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Design and feasibility analysis of a novel auto hold system in hydrostatic transmission wheeled vehicle

Lv Chang[a,b], Jianguo Dai[b] and Shuo Liu[b]

[a]Public Logistics Information Technology Service Platform of Small and Medium Enterprises in Jiangsu Province, Huaian Vocational College of Information Technology, Huai'an, People's Republic of China; [b]Jiangsu Key Laboratory of Traffic and Transportation Security, Huaiyin Institute of Technology, Huai'an, People's Republic of China

ABSTRACT

Auto hold refers to vehicle's function of automatic braking under shutdown condition and automatic relieving braking force when vehicle starts. This paper presents a concrete structure of the novel auto hold system of hydrostatic transmission wheeled vehicle, and deeply analyses the working mechanism and control method. To validate its feasibility, AMESim software is adopted to establish the simulation model of the auto hold system based on mathematical theories. By using the data of a certain comprehensive operation mode as input, the curve of braking force of the auto hold system is obtained through simulation. It can be known through analysis that the proposed auto hold system can realize rapid response and provides stable braking force under different road conditions. Finally, the feasibility of the proposed auto hold system is validated by comparing the simulation data with actual braking force required and the electronic vehicle parking technology. It turns out that the auto hold system can meet the requirements of all road conditions. Besides, it also can provide a fault tolerance range for real vehicle experiments, and it will not cause adverse impacts due to excessive parking brake force.

1. Introduction

With the development of car factory and increase of people's living standard, the vehicle holdings are increasing, roads are becoming increasingly crowded, and the disadvantages of conventional mechanical hand braking system are more and more prominent. In the case of uphill or downhill start, vehicles with conventional hand braking system require operator to proficiently control the brake and clutch, otherwise the phenomenon of vehicle sliding may occur, leading to potential safety hazard. In addition, in the case of frequent starting and braking on a crowded road, the driver needs to control the brake pedal or hand brake all the time, which will cause driving fatigue if lasting too long, and thus affecting safe driving. If the driver forgets to pull up the hand brake after shutting down the vehicle, the vehicle may slide; in contrast, if the driver forgets to relieve braking force before starting the vehicle, the consequences will be power waste and lead to environmental pollution [1,2], loss of braking components and unnecessary wear. The release of auto hold concept perfectly overcomes the disadvantages of conventional hand braking system, reduces the rate of vehicle sliding, and enhances the safety of vehicle. The auto hold system has been widely applied in car factory [3–5].

By incorporating electronic devices such as clutch distance sensor, clutch engagement speed sensor and pedal position sensor as well as oil pump system [6], the auto hold system is an advanced substitute for conventional hand braking system, which can continue to provide braking force after release hand brake. Moreover, when the vehicle needs to start, the auto hold system can make a judge that the driving force is larger than driving resistance via sensors, and thus relieve the brakes and allow the vehicle to start safely.

Currently, common auto hold systems of vehicle mainly include mechanical electronic braking system, hydraulic electronic braking system, pneumatic electronic braking system and so on [7]. Compared with the conventional hand braking system, mechanical electronic parking brake system eliminates hand brake lever and other unnecessary components, which reduces space for braking system and lowers manufacturing cost. Hydraulic or pneumatic electronic braking system adopts electronic control modules on the basis of mechanical electronic braking system [8–10]. The auto hold systems mentioned above are consisted of a large number of components and require independent braking system and circuit control system, which are limited by high cost, complex structure and inconvenience in post-maintenance.

To simultaneously lower the manufacturing and maintenance costs of auto hold system of vehicles and increase its reliability, a novel auto hold system of hydrostatic transmission wheeled vehicle was proposed

in this paper. Firstly, the auto hold system is designed according to the characteristics of the hydrostatic transmission vehicle, and then the related simulation models are built and the simulation experiment is carried out by using AMESim software. Finally, the feasibility and reliability of the auto hold system are verified by comparing with the method of actual parking braking force and the method of the electrical park brake system (EPB). The system enjoys simple structure, lightweight, high space utilization, lowered manufacturing and maintenance costs, without requiring installations of independent devices.

2. Hydrostatic transmission wheeled vehicle

For hydrostatic transmission wheeled vehicle, oil pump and hydraulic motor are adopted to realize power transmission from engine to drive wheel. Hydrostatic transmission has advantages such as high transmission efficiency, smaller volume, lightweight, easy setting, simple operation and larger speed range (stepless change in speed) [11,12]. According to the difference of structure, the hydrostatic transmission system can be divided into two types: indirect drive-axle driving type and direct wheel driving type. In this study, the researched vehicle is indirect drive-axle driving type.

Hydrostatic transmission system mainly consists of engine, variable pump, and variable motor and drive axle and so on [13], as shown in Figure 1. Regarding the working principle of hydrostatic transmission system, it can be described as that first the mechanical energy of engine is converted into hydraulic energy via variable pump, and then converted back into mechanical energy via variable motor, thus realizing full hydraulic driving of the whole vehicle. Through analysing the combination mode, performance matching and control strategy of variable pump and variable motor, the power transmission can be optimized [14]. Hydrostatic transmission system adopts volumetric speed control strategy, i.e. implementing variable displacement control for hydraulic pump and hydraulic motor. This control strategy allows the vehicle to smoothly perform automatic stepless change in speed, so as to improve the dynamic performance and fuel economy of vehicle [15,16]. The hydrostatic transmission system takes up very little space, which is beneficial for engine arrangement, reducing engine room dimension and enlarging passenger space. Moreover, it enjoys large transmitted power per unit volume displacement, small inertia, sound power transfer performance and can effectively reduce the noise produced by vehicle vibration.

In hydrostatic transmission of vehicle, variable pump and variable motor are able to adjust output characteristics to certain extend under the action of variable control devices according to operational needs. Based on this feature, hydraulic brake can be realized, and the service life of braking device can be prolonged.
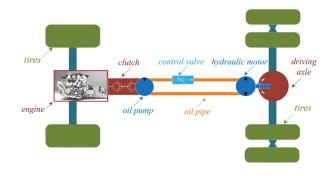

Figure 1. Power transmission route of hydrostatic transmission wheeled vehicle.
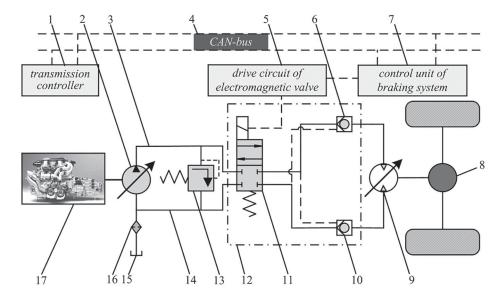
3. System design

Hydrostatic transmission wheeled vehicles are normally large engineering vehicles, of which operating environment is harsh and the demand for braking performance will be higher [17]. The auto hold system of hydrostatic transmission wheeled vehicle needs to satisfy the following requirements: fast respond at the moment of braking, stable braking force during the braking process, timely relieving braking at the end of braking. In addition, auto hold system should be reliable enough to guarantee stable operation under any operating condition. In design phase, the amount of additional parts applied should be as little as possible to reduce vehicle weight and lower manufacturing and maintenance costs.

Based on the above requirements, a novel auto hold system of hydrostatic transmission wheeled vehicle was designed in this paper, as shown in Figure 2. In the proposed auto hold system, a set of hydraulic locking device 12 is installed between variable pump 2 and variable motor 9 of original hydrostatic transmission system. Such hydraulic locking device consists of a two-way electromagnetic directional valve 11 and two hydraulic control one-way valves 6/10, of which the connection type is shown in Figure 2. In the scheme, the two-way electromagnetic directional valve is used to control braking line, because it has fast response rate, normal operation under vacuum, negative pressure or zero pressure conditions, high reliability upon braking or non-braking, simple structure and low failure rate. The hydraulic control one-way valve is sealed by a cone valve core. The higher the oil pressure, the more tightly the cone core is pressed, and the larger the parking brake force provided. The system can keep high seal pressure for a long time with good stability and high reliability.

The structure of auto hold system will be elaborated as follow. For the auto hold system proposed in this study, upper oil-way 3 is connected to the oil-out of variable pump 2, while the lower oil-way 14 is connected to the oil-in of variable pump 2. The two-way electromagnetic directional valve 11 is installed on upper oil-way 3 and lower oil-way 14. To secure safe

1—converter controller; 2—variable pump; 3—upper oil-way; 4—CAN bus; 5—electromagnetic valve drive circuit; 6—the first hydraulic control one-way valve; 7—parking brake control unit; 8—drive axle; 9—variable motor; 10—the second hydraulic control one-way valve; 11—two-way electromagnetic directional valve; 12—hydraulic lock; 13—decompression valve; 14—lower oil-way; 15—hydraulic oil tank; 16—oil filter; 17—engine
Fig.2 Auto hold system of hydrostatic transmission wheeled vehicle

Figure 2. Auto hold system of hydrostatic transmission wheeled vehicle.

range of oil pressure of upper oil-way 3 and lower oil-way 14, a decompression valve 13 is installed on upper oil-way 3 and lower oil-way 14 before the two-way electromagnetic directional valve 11. The oil-in of the first hydraulic control one-way valve 6 is connected to upper oil-way 3, its oil-out is connected to variable motor 9, and the control port is connected to lower oil-way 14. The oil-in of the second hydraulic control one-way valve 10 is connected to lower oil-way 14, its oil-out is connected to variable motor 9, and the control port is connected to upper oil-way 3. CAN bus 4, parking brake control unit 7, electromagnetic valve drive circuit 5 and two-way electromagnetic directional valve 11 are successively connected via electrical signal. Similarly, the transmission controller 1 and CAN bus 4 are connected via electrical signal as well. The hydraulic oil tank 15 of the auto hold system is connected to lower oil-way 14 via oil filter 16.

4. System control method

For the auto hold system proposed in this study, signals such as vehicle speed, road slope accelerator pedal and brake pedal signals are collected for processing and determining whether the brake system needs to start. Once the braking condition is met, the system will automatically trigger brake signal to brake the vehicle. Similarly, if the system determines to start the vehicle after analysis, the braking will be relieved timely.

4.1. Control method for braking start

The vehicle speed, road slope, acceleration pedal and brake pedal signals are used as inputs for determining
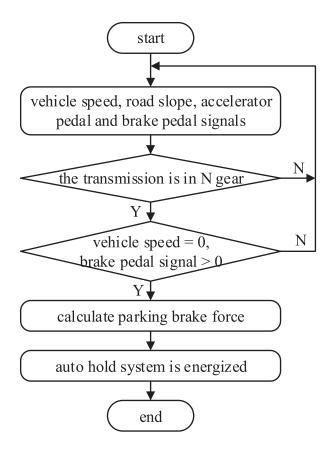

Figure 3. Flow chart for starting the auto hold system.

the states of the vehicle and calculating the parking brake force required for the vehicle. When the brake signal is received and the vehicle speed is 0, it is determined that the vehicle is in the parking state at this time. And the auto hold system is energized. The control

method for braking start is described by the flow chart in Figure 3.

When the vehicle is in temporarily parking state or shut down state, it will automatically shift into braking state. In the case of parking state, the vehicle speed is zero. When the vehicle speed collected on CAN bus 4 by braking control unit 7 is zero and the brake pedal is pressed, the coil of two-way electromagnetic directional valve 11 will be energized, thus blocking upper oil-way 3 and lower oil-way 14. At this point, the first hydraulic control one-way valve 6 and the second hydraulic control one-way valve 10 are in unidirectional conducting state, hydraulic lock 12 is on, variable motor 9 cannot make hydraulic oil enter into flowing state, and therefore the wheel rotation will be inhibited. If the vehicle has moving tendency under external influence, variable motor 9 will generate rotating tendency, however the hydraulic lock 12 is on, which makes the variable motor 9 stayed locked, so that the wheel rotation is inhibited, the vehicle movement is restricted, and the braking system will automatically start.

When the vehicle is in moving state, hydraulic lock 12 is in closed state, the variable motor 9 will rotate under the pressure difference between oil-in and oil-out, so that the vehicle moves normally and the braking system components will not be subjected to unnecessary wear.

4.2. Control method for relieving the braking system

At this time, the vehicle is in the parking state and the auto hold system is activated. When the accelerator pedal signal is detected, the auto hold system is turned off after the vehicle meets the starting condition. The control method for braking relieving is described by the flow chart in Figure 4.

When the vehicle is in parking state, the transmission should be engaged in D gear or R gear accompanying with accelerator pushing to start off the vehicle. At this point, the transmission gear changing signal and accelerator pedal pushing signal on CAN bus 4 are collected by parking brake control unit 7, the coil of two-way electromagnetic directional valve 11 will be de-energized, hydraulic lock 12 will be off, variable motor 9 can rotate under the pressure difference between oil-in and oil-out, the vehicle wheels rotate freely, and braking will be relieved.

5. Simulation procedure and results analysis

To validate the feasibility of proposed auto hold system, AMESim software was applied for modelling analysis. In this study, the basic parameters of vehicle are listed in Table 1.

In this experiment, the variable pump displacement was $0 \sim 24.6$ ml/r, the rated pressure was 21 MPa,
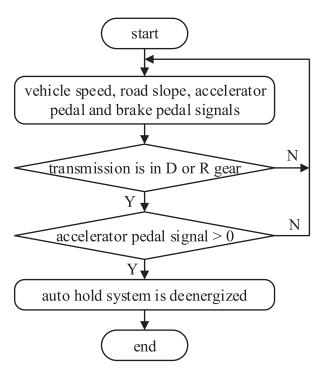

Figure 4. Flow chart for relieving the auto hold system.

maximum bearing pressure was 34.5 MPa; the variable motor displacement was $0 \sim 25$ ml/r, rated pressure was 21 MPa, maximum bearing pressure was 41.5 MPa; the working pressure of decompression valve was 25 MPa.

5.1. Simulation model establishment
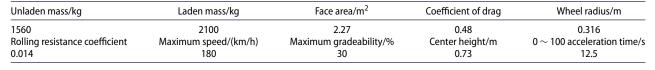
5.1.1. Vehicle model
In this study, the vehicle model with seven degrees of freedom (DOFs) in Vehicle Dynamics library of AMESim software was selected [16], as shown in Figure 5. The seven DOFs include vertical DOF, pitching DOF and rolling DOF of vehicle body as well as vertical DOFs of four wheels.

5.1.2. Driver and power system models
The driver model, internal combustion engine model and transmission model that are suitable for a vehicle with an automatic transmission were selected from IFP Drive bank of AMESim software [18]. A revolution speed sensor was installed on the engine output shaft to collect gear changing signal parameters of driver model, as shown in Figure 6. According to rotating speed signals of engine and wheels, the driver model can change the control of brake and accelerator pedals.

The driver model provided by AMESim software can be divided into two types: manual transmission driver model and automatic transmission driver model, respectively. In the present system, the driver model based on automatic transmission was adopted, as shown in Figure 7. The input port 3 of the module is speed control signal V_c (m/s), input port 4 is actual speed signal of the vehicle V_a (m/s), input

Table 1. Basic parameters of the vehicle.

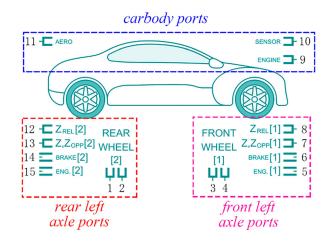| Unladen mass/kg	Laden mass/kg	Face area/m^2	Coefficient of drag	Wheel radius/m
1560	2100	2.27	0.48	0.316
Rolling resistance coefficient	Maximum speed/(km/h)	Maximum gradeability/%	Center height/m	0 ~ 100 acceleration time/s
0.014	180	30	0.73	12.5

Figure 5. Vehicle model.

port 2 is brake pedal opening signal a_b(varying within $0 \sim 1$, 1 represents the maximum opening), input port 1 is accelerator pedal opening signal a_g(varying within $0 \sim 1$, 1 represents the maximum opening).

The accelerator pedal opening signal a_g of driver model is shown as Equation (1):

$$a_g = C_{Pc} \cdot \Delta V + C_{Ic} \cdot \int \Delta V \cdot dt + C_{Ac} \cdot dV_c \quad (1)$$

where ΔV is the difference between controlled vehicle speed and actual vehicle speed; dV_c represents the derivative of speed control parameter V_c; C_{Pc} is proportionality coefficient of acceleration control loop; C_{Ic} is the overall coefficient of acceleration control loop; C_{Ac} is the coefficient of expectation of acceleration control loop. C_{Pc}, C_{Ic}, C_{Ac} are used to adjust acceleration control.
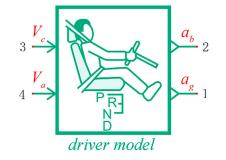

Figure 7. Driver model based on automatic transmission.

The brake pedal opening signal a_b of driver model is shown as Equation (2):

$$a_b = -C_{Pb} \cdot \Delta V - C_{Ib} \cdot \int \Delta V \cdot dt - C_{Ab} \cdot dV_c \quad (2)$$

where $C_{Pb}$is the proportionality coefficient of brake control in loop; C_{Ib} is the overall coefficient of brake control loop; C_{Ab} is the coefficient of expectation of brake control loop. These three coefficients are used to adjust brake control.

5.1.3. Wheel model and suspension model

Tire kinematics model as well as kinetic models of wheel and road were selected from Vehicle Dynamics bank of AMESim software. In addition, wheel speed sensor was installed to monitor wheel speed and provide information parameters for driver model and parking brake control unit, so as to realize switch-on and switch-off of auto hold system. The linking relation among modules is shown in Figure 8.

Automobile is a complex vibration system, which should be appropriately simplified during the design
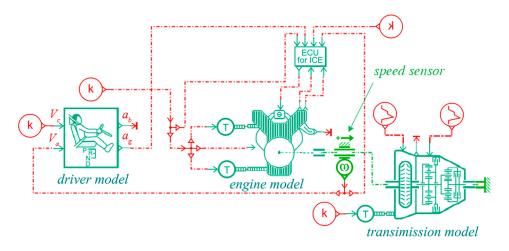

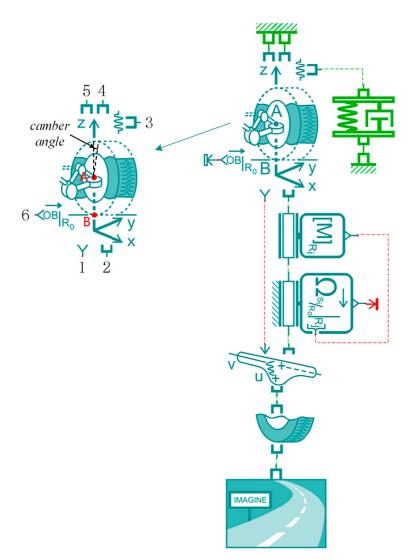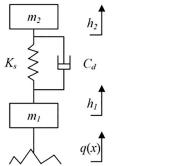**Figure 6.** Driver and power system models.

Figure 8. Wheel and suspension model.

a) vibration system consisting of
   body DOF and wheel DOF

b) suspension module

Figure 9. Suspension model.

phase. In exploring the kinematic equation of vehicle suspension system, the wheel model was simplified into two DOFs, the wheel is assumed to be rigid, with elasticity of tire being neglected, as shown in Figure 9. According to the modal analysis of the tires [19], the natural frequency of the overall motion mode of the in-plane tires is about 80 Hz. And the tires belt movement

below 80 Hz can be regarded as rigid body motion. Besides, when the vehicle is in the parking state, the tires are fixed due to the limitation of the parking brake force. The deformation of the tires not affect the parking state of the vehicle, so it is assumed that the tires are simplified for the rigid tires.

In Figure 9(a), m_1 is the unsuspended mass, m_2 is the suspended mass, K_s is the suspension stiffness, K_t is the tire stiffness, C_d the damping coefficient of damper, h_1, h_2 are vertical displacement coordinate of wheel and vehicle body, respectively, and the origin of them is their own equilibrium position, $q(x)$ is the road roughness function.

The kinematic equation is

$$m_2\ddot{h}_2 - C_d(\dot{h}_1 - \dot{h}_2) - K_s(h_1 - h_2) = 0 \qquad (3)$$

$$m_1\ddot{h}_1 + C_d(\dot{h}_1 - \dot{h}_2) + K_s(h_1 - h_2)$$
$$- K_t(q(x) - h_1) = 0 \qquad (4)$$

Tire model is the only model contacting with ground, which plays an important role for the whole vehicle model and directly affects the kinetic performances of vehicle, including wheel kinematics
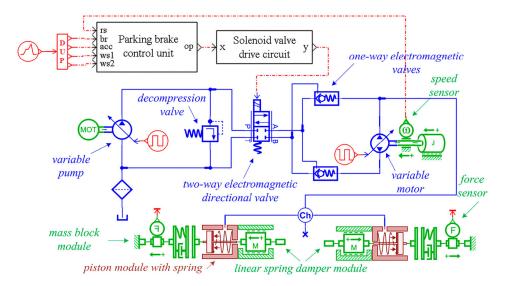
Figure 10. Auto hold system model.

model, damp model, tire elastic model, grounding model and road model, as shown in Figure 8.

The wheel force bearing equation under non-arc coordinate system O_{st} of point B is:

$$\vec{F}|_{O_{st}} = \begin{bmatrix} \vec{F} \cdot \vec{x}_w \\ \vec{F} \cdot \vec{y}_w \\ \vec{F} \cdot \vec{z}_w \end{bmatrix} \tag{5}$$

$$\vec{M}(B)|_{O_{st}} = \begin{bmatrix} \vec{M}(B) \cdot \vec{x}_w \\ \vec{M}(B) \cdot \vec{y}_w \\ \vec{M}(B) \cdot \vec{z}_w \end{bmatrix} \tag{6}$$

In the case, that rim migration is zero, the wheel force bearing function under principal axis coordinate system O_{sp} of point A is:

$$\vec{F}|_{O_{sp}} = \begin{bmatrix} \vec{F} \cdot \vec{x}_w \\ (\vec{F} \cdot \vec{y}_w)\cos\varphi + (\vec{F} \cdot \vec{z}_w)\sin\varphi \\ -(\vec{F} \cdot \vec{y}_w)\sin\varphi + (\vec{F} \cdot \vec{z}_w)\cos\varphi \end{bmatrix} \tag{7}$$

$$\vec{M}(A)|_{O_{sp}} = \begin{bmatrix} \vec{M}(B) \cdot \vec{x}_w + R_0((\vec{F} \cdot \vec{y}_w) \\ \cos\varphi + (\vec{F} \cdot \vec{z}_w)\sin\varphi \\ (\vec{M}(B) \cdot \vec{y}_w)\cos\varphi + (\vec{M}(B) \cdot \vec{z}_w) \\ \sin\varphi - R_d(\vec{F} \cdot \vec{x}_w) \\ -(\vec{M}(B) \cdot \vec{y}_w)\sin\varphi \\ +(\vec{M}(B) \cdot \vec{z}_w)\cos\varphi \end{bmatrix} \tag{8}$$

where R_0 is the wheel radius under static load; R_d is effective rolling radius; φ is camber angle.

5.1.4. Auto hold system model

According to the working principle of auto hold system of hydrostatic transmission wheeled vehicle mentioned above, variable pump, decompression valve, two-way electromagnetic directional valve, one-way electromagnetic valves, variable motor were added in Hydraulic bank of AMESim software and connected. Moreover, the revolution speed sensor was installed at output axle

of variable motor to monitor the output rotation speed of variable motor. Mass block modules with two ports, linear spring damper module, and piston module with spring were added in Mechanical library and Hydraulic Component Design library and connected to simulate hydraulic wheel side brake of vehicle, as shown in Figure 10 [20,21]. A force sensor was installed on linear spring damper module to monitor the change of wheel edge braking force more visually.

The variable pump and variable motor provided by AMESim are both ideal models, as shown in Figure 11, which do not consider flow loss or mechanical loss. The flow rate is only determined by rotate speed, oblique flow ratio, pump displacement and inlet pressure. The swash plate of variable pump is limited within 0–1, and that of variable motor is limited from −1 to 1.

The nominal flow rate of variable pump and variable motor is

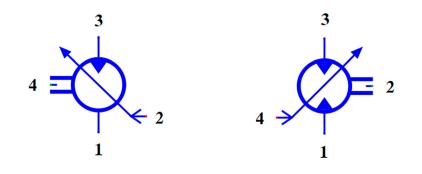$$Q_n = \frac{V_0 \cdot n_s \cdot \phi_s}{1000} \tag{9}$$

where Q_n is the nominal flow rate of variable pump or variable motor; V_0 is the displacement of variable pump or variable motor; n_s is the rotation speed of variable pump or variable motor; ϕ_s is swash plate threshold.

The reference pressure of variable pump and variable motor is equal to the linear combination of input port pressure and output port pressure:

$$P_r = [\eta_{re} \cdot P_{in} + (1 - \eta_{re}) \cdot P_{out}] \cdot 10^5 \tag{10}$$

where P_r is the reference pressure of variable pump or variable motor; P_{in} is input port pressure; P_{out} is output port pressure; η_{re} is actual conversion coefficient, varying within $0 \sim 1$, which can be calculated by Equation (11):

$$\eta_{re} = \frac{\tanh(\frac{n_s \cdot 1000}{n_w} + 1)}{2} \tag{11}$$

a) variable pump module   b) variable motor module

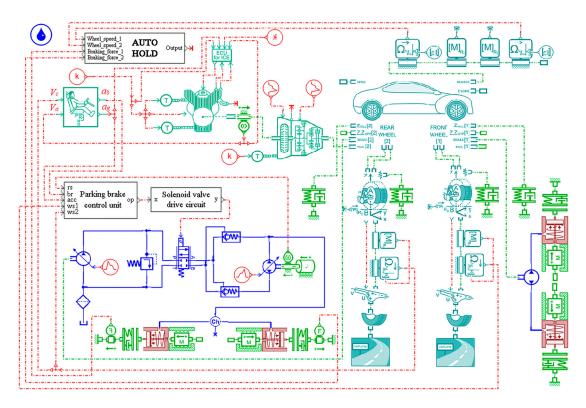**Figure 11.** Variable pump module and variable motor module.

Figure 12. Full vehicle simulation model of auto hold system based on AMESim.

where n_w is the typical rotation speed of variable pump or variable motor.

The output shaft torque of variable pump or variable motor is

$$T_{\text{out}} = \frac{(P_{out} - P_{in}) \cdot V_0 \cdot \phi_s}{20\pi} \quad (12)$$

At this point, the modelling and analysis of main components have been basically finished. According to the constructed models of various main components, as shown in Figure 12, and through actual parameters setting, simulation calculation was conducted.

According to the vehicle model, driver and power system models, wheel model, suspension model and auto hold system model introduced above, we built the full vehicle simulation model of auto hold system. The parameters of each model are set by the actual parameters of the vehicle and the selected devices, and as a

preparation for simulation experiment. In the simulation experiment, wheel speed, road slope, accelerator pedal and brake pedal signals were used as input signals. The experiment fully simulated the parking situation of vehicles on roads with different gradients, and verified whether the automatic parking system could guarantee the stability of vehicle's parking.

5.2. Determination of input signals

The full vehicle simulation model of auto hold system based on AMESim has been constructed, it is worth noting that the input signals of the model need to be determined prior to simulation. According the requirements of auto hold system of hydrostatic transmission wheeled vehicle, the signal collected by wheel rotation speed sensor, accelerator pedal open/close signal
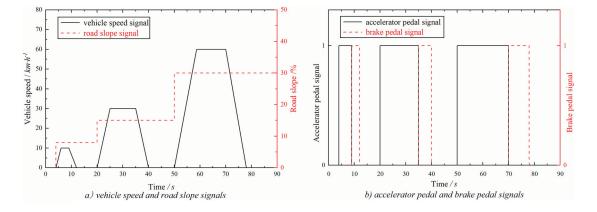
a) vehicle speed and road slope signals

b) accelerator pedal and brake pedal signals

Figure 13. Input signals of simulation system.

and brake pedal open/close signal should be recognized and identified by parking brake control system, so that the energizing or deenergizing of two-way electromagnetic directional valve can be controlled and the auto hold system of hydrostatic transmission wheeled vehicle can be realized. Therefore, the input signals of simulation model must contain wheel rotation speed signal, accelerator pedal open/close signal and brake pedal open/close signal, etc.

Considering the comprehensive status of vehicle when moving normally, in this study the brake pedal was pushed down at vehicle speed of 10, 30 and 60 km/h and then released, and we investigated whether the braking force provided by auto hold system at this point met the parking requirement or not.

The input signals of the simulation system are shown in Figure 13. In Figure 13(b), 0 means that the accelerator pedal and the brake pedal are not working, 1 means that the accelerator pedal or the brake pedal works and ignores the opening of the pedals. From phase of 0–4 s, the vehicle stops on a horizontal road without shutting off engine; from 4 to 20 s, the vehicle moves on a road with 8% gradient, during which the vehicle accelerates to 10 km/h from 4 to 6 s, moves at a constant speed from 6 to 9 s, the brake pedal is pushed down from 9 to 12 s, and the parking time is from 12 to 20 s. During the phase of 20–50 s, the vehicle moves on a road with 15% gradient, in which the vehicle accelerates to 30 km/h from 20 to 25 s, moves at a constant speed for 10 s, then 35–40 s is the braking time, and 40–50 s is the parking time. During the phase of 50–90 s, the vehicle moves on a road with 30% gradient, in which the vehicle accelerates to 60 km/h from 50 to 58.7 s, moves at a constant speed from 58.7 to 70 s, 70 to 78 s is braking time, and 78 to 90 s is parking time.

5.3. Discussion of simulation results

Simulation of system model was carried out with the data in comprehensive working condition as input, and the simulation curve of braking force can be obtained, as shown in Figure 14.
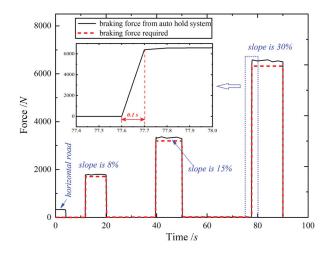

Figure 14. Comparison of braking force provided by auto hold system and actual braking force required.

According to Figure 14, the auto hold system can provide certain braking force when the vehicle stops and braking pedal is released. The braking force provided was stable, which was found to be varying within 4.0% under different working conditions according to simulation data. From the relation between simulation curve and time, it can be seen that the response time for forming or relieving braking force was extremely short, which is about 0.1 s. It indicates that the system has a fast response and can provide or relieve braking force timely. Through the relation between vehicle speed and time point for initially producing braking force, it can be known when the wheel rotation speed tended to zero, the one-way valve stopped, and the auto hold system started to work 0.1–0.2 s in advance to provide enough the braking force after the vehicle fully stopped. When the accelerator pedal opening signal was monitored by parking brake control unit, the electromagnetic valve was quickly energized, auto hold system quickly relieved braking force, so that the components wear can be avoided.

To evaluate the performance of auto hold system, we should consider whether it can realize auto hold function, as well as whether the braking force provided can meet the requirements. Based on the given basic
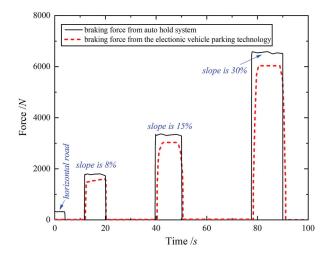
Figure 15. Comparison of braking force provided by auto hold system and the electronic vehicle parking technology.

parameters of vehicle and road conditions, the braking force required for the researched vehicle to realize parking on roads with different gradients can be calculated, and then the curve of braking force can be plotted based on the calculation results, as shown in Figure 14. It can be clearly seen that even minimum braking force provided by auto hold system can meet the requirements of braking on road with different gradients, indicating the feasibility of the proposed auto hold system.

In order to further verify the reliability of the auto hold system, we use the EPB proposed by Xu [22] for comparative verification. Simulink models were built to carry out simulation experiment, including vehicle mass model, braking force calculation model, road slope model and disc brake compression model. Then we set the same parameters for the models and carried out the experiment with the same simulation time, and obtained the experimental result based on the EPB. The comparison of braking force provided by auto hold system and the EPB can be seen in Figure 15. Since the disc brake has a free path, it has a certain hysteresis compared to the auto hold system and its hysteretic time is 0.89 s. When vehicle parking on a road with 8% gradient, the minimum parking brake force provided by the auto hold system is 1768.3 N and the one provided by the electronic vehicle parking technology is 1601.7 N, which is 10.4% higher. When vehicle parking on a road with 15% gradient, the minimum parking brake force provided by the auto hold system is 3367.3 N and the one provided by the EPB is 3034.2 N, which is 11% higher. When vehicle parking on a road with 30% gradient, the minimum parking brake force provided by the auto hold system is 6583.1 N and the one provided by the EPB is 6036.5 N, which is 9.1% higher.

According to the comparisons between the results of the auto hold system and the other two methods, the auto hold system has three advantages, as follows:

(1) The auto hold system proposed by this manuscript provides a valley value of the parking brake forces greater than the ones required by the vehicle, thus ensuring that the vehicle can be parked safely on the ramp. Moreover, the parking brake force provided by the system can be changed in real time with the slope to ensure that the vehicle can be safely parked on any ramp.

(2) Through the comparative analysis of the parking brake force provided by the auto hold system and the EPB system, the parking brake forces provided by the auto hold system are about 10% larger than that provided by the EPB system. Therefore, the system provides a greater security and can provide a fault tolerance range for subsequent real vehicle experiments. Due to the influence of the sensors' errors and ramp information, the parking brake forces in the real vehicle experiments may be greater than the theoretical ones required by the vehicle, so it is necessary to provide a fault tolerance range to ensure that the vehicle can be safely parked on any ramp.

(3) If the parking brake force is very large, more power is required to be supplied to the auto hold system, which will result in waste of power and the parking brake force. And it is also possible that hysteresis occurs when the vehicle starts. Hence, the advantages of the auto hold system are not proportional to the parking braking force. However, the parking brake forces provided by the auto hold system are only 10% larger than ones of the EPB system, and will not cause adverse impacts due to excessive parking brake force. It is also possible to provide a security domain for the vehicle to be parked safely.

6. Conclusions

This study elaborates the design scheme of a novel auto hold system of hydrostatic transmission wheeled vehicle and proposes control strategy for starting or relieving braking force. The proposed auto hold system realizes full-automatic control of braking and timely provides or relieves braking force when the vehicle stops or starts frequently. Through modelling and simulation analysis based on AMESim software, the proposed auto hold system can exert rapid response (the response time is 0.1 s) under different working conditions and provide a stable braking force (the braking force is controlled within 4.0%). Through comparing simulation results with actual braking force required in different road conditions, it can be confirmed that the braking force provided by the auto hold system meets the requirements of all road conditions. The auto hold system provides the parking brake force is only 10% larger than the EPB system. It can provide a fault tolerance range for real vehicle experiments, and it will not cause adverse impacts due to excessive parking brake

force. The study verifies the reliability of the auto hold system, and provides theoretical significance and engineering value for further development of the automatic maintenance system.

Based on the conclusions of this paper, we propose the future research direction of the auto hold system for hydrostatic transmission vehicles. Subsequent research should modify the vehicle based on the auto hold system, and verify the parking capacity of a real vehicle on different roads.

Disclosure statement

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