, because the overflow area gradient of each orifice is smaller than that of the flow rate.

Figure 8 Simulation results under no-load condition of electromotor speed is 708 r/min. (a) P ; (b) P_{LS} ; (c) $P_{R/L}$; (d) Q

Figure 9 Simulation results under no-load condition of steering wheel speed is 20 r/min. (a) P ; (b) P_{LS} ; (c) $P_{R/L}$; (d) Q

Figure 10 Simulation results under steering wheel speed 8r/min, electromotor speed is 708 r/min. (a) P ; (b) P_{LS} ; (c) P_L ; (d) P_R

Based on the simulation results, the model completely

Figure 11 Photograph of the redirector experiment platform. 1. Steering wheel; 2. Pressure sensor; 3. Data acquisition; 4. Redirector; 5. Pump; 6. Electromotor; 7. Controller.

Table 2 Main parameters and size of the experiment platform

Definition	Parameters
Pump	Displacement:80cc/rev, Max. rotation speed 2350rpm;
Data acquisition	Signal:1-5V, Accuracy: 0.02%, Channl:16;
Pressure sensor	Range:0-600bar, Accuracy: 0.5%, Voltage:12VDC;
Flow meter	Range:300L/min, Accuracy: 1%;
Tachometer	Range:0-3000r/min, Accuracy: 1%;

represents to the actual structure of the redirector, which can reflect the working principle and have good follow characteristics.

4.4 Simulation Results

The dynamic characteristics of the redirector with by-pass damping are analyzed from three aspects: different steering wheel speed, different electromotor speed and different steering load. Figure 8 shows the simulation results of the redirector under different steering wheel speeds with no-load condition of electromotor speed is 708 r/min. It can be seen from the simulation curves that that the flow rate into the redirector increases with the increase of the steering wheel speed. The flow rate has a linear relationship with the opening area A_1 when ΔP remains stable, the redirector has good operating performance. The flow amplification holes are opened wen the rotation speed is 20r/min and 30r/min, and the flow rate increases rapidly. Besides, this part of the flow rate directly reaches outlet of the redirector without passing through the metering motor.

Figure 9 shows the simulation results of the redirector under different electromotor speeds with no-load condition of steering wheel speed is 20 r/min. Change the electromotor speed, the flow rate into the redirector and the pressure of ports P, LS, L/R remain constant. This shows that the flow rate required by the steering system is constant and not affected by the electromotor speed when the steering wheel speed is fixed.

Figure 10 shows the simulation results under no-load and heavy-load conditions with the steering wheel speed and electromotor speed are fixed. The output pressure and

pressure shock of the redirector under heavy-load condition are higher than that of no-load, and the oil return pressure in both cases is the same. The output pressure and pressure fluctuation of the steering gear under heavy load are higher than that of no load, and the oil return pressure is the same in both cases. According to the analysis of simulation results, the change of load has no effect on the flow rate and differential pressure of the redirector with the same steering speed.

It can be seen from Figure10 that input the step signal, the rise time under no-load condition is 0.23s, which is shorter than 0.49s under heavy-load condition. The steering system response becomes slower when the load gets heavy.

5 Experiment of the Redirector

5.1 Simulation Experiment Platform

Although the simulation model has been set according to the actual structural parameters, the influence of pipeline has been ignored. Therefore, it is necessary to verify the correctness of the equivalent load and the model by experiments. According to the simulation system model in Figure 5, the simulation experiment platform of by-pass damping redirector is established, as shown in Figure 11. When assembling the platform, the bending degree of pipelines should be reduced as much as possible to reduce the influence of pressure loss on the measured data. Sensors should be installed as close to the pressure point as possible. In the experiment, the angular velocity of the steering wheel is collected by a speed measuring instrument. Pressure sensors are used to collect pressure of the hydraulic pump, the inlet and outlet of the redirector and the port LS. Flow meters are used to collect the flow rate of the redirector inlet and outlet. The experimental instruments required are given in Table 2.

5.2 Experimental Results

Three steering conditions with different steering wheel speed, different electromotor speed and different steering load were used to test and analyze the performance of the redirector with by-pass damping. The steering condition in the experiment is consistent with the simulation steering condition to ensure the accuracy of the experiment result. Figure 12 shows the experimental results of redirector with by-pass damping under the no-load condition with the electromotor speed is 708r/min when the steering wheel speed is 8r/min, 20r/min, and 30r/min, respectively. It can be seen that when the steering wheel speed increases, the flow rate through the redirector increases, resulting in the increase of pressure at port LS and port L/R.

Figure 12 Experimental results under no-load condition of electromotor speed is 708 r/min. (a) P ; (b) P_{LS} ; (c) $P_{R/L}$; (d) Q

Figure 13 Experimental results under no-load condition of steering wheel speed is 20 r/min. (a) P ; (b) P_{LS} ; (c) $P_{R/L}$; (d) Q

Figure 14 Experimental results under steering wheel speed 8r/min, electromotor speed is 708 r/min. (a) P ; (b) P_{LS} ; (c) P_L ; (d) P_R

The steering wheel speed is the main factor that affects the

flow rate of the steering system. When the speed is higher, the inlet flow rate of the redirector changes more obviously with the increase of steering wheel speed.

Figure 13 shows the experimental results under the no-load condition with the steering wheel speed is 20r/min when the electromotor speed is 708r/min, 1200r/min, and 1870r/min, respectively. It can be seen from curves that when the electromotor speed increases, the flow rate through the redirector and the pressure at port P, port LS, port R/L are basically constant. It can be concluded that the required flow rate of the steering system is determined by the steering wheel speed rather than the electromotor speed.

Figure 14 shows the experimental results under no-load and heavy-load conditions with the steering speed is 8r/min, and the electromotor speed is 708r/min, 1200r/min, respectively. The pressure at port P, port LS and port R/L increases significantly with the increase of steering load, while the pressure variation of port R is not obvious. Combined with the previous simulation analysis, the pressure of steering system increases, and the response slows down when the steering load increases.

Part of the simulation and experiment results are shown in Table 3, the maximum error is 11.1%, which proves that the simulation model has good accuracy. The model can be used to further analyze the steering performance.

Table 3 Part of the simulation and experimental results

Working condition	Category	P (bar)	P_{LS} (bar)	$P_{R/L}$ (bar)	Q (L/min)
708r/min	Simulation	20.05	9.51	7.45	4
8r/min	Experiment	20.38	10	8.34	3.6
No-load	Error	1.6%	4.9%	10.6%	11.1%
708r/min	Simulation	36.8	26.61	17.75	31.53
30r/min	Experiment	39.38	29.72	19.34	30.22
No-load	Error	6.5%	10.4%	8.2%	4.3%
1200r/min	Simulation	29.63	18.91	13.33	16.91
20r/min	Experiment	30.19	19.47	13.6	15.86
No-load	Error	1.8%	2.9%	0.5%	6.6%
1870r/min	Simulation	30.2	19.06	13.44	16.56
20r/min	Experiment	31.2	19.7	13.8	15.97
No-load	Error	3.2%	3.2%	2.6%	3.7%

6 Analysis of Dynamic Characteristics and Oscillation

According to the force equation and flow continuity equation of the redirector and steering load during the steering process

Figure 15 The schematic of redirector mathematical model

$$A_L P_L - A_R P_R = m \frac{du}{dt} + Bu(t) + F(t), \quad (23)$$

$$q_{L-R}(t) = A_L u(t) + \frac{V}{E} \frac{dP_L(t)}{dt} + \lambda_c [P_L(t) - P_R(t)], \quad (24)$$

where V is the chamber volume of the equivalent load, E is the effective bulk modulus of oil, F is the external load, $u(t)$ is the speed of the equivalent load response, B is the viscous damping, λ_c is the leakage coefficient of the equivalent load.

The redirector with by-pass damping has the pressure-flow rate linearization equation of steering load

$$q_8 = q_7 - q_{L-R} = K_q A_8(s) + K_c(P_L - P_R), \quad (25)$$

where $K_q = \frac{\partial q_8}{\partial x} = C_d \sqrt{2 \frac{P_L - P_R}{\rho}}$, $K_c = \frac{\partial q_8}{\partial P_L} = \frac{C_d A_8}{\sqrt{2 \rho (P_L - P_R)}}$

Dynamic characteristics block diagram of the two redirector structures is obtained by Laplace transformation, as shown in Figure 15. Variables inside the dotted line is the influence of the by-pass damping structure.

If the right outlet pressure $P_R(s) = 0$, the flow rate $q_7(s) = 0$, the dynamic transfer function with the load as the input and left outlet pressure as the output can be obtained

$$\frac{P_L(s)}{F(s)} = \frac{\frac{A_L E}{mV}}{s^2 + \left(\frac{\lambda_c E}{V} + \frac{B}{m} \right) s + \frac{B \lambda_c + A_L^2}{mV} E} = \frac{K_{A1} \omega_{n1}^2}{s^2 + 2\xi_1 \omega_{n1} s + \omega_{n1}^2} \quad (26)$$

Combining equations (23), (24), and (25), the dynamic transfer function of the redirector with by-pass damping is obtained

$$\begin{aligned} \frac{P_L(s)}{F(s)} &= \frac{\frac{A_L E}{mV}}{s^2 + \left(\frac{E(\lambda_c + K_c)}{V} + \frac{B}{m} \right) s + \frac{(\lambda_c + K_c)B + A_L^2}{mV} E} \\ &= \frac{K_{A2} \omega_{n2}^2}{s^2 + 2\xi_2 \omega_{n2} s + \omega_{n2}^2} \end{aligned}, \quad (27)$$

Gains of the two structure redirectors are

$$K_{A1} = \frac{A_L}{B \lambda_c + A_L^2}, \quad (28)$$

$$K_{A2} = \frac{A_L}{(\lambda_c + K_c)B + A_L^2}, \quad (29)$$

Undamped natural frequencies are

$$\omega_{n1} = \sqrt{\frac{E}{mV} (B \lambda_c + A_L^2)}, \quad (30)$$

$$\omega_{n2} = \sqrt{\frac{E}{mV} (B \lambda_c + B K_c + A_L^2)}, \quad (31)$$

System damping ratios are

$$\xi_1 = \frac{\omega_{n1}}{2A_L^2} \left(m \lambda_c + \frac{BV}{E} \right), \quad (32)$$

$$\xi_2 = \frac{\omega_{n2}}{2A_L^2} \left(m \lambda_c + m K_c + \frac{BV}{E} \right), \quad (33)$$

Outlet characteristic equations of the two structure redirectors are as follows

$$s^2 + \left(\frac{\lambda_c E}{V} + \frac{B}{m} \right) s + \frac{E}{mV} (B \lambda_c + A_L^2) = 0, \quad (34)$$

$$s^2 + \left(\frac{(\lambda_c + K_c)E}{V} + \frac{B}{m} \right) s + \frac{E}{mV} (B \lambda_c + B K_c + A_L^2) = 0, \quad (35)$$

The pressure overshoot is

$$M_{pn} = e^{-\frac{\xi_n \pi}{\sqrt{1-\xi_n^2}}}, \quad (36)$$

Figure 16 Flow rate through by-pass orifices.

Figure 17 Pressure of the equivalent load inlet.

According to the Routh–Hurwitz stability criterion, it is known that both systems formed by the two redirectors are stable.

Based on above equations, when the steering load F increases, P_L increases, K_c decreases, ω_n , ξ will decrease and the pressure overshoot M_p will increase. The number of system oscillations increases, the stability decreases, and the adjustment time increases. Therefore, the sudden increase of the steering load will lead to the system oscillation. If the opening area A_8 of the orifice R_8 increases, K_c increases, ξ will increase and the pressure overshoot M_p will decrease. the number of oscillations will decrease, and the corresponding stability of the system will increase. In this study, $\xi_1 < \xi_2$, $M_{p1} > M_{p2}$, and the overshoot of the redirector with by-pass damping is relatively small.

The severity of system oscillation will be determined by the damping ratio ξ . The smaller ξ , the more severe the oscillation. The decay speed of the oscillation process is determined by $|\xi\omega_n|$. The larger $|\xi\omega_n|$, the faster the decay will be. According to the above analysis

$$|\xi\omega_n|_1 < |\xi\omega_n|_2, \quad (37)$$

Therefore, compared with the original redirector, the output pressure oscillation decays more quickly and the system is more stable when a step signal is input to the redirector with by-pass damping.

Figure 16 shows the flow rate curve through the by-pass loop under different steering wheel speeds of no-load condition with electromotor speed is 708 r/min. The by-pass loop flow rate is 0.66L/min, 3.15L/min, 6.03L/min when the steering wheel speed is 8r/min, 20r/min, 30r/min, respectively. Accounting for 16.5%, 18.6% and 19.1% of the total flow rate into the redirector, respectively. The simulation results prove that the flow rate through by-pass loop increases with the increase of steering wheel speed, but the increase rate decreases. The proportion of flow rate through by-pass loop is relatively stable.

Figure 17 shows the equivalent load inlet pressure simulation curves of the two redirectors under no-load and heavy-load conditions with electromotor speed is 708 r/min. Under no-load condition and the steering wheel speed is 20r/min, compared with the original redirector, the equivalent load inlet pressure overshoot of the redirector with by-pass damping reduces from 8.6bar to 7.9bar. Under heavy-load condition, the pressure overshoot reduces from 65.04bar to 56.35bar, with a decrease rate of 13.4%. The pressure oscillation duration is reduced from 2.09s to 1.6s, with a decrease rate of 23.4%. The research work in this paper can make a certain contribution to the stability optimization of articulated vehicle steering technology.

7 Conclusions

In this paper, to reduce the pressure oscillation and enhance the stability of the articulated vehicle steering system, a novel redirector with by-pass damping is proposed. The steering load is equivalent replaced to make the research conditions more convenient control. Through the simulation and experimental research and analysis of the novel redirector under different working conditions, the following conclusions can be concluded:

- (1) For the redirector with by-pass damping, the peak value of the equivalent load inlet pressure is reduced from 65.04bar to 56.35bar under heavy-load condition during steering. Compared with the original redirector, the overshoot is reduced by 13.4%, which reduces the number of system oscillations.
- (2) Compared with the original redirector, the output pressure oscillation duration caused by the load change of the redirector with by-pass damping is reduced by 23.4%. The pressure oscillation decays faster, and the

system is more stable.

- (3) For the redirector with by-pass damping, the flow rate through by-pass loop increases with the increase of steering wheel speed, but the increase rate decreases. The proportion of flow rate through by-pass loop is relatively stable. Therefore, the by-pass damping structure has little effect on the supply of flow rate during steering.

8 Declaration

Availability of data and materials

The datasets supporting the conclusions of this article are included within the article.

Competing interests

The authors declare no competing financial interests.

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Authors' contributions

The author's contributions are as follows: XH Liu, JS Chen conceived the idea of the study; YQ Wang, DY Huo performed the theoretical derivation and simulation experiment. FY contributed to structure design. YQ Wang analyzed the data and wrote the manuscript. All authors read and approved the final manuscript.

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Author Details

School of Mechanical and Aerospace Engineering, Jilin University, Jinlin 130022, China.

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Biographical notes

Yu-Qi Wang, born in 1993, is currently a PhD candidate at School of Mechanical and Aerospace Engineering, Jilin University, China. She received bachelor's degree from Jilin University, China, in 2016. Her research interests include fluid power transmission and control.

Xin-Hui Liu, born in 1962, is currently a professor at School of Mechanical and Aerospace Engineering, Jilin University, China.

Jin-Shi Chen, born in 1983, is currently an associate professor at School of Mechanical and Aerospace Engineering, Jilin University, China.
E-mail: spreading@jlu.edu.cn

Dong-Yang Huo, born in 1996, is currently a master candidate at School of Mechanical and Aerospace Engineering, Jilin University, China.

Figures

Figure 1

Schematic diagram of the redirector. (a) Structure of the redirector; (b) Structure of the rotary valve; (c) Principle diagram of the redirector.

Figure 2

Expanded view of control spool and sleeve. 1. Inlet module; 2. Flow rate amplification module; 3. Metering motor module; 4. Load sensor module; 5. Working module; 6. Return module; 7. By-pass module.

Figure 3

Relation between the opening area of each module and the control spool displacement

Figure 4

Schematic diagram of the equivalent damping

Figure 5

Simulation model of the novel redirector.

Figure 6

Displacement of control spool and sleeve.

Figure 7

Relation between speed signal and , .

Figure 8

Simulation results under no-load condition of electromotor speed is 708 r/min. (a) ; (b) ; (c) ; (d)

Figure 9

Simulation results under no-load condition of steering wheel speed is 20 r/min. (a) ; (b) ; (c) ; (d)

Figure 10

Simulation results under steering wheel speed 8r/min, electromotor speed is 708 r/min. (a) ; (b) ; (c) ; (d)

Figure 11

Photograph of the redirector experiment platform. 1. Steering wheel; 2. Pressure sensor; 3. Data acquisition; 4. Redirector; 5. Pump; 6. Electromotor; 7. Controller.

Figure 12

Experimental results under no-load condition of electromotor speed is 708 r/min. (a) ; (b) ; (c) ; (d)

Figure 13

Experimental results under no-load condition of steering wheel speed is 20 r/min. (a) ; (b) ; (c) ; (d)

Figure 14

Experimental results under steering wheel speed 8r/min, electromotor speed is 708 r/min. (a) ; (b) ; (c) ; (d)

Figure 15

The schematic of redirector mathematical model

Figure 16

Flow rate through by-pass orifices.

Figure 17

Pressure of the equivalent load inlet.