A Study on the Design Approach and Theory for the Hydrostatic Propulsion Drive System of a Tamping Machine

Wenlin Wang\textsuperscript{1*} and Xiong Cai\textsuperscript{2}

\textsuperscript{1}School of Mechanical Engineering, Dongguan University of Technology, Dongguan 523808, China.
\textsuperscript{2}Dongfeng Honda Automobile Co., Ltd, Wuhan 430056, China

*Corresponding Author: E-mail: pianoww1@163.com

Abstract. With the fast developing of modern high-speed railways, it is urgent and meaningful to introduce effective design approach and theory for advanced railcar development. A design approach and theory for the hydrostatic propulsion drive system of a tamping machine is proposed in this study, the design approach and theory includes the overall design method, specification of the hydraulic pump and motor, the optimally matching of the hydraulic pump and diesel engine and the force equilibrium verification. A design case study is finally conducted and a series of good performance validates the effectiveness of the proposed design approach and theory: the designed closed-type hydrostatic propulsion drive system is smaller and more integrated in configuration, and is with high efficiency; the designed system is stable and fast in dynamic response with acceptable overshoot and errors, the displacement performance of the tamping machine is satisfactory for its cyclic tamping operations, and the tamping machine has also got a wide scope for high-speed mobility with satisfactory acceleration performance. Thus, the proposed design approach and theory in this work will be instructive for further tamping machine product development and design optimizations.

1. Introduction
A hydraulic tamping machines is a fluid-power-driven railway machinery which is crucial and efficient in modern railway construction and maintenance [1-3]. The hydrostatic propulsion drive system [4-6] of a tamping machine usually works in two modes: the low-speed mode for cyclic operation and another high-speed mode for mobility. Thus, it is meaningful to understand and study the design approach and theory for hydrostatic propulsion drive system of a tamping machine.

Kache [4] proposed an all-hydraulic hybrid system for railway machineries, Wang investigated the hydrostatic propulsion drive system of the 08-32 tamping machine [5], which has different acceleration modes by using a hydraulic proportional pump and a hydraulic variable motor; Zhou [6] studied the whole fluid power drive system of a continuous tamping machine. Yan [7] designed a hydrostatic propulsion drive system capable of running both on high speed for mobility and on low speed for tamping, Hu et al. improved the pressure stability [8] of the hydraulic clamping system of a tamping machine by using a resistance orifice, Zhai et al. [9] investigated the adjustment and maintenance technology for the hydraulic system of the 08-32 tamping machine. In addition, Dasgupta [10] and Ho et al. [11] introduced novel key hydraulic component technologies for use in the hydrostatic transmission system to improve the dynamic performance of the machine.
In this study, a design approach and theory for the hydrostatic propulsion drive system of a tamping machine is proposed. The design approach and theory includes the overall design method, specification of the hydraulic pump and motor, the optimally matching of the hydraulic pump and diesel engine and the force equilibrium verification process. A design case study is finally conducted and the simulation results show that the tamping machine has got a series of good performance. The proposed design approach and theory will be instructive for further tamping machine product development and design optimizations.

2. Design approach and theory

2.1. Overall design
There are usually two types of hydrostatic propulsion schemes for a tamping machine, as shown by Figures 1 and 2, the open-type and the closed-type.

The hydrostatic propulsion drive system using the open-type hydraulic power circuit (Figure 1) is popularly used in current tamping machines. The biggest advantage of open-type is flexible and easy to maintain, but the drawback is low efficient.

The closed-type hydrostatic propulsion drive system lets the oil circulate between the pump and motor, so it is smaller and more integrated in configuration, and is with high efficiency. In recent years, several more advanced tamping machines have used the closed-type hydrostatic propulsion drive system, and practices have validated the feasibility and efficiency of the closed-type system.

2.2. Specification of the hydraulic pump and motor
The hydraulic pump and motor are the most crucial devices in the hydrostatic propulsion drive system, they should be firstly specified. The required maximum rotation speed of the hydraulic motor $n_{\text{max}}$ in a tamping machine is described as 

$$ n_{\text{max}} = \frac{i v_{\text{max}}}{60 \pi D} $$

(1)
where \( i \) is the gear ratio, \( v_{\text{max}} \) is the maximum speed of the tamping machine, and \( D \) is the wheel diameter of the tamping machine. Thus, the displacement of the hydraulic pump \( q_b \) can be formulated by

\[
q_b = \frac{Q_b}{n_b} = \frac{Q_m}{n_k \eta_e} = \frac{n_{\text{max}} q_{\text{min}}}{n_b \eta_e}
\]

(2)

where \( Q_b, n_b, \eta_e \) are the flow, rotation speed and volumetric efficiency of the pump, \( q_{\text{min}} \) is the minimum displacement of the hydraulic motor.

2.3. Optimally matching of the hydraulic pump and the diesel

In mobile machinery applications, the swash plate type variable pump is more suitable for the working conditions, Figure 3 illustrates a type of variable pump with proportional electro-hydraulic control. Once the hydraulic pump is specified, it should be optimally matched with the diesel engine.

The output power of the diesel engine \( N_e \) is given by

\[
N_e = \frac{\pi}{3000} M_e n_e
\]

(3)

where \( M_e \) and \( n_e \) are the torque and rotation speed of the engine. The output power of the hydraulic pump \( N_b \) is given by

\[
N_b = \frac{P_b Q_b}{60} = \frac{P_b q_b n_e}{60000} = \frac{\pi M_b n_e}{3000}
\]

(4)

where \( P_b \) is the output pressure of the pump, \( M_b \) is the input torque of the pump, and \( M_b \) can be described as

\[
M_b = \frac{P_b q_b}{2\pi}
\]

(5)

Because there is a relationship between the output powers of the hydraulic pump and the diesel engine

\[
N_b = N_e \eta_1 \eta_2
\]

(6)

where \( \eta_1 \) is the mechanical transmission efficiency between the diesel engine and the hydraulic pump, \( \eta_1 \) could be 1 when the hydraulic pump is rigidly connected with the diesel engine; \( \eta_2 \) is the efficiency of the hydraulic pump and usually be regarded as 0.95. Thus, substituting Equations (3)-(5) to Equation (6) to obtain
\[ M_b = \frac{P_q \rho_n}{2\pi} = 0.95M_c \]  

(7)

As demonstrated by Figure 4, although when the engine works in ABCD, the maximum power can be obtained, but the engine is easily to be overloaded and stopped, so the best torque-speed performance of the engine is AEFG. When the engine works in its best points, it is usually assumed that the output torque of the engine remains constant, therefore, the optimally matching of the hydraulic pump and the diesel engine can be described as

\[ M_b = 0.95M_c = \text{const} \]  

(8)

2.4. Verification

When the overall design, specification of the hydraulic pump and motor, optimally matching of the hydraulic pump and the diesel engine are completed, the verification of force equilibrium should be carried out.

The total resistance force \( \sum F \) of a moving tamping machine can be formulated by

\[ \sum F = mg \left( f_0 + kv \right) \cos \beta + mg \tan \beta + \frac{C_D A v^2}{21.15} + m \delta \frac{dv}{dt} \]  

(9)

where \( m \), \( A \) and \( C_D \) are the total resistance force, the mass, the windward area and the drag coefficient of wind of the tamping machine, \( (f_0+kv) \) is the coefficient of rolling friction of the wheels, \( \beta \) is the slope angle, \( \delta \) is an effective coefficient of the rotation mass of the tamping machine, and \( \delta \) is given [12] by

\[ \delta = 1 + \sum \frac{I_w}{mr^2} + \frac{I_i}{r^2} (\frac{\Delta p}{\rho_m} i \eta_m) \]  

(10)

where \( I_w \) and \( I_i \) are the moments of inertia of the wheel and the fly wheel of the diesel.

The driving force of the hydrostatic propulsion \( F_C \) is formulated by

\[ F_C = \frac{M_C}{r} = \frac{\Delta p q_m i \eta_m}{2\pi r} \]  

(11)

where \( M_C \) is the output torque of hydraulic motor, \( r \) is the wheel radius, \( \Delta p \), \( q_m \) and \( \eta_m \) are the pressure difference, displacement and volumetric efficiency of the hydraulic motor, \( n \) is the number of driving axles, \( i \) and \( \eta_m \) are the gear ratio and mechanical efficiency of the gear box.

3. A case study

Using the design approach and theory proposed in Section 2, a case study of the hydrostatic propulsion drive system design for a tamping machine is conducted. The open-type hydraulic power circuit is used, the pump and motor specification, the optimally matching of pump and diesel and the force equilibrium verification are performed, the designed parameters of the hydrostatic propulsion drive system are summarized in Table 1.

**Table 1.** The designed parameters of the hydrostatic propulsion drive system of a tamping machine.

| Parameter                              | Value  | Parameter                              | Value  |
|----------------------------------------|--------|----------------------------------------|--------|
| Rotation speed of engine (r/min)       | 2300   | Empty laden mass of tamping machine (Kg)| 52000  |
| Total maximum displacement of variable pump (mL/r) | 305   | Full laden mass of tamping machine (Kg) | 60000  |
| Displacement of fluid compensation pump (mL/r) | 68.1 | Wheel radius (m)                      | 0.42   |
| Set pressure of main relief valve (MPa) | 40     | Maximum friction force (N)             | 9172.8 |
| Set pressure of fluid compensation relief valve (MPa) | 2.5  | Rolling resistance force (N)           | 3057.6 |
A simulation model of the designed hydrostatic propulsion drive system is built in the AMEsim software environment, and both the system dynamic response and operation performance of the tamping machine are evaluated, the simulation results are illustrated by Figures 5-8.
As shown by Figure 5, the Bode plot of the hydrostatic propulsion drive system illustrates that the hydraulic system has a large magnitude margin with a cross-over frequency of 2.28 rad/s, the corresponding phase margin is 95.31°, thus, the system is very stable.

The step response of the hydrostatic propulsion drive system, Figure 6, demonstrates that the system performs rapidly with acceptable overshoot, and the system becomes stable with small errors.

Figure 7 shows that with the designed hydrostatic propulsion drive system, the step displacement is 1.087 m, so the displacement performance is satisfactory for the tamping machine during its cyclic tamping operations.

Figure 8 shows that the acceleration periods are 42 seconds, 74 seconds, 137 seconds and 197 seconds or so when the vehicle is accelerated to 60 km/h, 80 km/h, 100 km/h and 110 km/h, respectively. Thus, the tamping machine with the designed hydrostatic propulsion drive system has got a wide scope for high-speed mobility with satisfactory acceleration performance.

4. Concluding remarks
- The closed-type hydrostatic propulsion drive system is smaller and more integrated in configuration, and is with high efficiency, so it is a trend in the future tamping machine applications.
- A design case study validates the effectiveness of the proposed design approach and theory for the hydrostatic propulsion drive system of a tamping machine, including the hydraulic pump and motor specification, the optimally matching of the hydraulic pump and diesel engine and the force equilibrium verification: The designed system is stable and fast in dynamic response with acceptable overshoot and errors, the displacement performance of the tamping machine is satisfactory for its cyclic tamping operations, and the tamping machine has also got a wide scope for high-speed mobility with satisfactory acceleration performance.
- The proposed design approach and theory for the hydrostatic propulsion drive system of a tamping machine in this work will be instructive for further tamping machine product development and design optimizations.

Acknowledgements
The authors are grateful for financial support from the National Natural Science Foundation of China (NSFC) under Grant No. 11572123 and the Research Fund for High-level Talent of Dongguan University of Technology under Project No. GC200906-30.

References
[1] Famurewa S M, Xin T, Rantatalo M and Kumar U 2015 Optimisation of maintenance track possession time: A tamping case study. Proc. IMechE, Part F: J. Rail & Rapid Transit 229 12–22
[2] Wen M, Li R and Salling K B 2016 Optimisation of preventive condition-based tamping for railway tracks. European J. Operational Research 252 455–465
[3] Aursudkij B 2007 A laboratory study of railway ballast behaviour under traffic loading and tamping maintenance (Ph.D Thesis, Nottingham: University of Nottingham, UK)
[4] Kache M 2014 Investigation an all-hydraulic hybrid system for diesel-hydraulic rail cars European Transport Research Review 6 181–189
[5] Wang X 2008 Research on the hydrostatic transmission running system of 08-32 tamping machine (Master’s Degree Thesis, Changsha: Central South University, China)
[6] Zhou Y Z 2011 Research on the fluid power drive system of a continuous tamping machine (Master’s Degree Thesis, Changsha: Central South University, China)
[7] Yan C G 2010 The analysis of 08-16 tamping machine's hydraulic drive system in the high-speed running Equipment Manufacturing Technology 10 28–29
[8] Hu Y, Hu J K and Fang J K 2014 Research on pressure stability of clamping hydraulic system in railway tamping device J. Railway Science and Engineering 11 146–150
[9] Zhai S C and Huang Z J 2008 Tamping equipment's hydraulic system of 08-32 tamping machine and it's use and service Hydraulics Pneumatics & Seals 28 55–57
[10] Dasgupta K 2000 Analysis of a hydrostatic transmission system using low speed high torque motor Mechanism and Machine Theory 35 1481–1499
[11] Ho T H and Ahn K K 2010 Modeling and simulation of hydrostatic transmission system with energy regeneration using hydraulic accumulator J. Mech. Sci. Tech. 24 1163–1175
[12] David A C 2009 Automotive Engineering (Oxford: Butterworth-Heinemann)