High-Speed Performance of a Tamping Machine with Closed-Type Hydrostatic Propulsion Drive

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Abstract. It is significant to understand and optimize the parameters of the hydraulic system to pursue both better low-speed operation and high-speed mobility performance of a tamping machine. A mathematical model formulating the high-speed performance of a tamping machine with a closed-type hydrostatic propulsion drive system is established in this study, a simulation model is built in the AMEsim software environment based on the mathematical model. Simulation results show that the tamping machine with a closed-type hydrostatic propulsion drive system has got a 35-100 km/h scope for high-speed mobility, and the acceleration performance is also satisfactory. The mathematical model established and results obtained in this work will be instructive for further parameter sensitivity analysis and tamping machine design optimizations.

1. Introduction

Hydraulic tamping machines [1–3] are crucial and efficient apparatus in modern railway construction and maintenance [4,5]. In a railway tamping machine, the hydrostatic propulsion drive system [3,6] has two working modes, i.e., one low-speed mode for operation, and another high-speed mode for mobility, thus, it is significant to understand and optimize the parameters of the hydraulic system to obtain both better low-speed and high-speed performance of a tamping machine.

Wang et al. investigated different acceleration modes [7] for a fluid-power-driven engineering vehicle by using a hydraulic proportional pump and a hydraulic variable motor. Long [8] studied the acceleration performance of a military utility vehicle with hydrostatic propulsion drive. Yan proposed a hydraulic drive system to obtain [9] both high speed for mobility and low speed for tamping. Shi investigated the problems of out of step [10] in velocity and traction force of a tamping machine. Hu et al. improved the pressure stability [11] of the hydraulic clamping system of a tamping machine, by adding a resistance orifice in the circuit, and Ji et al. [12] fault diagnosis technology for the hydraulic system of their tamping machine.

A mathematical model formulating the high-speed performance of a tamping machine with closed-type hydrostatic propulsion drive system is established in this study, and a simulation model is built in the AMEsim software environment based on the mathematical model. Simulation results show that the tamping machine has got a maximum speed of 100 km/h for fast mobility, the acceleration performance at various speeds are feasible for operations with high efficiency. The model established
and results obtained in this work will be instructive for further parameter sensitivity analysis and tamping machine design optimizations.

2. Mathematical model of the high-speed hydrostatic propulsion drive

Figure 1 demonstrates that, in high-speed mode, the tamping machine is driven by three variable displacement hydraulic motors. Figure 2 illustrates the concrete high-speed hydrostatic propulsion drive system of the tamping machine; the hydrostatic propulsion drive system is a typical closed-type hydraulic system employing a two-way variable displacement hydraulic pumps and a two-way variable displacement hydraulic motors. The speed and direction of the tamping machine can be adjusted and alternated by volumetric speed control of the closed-type hydraulic system.

![Diagram of high-speed hydrostatic propulsion drive system](image)

Figure 1. Mechanism of the high-speed hydrostatic propulsion drive system of a tamping machine.

![Diagram of closed-type hydraulic power circuit](image)

Figure 2. Closed-type hydraulic power circuit of the high-speed 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 2, the output flow of the hydraulic pump \( Q_b \) can be formulated by

\[
Q_b = x_b D_b \omega_b - C_{ib} (p_1 - p_0) - C_{eb} p_b
\]  

(1)

where \( x_b, D_b, \omega_b, C_{ib}, C_{eb}, p_1 \) and \( p_0 \) are the adjustment coefficient, the maximum displacement, the
rotation speed, the inner leakage coefficient, the outer leakage coefficient, high pressure and back pressure of the pump, respectively. Thus, taking the Laplace Transform of Equation (1) to obtain

\[ Q_b(s) = x_b(s)D_b \omega_b - C_b p(s) \]  \hspace{1cm} (2)

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

\[ Q_b = C_{im}(p_i - p_0) + C_{em}p_t + x_m D_m \omega_m + V_0 \frac{dp_t}{\beta_c dt} \]  \hspace{1cm} (3)

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

\[ Q_b(s) = C_m p(s) + x_m(s)D_m \omega_m + D_m \omega_m(s) + \frac{V_0}{\beta_c} s p(s) \]  \hspace{1cm} (4)

where \( C_m \) is the total leakage coefficient of the hydraulic motor, and \( C_m = C_{im} + C_{em}, \omega_m(0) \) is the initial rotation speed of the hydraulic motor.

Because there exists

\[ v(s) = \omega_m(s) r \]  \hspace{1cm} (5)

where \( v \) and \( r \) are the vehicle speed and the wheel radius, respectively.

Thus, combining Equations (2), (4) and (5) will have

\[ x_b(s)D_b \omega_b - x_m(s)D_m \frac{v_0}{r} = \left( C + \frac{V_0}{\beta_c} s \right) p(s) + D_m \frac{v(s)}{r} \]  \hspace{1cm} (6)

where \( v_0 \) is the initial vehicle speed, \( C_i \) is the total leakage coefficient of the hydraulic system, and \( C_i = C_b + C_m, C_b = C_{ib} + C_{eb}. \)

In addition, the torque equilibrium equation of the hydraulic motor and load can be described as

\[ x_m D_m (p_i - p_0) = J_c \frac{d\omega_m}{dt} + B \omega_m + T_L \]  \hspace{1cm} (7)

where \( J_c \) and \( B \) are the effective moment of inertia and viscous damping coefficient of the powertrain, \( T_L \) is the load torque on the wheel, and can be formulated by [13]

\[ T_L = r \sum F = r \left[ mg \cos \beta (f_i + kv) + m g \tan \beta \frac{C_b A v^2}{21.15} + m \delta \frac{dv}{dt} \right] \]  \hspace{1cm} (8)

where \( \sum F, m, A \) and \( C_b \) are the total resistance force, the mass, the windward area and the drag coefficient of wind of the tamping machine, \( (f_i + kv) \) 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, combining Equations (7) and (8), and taking the Laplace Transform to obtain

\[ x_m(s)D_m p_{10} + D_m p(s) = \frac{J_c s + B}{r} v(s) + \left( m \delta s + m g r \cos \beta + \frac{C_b A r}{21.15} \right) v(s) \]  \hspace{1cm} (9)

Based on the above mathematical modelling of the high-speed hydrostatic propulsion drive system, the dynamic control block diagram of the hydraulic circuit can be obtained and demonstrated by Figure 3.
Figure 3. Dynamic control block diagram of the high-speed hydrostatic propulsion drive system of the tamping machine.

Thus, the transfer function of the high-speed hydrostatic propulsion drive system of the tamping machine can be formulated by

$$v(s) = \frac{D_0 D_m \omega_b x_b(s) - \left[ \frac{D_m^2}{r} V_0 - \left( C_i + \frac{V_0}{\beta_c} s \right) D_m p \right] x_m(s)}{as^2 + bs + c}$$

where the coefficients are described as

$$a = \frac{V_0 (J_c + m \delta r^2)}{r \beta_c},$$
$$b = \frac{V_0}{\beta_c} \left( \frac{B}{r} + mgrk \cos \beta + \frac{rC_p A}{21.15} \right) + C_i \left( J_c + m \delta r \right),$$
$$c = \frac{D_m^2 + C_i B}{r} + mgrk C_i \cos \beta + \frac{rC_p A C_i}{21.15}.$$

3. Simulation and result analysis

As shown by Figure 4, a simulation model of the high-speed hydrostatic propulsion drive system of the tamping machine is built in the AMEsim software environment, and simulations are carried out.

Figure 4. Simulation model of the high-speed hydrostatic propulsion drive system of the tamping machine.
Figure 5 demonstrates the profiles of the pressure and the output flow of the hydraulic motor during acceleration of the tamping machine, i.e., during volumetric speed control of the closed-type hydraulic system. Figure 5 shows that the hydraulic motor will get a stable pressure of 40 MPa in 20 seconds and a maximum and stable output flow of 668.1 L/min in about 50 seconds, as the result, Figure 6(a) shows the obtained acceleration speed of the tamping machine and calculated that the tamping machine has got a maximum speed of 100 km/h for mobility.

Figure 6(b) compares the acceleration performance of the tamping machine at various operation speeds, and shows that the acceleration periods are 32 seconds, 65 seconds and 143 seconds or so when the vehicle is accelerated to 50 km/h, 75 km/h and 100 km/h, respectively.

Thus, the tamping machine with a closed-type hydrostatic propulsion drive system has got a 35-100 km/h scope for high-speed mobility, and the acceleration performance is also satisfactory.

4. Conclusions

- During volumetric speed control of the closed-type hydrostatic propulsion drive system, the tamping machine will get a maximum speed of 100 km/h for mobility, the acceleration periods are 32 seconds, 65 seconds and 143 seconds or so when the vehicle is accelerated to 50 km/h, 75 km/h and 100 km/h, respectively.
- The tamping machine with a closed-type hydrostatic propulsion drive system has got a 35-100 km/h scope for high-speed mobility, and the acceleration performance is also satisfactory.
- The mathematical model established and results obtained in this work will be instructive for further parameter sensitivity analysis and tamping machine design optimizations.
Acknowledgements
The authors thank financial support from the National Natural Science Foundation of China (NSFC) under Grant No. 11572123, the Hunan Provincial Natural Science Foundation under Grant No. 2017JJ4015 and the Research Fund for High-level Talent of Dongguan University of Technology under Project No. GC200906-30.

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