Parameter Effects of the Hydrostatic Propulsion Drive System on the Operation Accuracy of a Tamping Machine

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Abstract. A dynamic mathematical model of the hydrostatic propulsion drive system of a tamping machine is established in this study, and a simulation model is built in the AMEsim software environment using the mathematical model. Parameter effects of the hydrostatic propulsion drive system on the operation accuracy of the tamping machine are analysed. Simulation results show that the effective moment of inertia of the powertrain and the oil volumetric elastic modulus have obvious influences on the operation accuracy of the tamping machine, the effect of the total volume of the high-pressure circuit of the hydraulic system on the displacement accuracy is not obvious, but it has some influence on the velocity amplitude. The models established and results obtained in this work are of significance and instructive for design optimization and maintenance of tamping machines.

1. Introduction

Fluid power drives are widely used in modern industrial apparatus [1] and railway construction and maintenance machineries [2,3]. In a railway tamping machine [3,4], the hydrostatic propulsion drive system [5,6] plays an important role in the machine’s operation accuracy, which includes the velocity stability and displacement accuracy.

Yan proposed a hydraulic drive system to obtain [7] both high speed for mobility and low speed for tamping, Wang et al. studied different acceleration modes [8] for a hydraulic engineering vehicle by using a hydraulic proportional pump and a hydraulic variable motor. Shi investigated the problems of out of step [9] in velocity and traction force of a tamping machine, Hu et al. improved the pressure stability [10] of the hydraulic clamping system of a tamping machine, by adding a resistance orifice in the circuit, and Zeng et al. [11] and Zhai et al. [12] respectively investigated the adjustment and maintenance technology for the hydraulic system of their tamping machines.

A dynamic mathematical model of the hydrostatic propulsion drive system of a tamping machine is established in this study, and a simulation model is built in the AMEsim software environment based on the mathematical model. Parameter effects of the hydrostatic propulsion drive system on the operation accuracy of the tamping machine are analysed, simulation results show that the effective moment of inertia of the powertrain, the total volume of the high-pressure circuit of the hydraulic system and the oil volumetric elastic modulus have obvious influences on the operation accuracy of the tamping machine. The models established and results obtained in this work will be instructive for tamping machine design and maintenance.
2. Mathematical model of the hydrostatic propulsion drive system

Figure 1(a) shows that the tamping machine is driven by two hydraulic motors, and Figure 1(b) illustrates that the hydrostatic propulsion drive system of the tamping machine is a typical closed-type hydraulic system employing a two-way variable displacement pump and a two-way constant displacement motor. Varying the displacement of the hydraulic pump will lead to running speed change of the tamping machine, and alternating the rotation direction of the hydraulic pump will lead to moving direction alternation of the tamping machine.

![Mechanism and hydraulic power circuit](image_url)

**Figure 1.** Mechanism (a) and hydraulic power circuit (b) of the hydrostatic propulsion drive system of a tamping machine (1. diesel engine, 2. variable pump, 3. fluid compensation pump, 4. check valve, 5. high-pressure relief valve, 6. fluid compensation relief valve, 7. flush valve, 8. back-pressure valve, 9. hydraulic motor, 10. load, 11. reservoir).

Referring to Figure 1(b), the flow continuity equation of the variable pump is formulated by

\[ Q_b = K_b \omega_b \varphi_b - C_{ib} \left( p_1 - p_0 \right) - C_{eb} p_1 \]  

(1)

where \( Q_b \), \( K_b \), \( \omega_b \), \( \varphi_b \), \( C_{ib} \) and \( C_{eb} \) are the output flow, the rotation speed, the swashplate angle, the inner leakage coefficient and the outer leakage coefficient of the pump, respectively; \( p_1 \) and \( p_2 \) are pressures of the high-pressure chamber and the back-pressure chamber of the pump. Therefore, taking the Laplace Transform of Equation (1) to obtain

\[ Q_b(s) = K_b \omega_b \varphi_b(s) - C_b p (s) \]  

(2)

The flow continuity equation of the high-pressure circuit of the hydrostatic propulsion drive system can be expressed by

\[ Q_b - C_{im} \left( p_1 - p_0 \right) - C_{em} p_1 = D_m \omega_m + \frac{V_0}{\beta_e} \frac{dp_1}{dt} \]  

(3)

where \( C_{im} \), \( C_{em} \), \( D_m \) and \( \omega_m \) are the inner leakage coefficient, the outer leakage coefficient, the displacement and the shaft rotation speed of the hydraulic motor, respectively; \( V_0 \) is the total volume of the high-pressure circuit of the hydraulic system, and \( \beta_e \) is volumetric elastic modulus of the oil. Similarly, taking the Laplace Transform of Equation (3) to obtain

\[ Q_b(s) - C_m p(s) = D_m \omega_m (s) + \frac{V_0}{\beta_e} s p (s) \]  

(4)

where \( C_m \) is the total leakage coefficient of the hydraulic motor, and \( C_m = C_{im} + C_{em} \).

Thus, combining Equations (2) and (4) will have

\[ K_b \omega_b \varphi_b (s) = D_m \omega_m (s) + \left( C_i + \frac{V_0}{\beta_e} s \right) p (s) \]  

(5)

where \( C_i \) is the total leakage coefficient of the hydraulic system, \( C_i = C_b + C_m \) and \( C_b = C_{ib} + C_{eb} \).
The torque equilibrium equation of the hydraulic motor and load can be described as

\[ D_m (p_1 - p_0) = J_e \frac{d\omega_m}{dt} + B\omega_m + T_L \]  

(6)

where \( J_e \) is the effective moment of inertia of the powertrain, \( B \) is the viscous damping coefficient, and \( T_L \) is the load torque. If neglecting the air drag resistance, the load torque can be formulated by

\[ T_L = r \sum F = r \left[ mg \cos \beta (f_0 + kv) + mg \tan \beta + m\delta \frac{dv}{dt} \right] \]  

(7)

where \( r, \sum F \) and \( m \) are the wheel radius, the total resistance force and the mass of the tamping machine, \((f_0+kv)\) is the coefficient of rolling friction of the wheels, \( \beta \) is the slope angle, and \( \delta \) is an effective coefficient of the rotation mass of the tamping machine.

Thus, considering the vehicle speed \( v = r\omega_m \), so combining Equations (6) and (7), and taking the Laplace Transform to obtain

\[ D_m p(s) = \frac{J_e + B}{r} v(s) + (m\delta rs + mg \cos \beta r k) v(s) \]  

(8)

Thus, if in terms of the swashplate angle of the pump \( \phi_h \) as the input signal and vehicle speed \( v \) as the output signal, combing Equations (5) and (8) to obtain the transfer function of the hydraulic system, as

\[ G(s) = \frac{\frac{v(s)}{\phi_h(s)}}{s^2 + \frac{rK_c \beta_s D_m}{\beta_e} + \frac{V_0 (J_e + m\delta r^2)}{\beta_e} + \frac{V_0 (B + mgr^2 \cos \beta)}{\beta_e} + C_s} \]  

(9)

In addition, if in terms of the load torque \( T_L \) as the input signal and vehicle speed \( v \) as the output signal, the transfer function of the hydraulic system can be described as

\[ G_L(s) = \frac{v(s)}{T_L(s)} = \frac{1}{m\delta rs + mg \cos \beta r k} \]  

(10)

Thus, based on the above mathematical modelling of the hydrostatic propulsion drive system, the dynamic control block diagram of the hydraulic circuit can be obtained and demonstrated by Figure 2.

![Figure 2. Dynamic control block diagram of the hydrostatic propulsion drive system.](image)

3. Simulation model

Referring to Figure 1(a) and the mathematical model established in Section 2, a simulation model of the hydrostatic propulsion drive system of the tamping machine is built in the AMEsim software environment, and the simulation model is illustrated by Figure 3.

Simulation of the parameter effects of the hydrostatic propulsion drive system on the operation accuracy of the tamping machine is performed, the parameters and values used in the simulation are summarized in Table 1.
Figure 3. Simulation model of the hydrostatic propulsion drive system in the AMEsim environment.

Table 1. Parameters and values used in the simulation

| Parameter                                         | Value       | Parameter                                         | Value   |
|---------------------------------------------------|-------------|---------------------------------------------------|---------|
| Rotation speed of engine (r/min)                  | 2300        | Gear ratio of gear box                            | 4.73    |
| Maximum displacement of variable pump (mL/r)      | 125         | Full laden mass of tamping machine (Kg)           | 60000   |
| Displacement of fluid compensation pump (mL/r)    | 28.3        | Wheel radius (m)                                  | 0.42    |
| Set pressure of main relief valve (MPa)           | 40          | Rotation damping coefficient                      | 0.039   |
| Set pressure of fluid compensation relief valve   | 2.5         | Braking damping coefficient                       | 0.05    |
| Relief pressure of flush valve (MPa)              | 2           | Volumetric efficiency of hydraulic pump and motor | 0.95    |
| Displacement of hydraulic motor (mL/r)            | 500         | Mechanical efficiency of wheel gear box           | 0.92    |

4. Parameter effects analysis

Figure 4 and Table 2 demonstrate the effect of the effective moment of inertia of the powertrain $J_e$ on the operation accuracy of the tamping machine, and shows that $J_e$ has an obvious influence on displacement accuracy of the tamping machine. For instance, if with a bigger $J_e=12$ kgm$^2$, the tamping machine will have a relative error of 3.78% on the displacement of the 1st cycle and a relative error of 5.84% on the velocity amplitude of the 1st cycle.

Figure 4. Effects of effective moment of inertia of the powertrain $J_e$ on (a) the displacement accuracy and (b) the velocity stability of the tamping machine.
Table 2. The Effect of $J_e$ on operation accuracy of the tamping machine

| Operation Accuracy Indices | $J_e=7 \text{ kgm}^2$ (Standard) | $J_e=5 \text{ kgm}^2$ | $J_e=12 \text{ kgm}^2$ |
|----------------------------|---------------------------------|---------------------|---------------------|
| Displacement of the 1$^{\text{st}}$ cycle (m) | 1.086 | 1.108 (Relative error 2.03%) | 1.045 |
| Displacement of the 2$^{\text{nd}}$ cycle (m) | 2.174 | 2.218 (Relative error 2.02%) | 2.092 |
| Displacement of the 3$^{\text{rd}}$ cycle (m) | 3.262 | 3.328 (Relative error 2.02%) | 3.140 |
| Velocity amplitude of the 1$^{\text{st}}$ cycle (m/s) | 0.994 | 1.033 (Relative error 3.92%) | 0.936 |

Figure 5. Effects of $J_e$ on acceleration performance of the tamping machine.

Figure 6. Effects of the total volume of the high-pressure circuit of the hydraulic system $V_0$ on (a) the displacement accuracy and (b) the velocity stability of the tamping machine.

Figure 7. Effects of the oil volumetric elastic modulus $\beta_e$ on (a) the displacement accuracy and (b) the velocity stability of the tamping machine.
Figure 5 also shows that the acceleration performance of the tamping machine will be obviously affected by the effective moment of inertia of the powertrain.

Figure 6 shows that the effect of the total volume of the high-pressure circuit of the hydraulic system $V_0$ on the displacement accuracy is not obvious, but $V_0$ has some influence on the velocity amplitude. However, Figure 7 shows that the oil volumetric elastic modulus $\beta_e$ has obvious effects on both of the displacement accuracy and velocity amplitude.

Generally speaking, a smaller $J_e$ and $V_0$ are expected in the engineering development of a tamping machine, while $\beta_e$ is easily affected by the entrapped air ratio in the oil, so a bigger $\beta_e$ is expected in the operation and maintenance of the tamping machine.

5. Conclusions

- The effective moment of inertia of the powertrain $J_e$ has an obvious influence on operation accuracy of the tamping machine, a smaller $J_e$ will improve the response and operation accuracy of the tamping machine, and can be pursued by means of lightweight designs.
- The effect of the total volume of the high-pressure circuit of the hydraulic system $V_0$ on the displacement accuracy is not obvious, but $V_0$ has some influence on the velocity amplitude. A smaller $V_0$ will improve the response and operation accuracy of the tamping machine.
- The oil volumetric elastic modulus $\beta_e$ has obvious effects on both of the displacement accuracy and velocity amplitude. $\beta_e$ is easily affected by the entrapped air ratio in the oil, so a bigger $\beta_e$ is expected in the operation and maintenance of the tamping machine.
- The mathematical model established and results obtained in this work will be instructive for further tamping machine design optimization and maintenance.

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