Hydraulic system elaboration and simulation for single-drive light-load monorail locomotive in fully mechanized coal mining applications

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Abstract. To solve the problem of light-load material transportation at the top and bottom of fully mechanized mining face, a hydraulic control system for single-drive monorail light-duty locomotive was designed. The importance of proper clamping force is mainly introduced, and the clamping pressure and braking time are determined. The relationships between the average travelling speed of the locomotive under different loads, the load and slope path of the locomotive are obtained. The simulation path was designed according to the field operation, and the AMESIM software was used for modeling and simulation. The simulation results meet the technical and engineering requirements and briefly introduce the use of the rotary valve in the traction circuit and the clamping circuit.

1. Introduction

In fully mechanized coal mining applications, the transportation of materials and equipment, at present, is generally ensured by pneumatic or electric monorail cranes with explosion-proof motor and explosion-proof switch power supply. The wire travels along the guide rail along the locomotive, and the distance of the excavation is the driving distance of the locomotive. It is symmetrically arranged on the guide rail + reducer + drive wheel by the explosion-proof motor, and the driving mode of the explosion-proof component such as the brake power failure is realized by the disc motor. The heavy-duty Scharf Monorail System (Germany) uses a low-speed, high-torque motor controlled by a variable displacement pump, which applies an electro-hydraulic valve to achieve braking [1-2].

The hydraulic single-drive monorail locomotive introduced in this paper is illustrated in figure 1. It is aimed to solve the problem of light-duty transportation in underground coal mines. Its braking and clamping are controlled by the hydraulic pressure. Only the motor needs the protection from explosion hazard, insofar as it has to run in the inflammable gas environment. It is especially suitable for mines with a severe bottom heave and poor floor conditions. The maximum lifting weight of the locomotive is 5 tons. It can move forward and backward along the suspension guide and needs only one operator to fulfil the task that previously required 3 or 4 ones. The alleviation of labor intensity of operators improves the production efficiency.

This paper starts from the light-load transportation law of a single-drive monorail crane. The hydraulic system of the hydraulic control system has the advantages of the existing heavy-duty transportation monorail transportation main circuit and safety auxiliary circuit, travelling, clamping, braking, traction, etc. Full hydraulic controls the manual and electric hoist lifts. At present, the design
requirements for such technical indicators as virtual and physical prototypes have been satisfied and verified via the underground industrial tests.

Figure 1. The single-drive monorail crane hydraulic virtual prototype.

2. Transport main circuit

2.1. Clamping circuit

The hydraulic clamping circuit consists of an inlet check valve, a pressure reducing valve, a rotary clamping valve, and a clamping cylinder. As shown in figure 2, the function of the check valve is to complete the first clamping. The check valve and the clamping cylinder chamber will close the pressure of the hydraulic medium between the two. Even if the machine is shutdown, it can connect the drive wheel to the guide rail. The control of the board, so that whenever the start, the brake cylinder is supplied with pressure oil to improve the brake release time and ensure the safety of the locomotive. When the driving portion is detached from the guide rail, the rotary valve can be placed in the position of the switch, and the liquid in the clamping cylinder enters the oil tank to release the clamping [3].

Figure 2. The single driving monorail crane control system.

According to the design requirements, the traction value is 5kN. The periphery of the drive wheel is made of rubber. The friction coefficient with the steel guide rail is 0.25. The positive pressure of the drive wheel and the guide rail is:

\[ F_J = \frac{5}{0.25} = 20 \text{kN} \]

The force is generated by the rear pump pressure liquid of the double gear pump acting on the rod cavity of the clamping cylinder to produce a pulling force on the driving wheel. The set pressure of the pressure reducing valve is:

\[ p_j \geq \frac{4F_j}{\pi(D_j^2 - d_j^2)} \]  

(1)
where \( N_j \) is driving wheel clamping force (N); \( D_j \) is clamping cylinder piston diameter (designed \( D_j = 60 \text{mm} \)); \( d_j \) is piston rod diameter of clamping cylinder (designed \( d_j = 36 \text{mm} \)); \( p_j \) is setting pressure after the pump overflow valve (N/m\(^2\)). The calculation of setting pressure of the pressure relief valve yields: \( p_j \geq 1.98 \text{MPa} \).

According to reference GB2346-80 on the outlet pressure of the pressure-reducing valve circle, the value \( p_j = 2.5 \text{MPa} \). Therefore, the size of the clamping force to meet the design requirements is:

\[
F_j = \frac{\pi}{4} (60^2 - 36^2) \times 2.5 = 252.8 \text{kN} > 200 \text{kN}
\]

### 2.2. Travelling circuit.

The travelling circuit is composed of a manual hydraulic control reversing valve (manual pilot valve and main hydraulic valve), a two-way balancing valve and a hydraulic motor, which operate the manual pilot valves located on both sides of the locomotive. The control fluid from the rear pump acts on one side of the main pilot valve. Pushing the main spool to move [4-6], the oil generated by the front pump controls the direction of rotation of the hydraulic motor. The actual flow output of the pump before the double pump is:

\[
q_B = V_B \eta \nu
\]

where \( q_B \) is the actual flow of the output of the hydraulic pump (m\(^3\)/s); \( \nu \) is motor speed (n = 1450 rev/min); \( V_B \) is displacement of the front pump (\( V_B = 10 \times 10^{-6} \text{ m}^3/\text{ r} \)); \( \eta \) is volumetric efficiency (\( \eta = 92\% \)).

The output of the hydraulic pump in the front part of the double pump drives a pair of motors. Assuming that the flow of both engines is equal, the speed of each motor can be calculated as follows:

\[
n_u = \frac{V_M \eta \nu}{2V_M}
\]

where \( V_M \) is motor displacement. In the case under study, \( V_M = 470 \times 10^{-6} \text{ m}^3/\text{ r} \).

The average running speed is related to the motor axial driving wheel diameter \( D_o \) as follows:

\[
v = D_o \nu n_u
\]

From Eq.4, we get \( D_o = 355 \times 10^{-3} \text{ m} \).

The data input into formula (4) yields the average running speed of the locomotive:

\[v = 0.2638 \text{(m/s)}\]

Assuming that the friction coefficient of bearing trolley’s travelling wheel and rail is \( \mu \), the traction force required for a smooth road is:

\[
F_p = mg \mu
\]

where \( m \) is the total weight, including of the dead weight (kg); \( F_p \) is traction force of flat road surface \( (F_p = 10 \text{kN}) \); \( \mu \) is the friction coefficient of the travelling wheel and track. In our case, \( \mu = 0.2 \).

In the upward motion, the necessary traction for the total weight \( m \) is:

\[
F_u = mg (\sin \theta + \mu \cos \theta)
\]

where \( F_u \) is maximum traction load required for upward motion (N); \( \theta \) is ramp inclination angle (the
maximum value is 25º).

The single driving monorail crane load can be measured by the car weight testing tools: its value plus locomotive weight 1218 kg yields the total weight of the single driving monorail crane \( m [7-9] \).

When \( \theta \) changes from 0º to 25º, the locomotive moves uphill, lifting the weight \( m \) under different slope values, the relationship is shown in figure 3: the higher the slope, the smaller the lifting weight.

When the locomotive moves downhill, the driving force is as follows:

\[
F_d = mg (\sin \theta - \mu \cos \theta),
\]

where \( F_d \) is downhill required traction load, N.

Above each action on the driving wheel, the product of the traction force and the radius of the driving wheel is torque acting on the motor shaft.

\[
T = F \times R,
\]

where \( R \) is motor drive wheel radius. In our case, \( R=0.1775 \) m.

\[ T = 2500 \times 10 \times 0.2 \times 0.1775 / 2 = 443.75 (N.m) \]

\[ T_u = 2500 \times 10 \times 0.1775 \times (\sin(10^\circ) + 0.2 \times \cos(10^\circ)) / 2 \]

\[ = 2518.725 (N.m) \]

2.3. Travelling circuit simulation.

As an example of operation condition, we used the Anju Coalmine (China) with a slope of 10º. The load was set at 2.50t. For the design of a round-trip transportation mission, AMESIM system was used to carry out simulation tests.

2.3.1. Path planning.

**Forward motion.** The required steps are as follows. Use the hand hoist to load the transported materials - pull roadway side manual pilot valve handle to the left position - smooth road - 10º upward slope – put the manual pilot valve handle to the middle position - ready to unload the transported materials [10].

For the smooth-road torque acting on a single motor shaft, formulas (5) and (8) yield:

\[ T_p = 2500 \times 10 \times 0.2 \times 0.1775 / 2 = 443.75 (N.m) \]

For the upward slope of 10 degrees, the torque acting on a single motor shaft is also derived via equations (6) and (8):

\[ T_u = 2500 \times 10 \times 0.1775 \times (\sin(10^\circ) + 0.2 \times \cos(10^\circ)) / 2 \]

\[ = 2518.725 (N.m) \]

**Return trip.** The required steps are as follows. Use the hand hoist to unload the transported materials, push roadway side manual pilot valve handle to the right position - smooth road - 10º downward slope - put the manual pilot valve handle to the middle position - ready to load the transported materials.

For the downward slope of 10 degrees, the torque acting on a single motor shaft can be derived via formulas (6) and (8) as follows:
For the time period of 0~4s, the load on the flat surface of the hydraulic motor is shown in figure 4 (b). According to formula (9), during the time period of 4~8s, the hydraulic motor is subjected to the uphill load; at 8~12s, the hydraulic motor does not rotate. According to formula (10), at 12~16 s, the hydraulic motor is subjected to downhill load; at 16~20s, the hydraulic motor does not rotate.

2.3.2. Configuration scheme.

As shown in figure 5, the hydraulic circuit of AMESIM is simplified to scheme, in which the motor control is used to convert the mechanical energy of the main circulation pump into a hydraulic energy device. The motor rotation speed is set to 1450 rpm, and the displacement of the hydraulic pump is configured as follows. The pressure is controlled by a relief valve with a pressure set at 16 MPa. A three-position four-way valve with an O-type neutral function is selected as the hydraulic control main valve. There are two balancing valves between the hydraulic motor and the hydraulic control main valve, which are used to prevent the “landslide” of the slope during travelling and ensure a soft start, smooth running, soft and reliable stop at the required position on the slope. The displacement of the hydraulic motor is $470 \times 10^{-6} m^3 / s$, the rotary inertia is $34153 kg \cdot mm^2$. The reversing valve input signal I is set as shown in figure 4 (a), and the extra torque load input signals II and III are arranged as shown in figure 4 (b). After running AMESIM, the output signal provides the torque, angular velocity, differential pressure, and flow to form a resulting data file.

2.3.3. Simulation analysis.

Using MATLAB software to process the data of the simulation result file, after the data processing of the simulation results file, the program constructed characteristic curve shown in figure 6. From the simulation characteristic curve, one can see the following patterns.

$T_{dh} = 1218 \times 10 \times 0.1775 \times (\sin(10^\circ) - 0.2 \times \cos(10^\circ)) / 2$

$= -248.5(N.m)$

1) The differential pressure between the two ends of the hydraulic motor is related to the load by the following formula:

$$\Delta p = \frac{2\pi T}{V_M \eta_M}$$
where \(\Delta p\) is motor differential pressure, N/m\(^2\); \(\eta_M\) is motor mechanical efficiency. In our case, \(\eta_v = 0.9\).

The comparison of figures 4 (b) and 6 (a) shows that both characteristic curves are consistent, that is, the pressure is proportional to the load applied to the motor.

2) The locomotive running speed is inversely proportional to the torque load, so the speed and traction force satisfy the following formula:

\[
v = \frac{\Delta p q \eta_v \eta_M}{F}
\]

Using formula (8), we get:

\[
v = \frac{\Delta p q \eta_v \eta_M R}{T}
\]

Comparing Fig. 4 (b) with Fig. 6 (b), the rotational speed is inversely proportional to the torque load. At the time period of 4~8s, the torque load applied to the motor is the largest, and the rotational speed is the lowest; at 12~16s we get the downhill portion of the pass, with a high speed and easy stall.

The above findings prove that the simulation results are consistent with the theory of hydraulic transmission.

3. Safety auxiliary circuit

3.1. Working brake

The condition for the locomotive working brake is that the cartridge valve of the external emergency braking module in figure 2 is closed [11].

Regardless of the throttling effect and drag loss of the throttle valve, the brake action of the locomotive is equivalent to the hydraulic resistance of the orifice and the two throttle valves connected in series, according to the principle of flow continuity:

\[
C_d A_T \sqrt{\frac{2 p_X}{\rho}} = \frac{\pi d_0^4 (p_b - p_X)}{128 \mu l_0}
\]

where \(A_T\) is flow area of the throttle valve, m\(^2\); \(l_0\) is the length of the damping hole, m; \(d_0\) is the damping hole diameter, m; \(p_X\) is the cartridge valve control cavity pressure, MPa; \(p_b\) is effect of pressure on the surface of the ring cavity B, at the beginning of the brake \(p_b = 10 \text{MPa}\); \(\rho\) is the oil liquid density, kg/m\(^3\); \(\mu\) is the dynamic viscosity of the oil, N.s/m\(^2\).

The above formula can be reduced to the following form:

\[
A_T = \frac{\pi d_0^4 (p_b - p_X)}{128 \mu l_0 C_d} \sqrt{\frac{\rho}{2 p_X}}
\]

The last equation analysis yields several patterns:

(1) \(p_X = p_b\), \(A_T = 0\); and at the same time \(w p_X \rightarrow \infty\), \(p_X\) is adjusted by the overflow valve, indicating that the throttle valve at this time is closed. Thus, both ends of the damping hole are closed to the liquid flow. For a stationary fluid, the pressure value is constant, port A is not connected to B port, and the cartridge valve closes.

(2) \(p_X = 0\), \(A_T \rightarrow \infty\), the throttle valve and overflow channel are open. In the hydraulic control valve spring position, the cartridge valve is opened, ports A and B are connected, the locomotive implements the emergency brake. In the hydraulic control valve liquid control position, the cartridge
valve is closed.

3.2. Emergency brake

The centrifugal brake works mechanically and is integrated into the drive section of the monorail crane, consisting of a drive wheel and a mechanical structural unit with a centrifugal counterweight and a trigger pin. When the load is moved downward, it can be seen from figure 6(b) that when the empty car runs downward, the speed is high. When it is loaded is extremely prone to stalling, when the speed reaches the speed of the trigger of the factory settings (such as \( v=1 \text{m/s} \)), the trigger pin will hit the trigger control rocker, shifting the machine control valve (see figure 2), from the motor position to spring position, the oil under pressure moves back to the tank. The hydraulic control valve is shifted to the spring position, the liquid flow occurs by \( A>T \), \( p_X = 0 \).

When there is no liquid flow in the damping hole of the external emergency brake module, ignore the spool weight, and viscous friction, fluid statics analysis spool force of the two-way cartridge valve is:

\[
F_B = F_X + F_y + F_k
\]

where \( F_B \) is the force on the acting face of the ring cavity \( B \), \( F_B = \frac{\pi}{4} (D^2 - d^2) p_x \), \( N \); \( D \) is the diameter of the cartridge valve spool, \( m \); \( d \) is the control chamber of \( A \) cavity hole diameter, \( m \); \( p_B \) is pressure on the acting face of the ring cavity \( B \), \( N/m^2 \); \( F_X \) is the downward force on the acting face of the ring cavity, \( F_X = \frac{\pi}{4} D^2 p_x \), \( N \); \( F_k \) is cartridge valve reset stretch, \( F_k = k(x_0 + x) \), \( N \); \( k \) is the reset spring stiffness, \( N/m \); \( x_0 \) is initial compression of the reset spring, \( m \).

Steady-state fluid dynamics can be interpreted as the reaction force of the cartridge valve spool pointing in the direction, in which the valve port closes as liquid enters and exits the valve cavity due to changes in the direction of fluid flow.

The calculation formula for the steady-state flow force [12] is as follows:

\[
F_y = \rho q v \cos \alpha
\]

where \( \rho \) is the oil liquid density, \( \text{kg/m}^3 \); \( \alpha \) is the liquid jet angle, \( \alpha = 69^\circ \); \( q \) is flow through the valve port of the cartridge valve, \( \text{m}^3/\text{s} \);

\[
q = C_d \pi (D - d)x \sqrt{\frac{2(p_B - p_X)}{\rho}} \quad (11) \quad v = C_v \sqrt{\frac{2(p_B - p_X)}{\rho}}
\]

After substitution of equation (11) into (10), we get:

\[
F_y = 2C_d C_v \pi (D - d) \cos 69^\circ (p_B - p_X) x
\]

After re-arrangement, formula (9) can be reduced to the following form:

\[
p_B = \frac{(\pi D^2 - 42C_d C_v \pi (D - d) \cos 69^\circ x) p_X + 4k(x_0 + x)}{\pi (D^2 - d^2) - 42C_d C_v \pi (D - d) \cos 69^\circ x}
\]

The results obtained show that the opening and closing of the two-way cartridge valve depends on the \( X \) chamber pressure \( p_X \) and the reset spring stiffness of the cartridge valve, the size of the pressure \( p_X \) between the damping hole and the throttle valve related to the setting of the flow area of the throttle.
valve. During the centrifugal trigger action, the upper type can be reduced to the following form:

$$p_a = \frac{4k(x_0 + x)}{\pi(D^2 - d^2) - 42C_dC_c\pi(D - d)\cos 69^\circ x} \quad (14)$$

The force acting on the annular area of the cartridge valve is much larger than the return spring force of the cartridge valve. Therefore, the force balance is broken, the brake cylinder is subjected to high- and low-pressure variations, and the cylinder chamber is filled due to the expansion of the cylinder volume of the brake cylinder. In the piston chamber cycle, the oil return route is short, and the braking time is also shortened accordingly.

4. Conclusions

The main contents of this paper are summarized as follows:

✓ The relationship between the average running speed of the locomotive, slope, and the load under different working conditions are obtained. The simulation path is based on the engineering design requirements. The simulation results using the AMESIM software show that the operation of the hydraulic motor between the two end points is determined by the load. The running speed is inversely proportional to the traction force, which shows a high consistency of simulation results and the hydraulic drive theory.

✓ This paper describes the conditions for realizing the brake operation, deriving the expression of the flow passage area of the throttle valve, and analyzing the practical significance of adjusting the throttle valve. The clamp circuit of the check valve ensures the priority of the brake release after the motor rotation.

✓ The traction and clamping circuit is controlled by a rotary valve, closed traction rotary valve, manual operating pump, and brake release control. The opening of the rotary clamping valve can reduce the clamping force of the drive wheel and the guide rail.

✓ The hydraulic system analysis performed in this paper can be extended to the multi-drive or scheduling monorail cranes.

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