Advances in Machinery, Materials Science and Engineering Application
M. Chen et al. (Eds.)

doi:10.3233/ATDE220462

Design and Optimization of Cab Follow-Up Hydraulic System

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Abstract. A new type of cab follow-up hydraulic system and universal valve structure is designed to effectively reduce the design and manufacture cost of commercial vehicle's parts. In addition to satisfying cab overturning working conditions, the new valve also assumes the follow-up function of the hydraulic system instead of the hydraulic cylinder, and the spring parameters in the hydraulic valve can be adjusted according to different environments to improve the versatility of the valve. In order to achieve better performance, the maximum lift force is optimized using newly proposed multi-swarm PSO. In this paper, the cab overturning process model and objective optimization model are established with Matlab software, and the objective function is optimized and solved by improved multi-swarm PSO. The results show that the new cab follow-up hydraulic system and the new valve structure can meet the requirements of various working conditions of the cab well; the maximum lifting force during the cab overturning process was basically consistent with the experimental results; the improved multi-swarm PSO can solve the parameters with fast convergence, and the objective function value is reduced by 85%.

Keywords. Cab, follow-up system, hydraulic, valve, maximum lift force, improved multi-swarm PSO

1. Introduction

The connection of the commercial vehicle chassis and cab is realized by hydraulic system and cab suspension system. The hydraulic system is mainly used for chassis maintenance, etc [1]. The suspension system is used to reduce the bumps of the cab while driving, thus improving the ride comfort [2]. However, under driving conditions, the oil cylinder of the cab hydraulic system will have bad influences on movement of the cab, hence reducing the damping effect of the suspend system. Therefore, the cab follow-up system needs to have a follow-up function to minimize this effect. Currently on the market, this follow-up hydraulic function in the cab is realized by the hydraulic cylinder. The expansion cylinder structure is used by Xu [3,4] to achieve this follow-up function, which means the follower section at the bottom of the cylinder is expanded to get larger diameter. The structure is simple, but it will cause the piston rod to be eccentric, thereby causing wear and oil pollution, and making the cab falling process to be unstable at last. Zhang et al. adopted a straight cylinder with welded external steel pipes [5]. This structure will not cause partial wear of the piston rod, but it will also

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cause instability in the final stage of the descending process. The size of the hydraulic cylinder and the length of the follower section of the hydraulic cylinder are affected by the specific use, as well as by the assembly environment during manufacture [6,7].

The new cab hydraulic system adopts a new hydraulic system principle and integrates the follow-up function into the hydraulic valve structure, which simplifies the hydraulic cylinder structure and increases the versatility of hydraulic cylinder, thereby reducing the related cost. The new hydraulic valve with follow-up function is universal in the industry, and the oil pressure parameters of the hydraulic valve can be adjusted by springs.

2. The Schematic Diagram of the Hydraulic System

The new cab follow-up hydraulic system is shown in figure 1.The two-position-four-way reversing valve 2, the relief valve, and the oil pump are integrated together as an off-the-shelf oil pump product on the market. A' port and B' port belong to the oil pump. Between the oil cylinder and the hydraulic pump, there are two-position-three-way reversing valve 1, hydraulic control check valve 4 and throttle valve 3. A and B are the lower and upper chamber ports of the cylinder respectively. The technical requirement of the new follow-up hydraulic system is that the cylinder holding pressure is less than the opening pressure set by the hydraulic control check valve 4, when the cab stops at any position, thereby preventing the cab from sliding down.

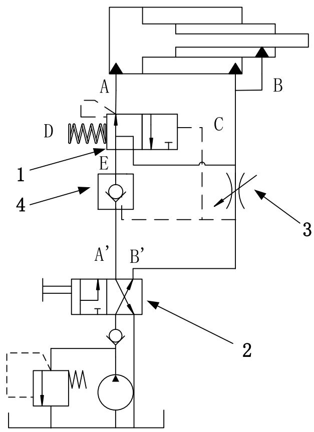


Figure 1. Follow-up hydraulic system.

3. Hydraulic Control Valve Structure Design

The structure diagram of the hydraulic control valve is shown in figure 2. The valve structure integrates valve 1, valve 4, valve 3 and oil pipes. The throttle valve 3 is substituted by throttling hole. The hydraulic valve consists of 9 kinds of parts: part 1 served as the left and right cover plates, part 2 used as the upper cover plate, and parts 3 and 9 both called spring, part 4 served as the push rod in the hydraulic control part of the two-position reversing valve, part 5 used as the valve core of the two-position reversing valve, part 6 served as the push rod in the hydraulic check valve, part 7 used as the core of hydraulic check valve, part 8 is the main body of the hydraulic valve, which is fixed .

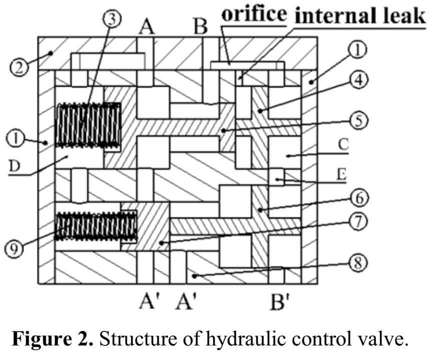


Figure 2. Structure of hydraulic control valve.

A' and B' are the external ports of the hydraulic valve, which are connected to the oil pump; A and B are the external ports of the hydraulic valve, which are connected to the oil cylinder; C and D oil chambers are used for hydraulic control.

In the hydraulic control part of the two-position reversing valve, the oil enters the C and D oil cavities, then pushes the two-position reversing valve for switching positions; the E port is the passage for the oil to flow through the B'B path; the springs can also realize the adjustment of the hydraulic control check valve spool's displacement.

4. Model and Method

4.1. Cab Maximum Lift Force Model

In order to find the maximum cylinder lifting force under various cab attitude positions, the cab lifting process is analyzed. The force diagram when the cab is turned over is shown in figure 3: The y-axis is in the horizontal direction, and the z-axis is in the vertical direction; O is the turning center of the cab; A is the lower fulcrum of the oil cylinder, and B is the upper fulcrum of the oil cylinder; the gravity center of the cab is C, and the weight of the cab is G; the lifting force of the oil cylinder is F_c, the overturning angle of the cab is θ.

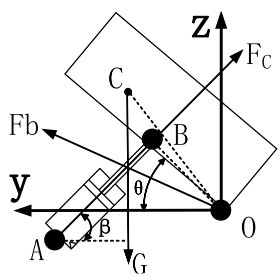


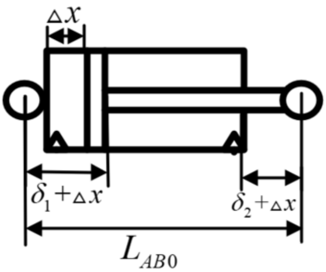
Figure 3. The force diagram when the cab is turned over.

When the cab is completely down (in a horizontal position), the angle between the OA line and the y-axis is α_A , the angle between the OB line and the y-axis is α_B , and the angle between the OC line and the y-axis is α_C .Define the length of line OA as L_A , the length of line OB as L_B , the length of line OC as L_c , the length of line AB as L_{AB} , and the initial length of line AB as L_{AB0} .Then F_c can be solved:

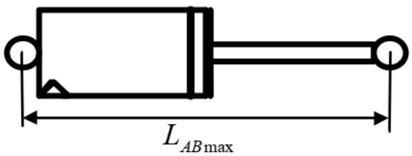
$$F_C = \frac{GL_c \cos(\alpha_c+\theta)\sqrt{{L_B}^2+{L_A}^2\text{-}2L_B L_A\cos(\alpha_B+\theta-\alpha_A)}}{L_B L_A\left|\sin(\alpha_B+\theta-\alpha_A)\right|} \tag{1}$$

This formula still holds when the cab is over the center of gravity, and is applicable to different layout of the cylinder [8]. When other parameters are known, the maximum force value can be obtained by using Matlab.

According to the example of heavy-duty truck [9], the cab adopts the differential lifting method. The initial position is defined when the cab is parked on ground. Before and after the cab is turned over, the position of the cylinder piston rod relative to the cylinder barrel is shown in figure 4.The displacement of the lower follower section of the piston rod is Δx . δ_1 and δ_2 are structure dimensions.



(a) Parking state before flipping



(b) Finished state of flipping

Figure 4. Piston rod position before and after cab rollover.

There are two types of cab suspension systems: full floating suspension system with both front and rear suspensions floating; semi-floating suspension system with fixed front suspension brackets and floating rear ones. Define the floating displacement of the cab suspension as s_1 , the distance between the front and rear suspension installation points of the cab as s_2 , the stroke of the oil cylinder as s_3 .

In the case of full floating suspension, according to the geometric relationship, we get:

$$L_{AB0}=\sqrt{(L_B\cos\alpha_B\text{-y}_A)^2+(L_B\sin\alpha_B\text{-z}_A)^2} \tag{2}$$

$$\Delta x=L_{AB0}\text{-}\sqrt{(L_B\cos\alpha_B\text{-y}_A)^2+(L_B\sin\alpha_B\text{-s}_1\text{-z}_A)^2} \tag{3}$$

Similarly, in the case of semi-floating suspension, according to the geometric relationship, we get:

$$\triangle x = L_{AB0} - \sqrt{(L_B \cos(\alpha_B + \arctan(-s_1/s_2)) - y_A)^2 + (L_B \sin(\alpha_B + \arctan(-s_1/s_2)) - z_A)^2} \tag{4}$$

4.2. Optimization Objectives and Mutation Particle Swarm Optimization

By optimizing the A and B positions, the maximum lift force can be optimally reduced. The objective function to be optimized is showed as follows:

$$\arg\min J(L_A, L_B, \alpha_A, \alpha_B) = Fc_{\max} + \left(L_{AB0} - \delta_1 - \delta_2 - 2\triangle x - s_3\right)^4 \tag{5}$$

Since the objective function is a nonlinear function, the analytical solution cannot be obtained, so the global optimization algorithm of multi-swarm PSO [10] is adopted, which includes the following basic formula:

$$v_{i+1} = w \cdot v_i + c1 \cdot rand \cdot (pbest_i - x_i) + c2 \cdot rand \cdot (gbest_i - x_i) \tag{6}$$

$$x_{i+1} = x_i + v_{i+1} \tag{7}$$

In formula (6), x is the input parameter; w is the current speed factor; $c1$ and $c2$ are learning factors; $rand$ is a random number; $pbest_i$ is the current individual optimal value, and $gbest_i$ is the history individual optimal value; v_i and v_{i+1} are the particle movement speed at the current moment and next moment respectively; x_i is the current particle position, and x_{i+1} is the particle position at the next moment.

In order to achieve the optimization results more quickly, an improved multi-swarm particle optimization is proposed in this article. In the iterative process, the objective function values corresponding to all input particles in the current particle swarms are calculated. Gaussian noise, which is realized by using Matlab function awgn, is added to each particle, in order to jump out of the local optimal solution, when the variance of the function values of all swarms is less than the given value. In order to jump out of the local convergence better, the whole particle swarms are divided into four categories according to the degree of added noise, namely x^1, x^2, x^3, x^4:

$$\begin{cases} x^1 = x^1 \times \mathrm{awgn}(\mathrm{size}(x^1), 0.1, '\mathrm{measured}') \\ x^2 = x^2 \times \mathrm{awgn}(\mathrm{size}(x^2), 1, '\mathrm{measured}') \\ x^3 = x^3 \times \mathrm{awgn}(\mathrm{size}(x^3), 10, '\mathrm{measured}') \\ x^4 = x^4 \times \mathrm{awgn}(\mathrm{size}(x^4), 20, '\mathrm{measured}') \end{cases} \tag{8}$$

5. Results and Discussions

5.1. Experimental Validation

According to the example of heavy-duty truck with full floating suspension [9] and formula (1), the maximum lifting force simulated by Matlab is 19112N, which is converted into an oil pressure of 19.9MPa, which is basically consistent with the maximum oil pressure 18.9MPa in the lifting condition measured in the experiment. The changing value of the lifting force related to the flip angle is shown in figure 5:

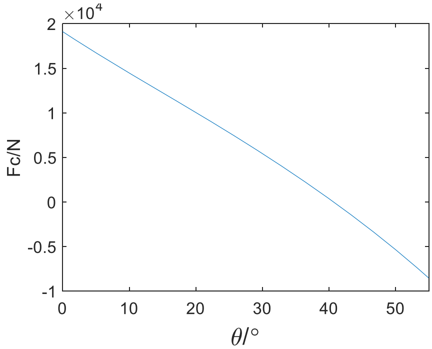


Figure 5. Maximum lift force during cab overturning.

5.2. Experimental Validation

According to formulas (6) and (7), the objective function can be optimized, taking $L_A \in [0,4L_c]$, $L_B \in [0,2L_c]$, $\alpha_A \in [-\alpha_C, \alpha_C]$, $\alpha_B \in [-\pi/6, \pi/2]$ and limiting the parameters range. During the simulation, the number of particle swarms is 12 (that is, 3 particle swarms per group), and the number of particles in each swarm is 4 (that means 4 input parameters). The variance of all particle swarms' objective values for criterion is 5. The speed factor is 0.3, and the learning factor is 1.6. The simulation results are shown in figure 6. The abscissa shows the number of iterations, and the ordinate shows the objective function value. The results show that the objective function value can quickly reach the global optimal value.

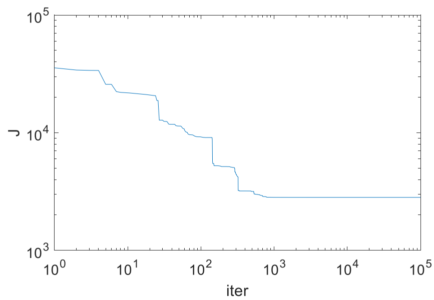


Figure 6. Iterative diagram of maximum lifting force optimization.

After optimization, the parameters are optimized as 3440, 2691, -0.3552, and 0.4535. The maximum lifting force is 2820N, which is reduced by 85% comparing to the original value in the above example.

6. Conclusion

The new cab follow-up hydraulic system meets the requirements of various working conditions of the commercial vehicle cab, and the hydraulic control valve structure designed according to the hydraulic system can satisfy work requirements. The simulation results of Matlab software show that the maximum lift force during the cab overturning process is basically consistent with the experimental results; the improved multi-swarm PSO can achieve rapid convergence, and the optimization result of the objective function value is reduced by 85%.

Acknowledgement

The Scientific Research Program Guidance Project B2021409 of Hubei Provincial Department of Education

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