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Simulation and Data Analysis of Energy Recovery Sensing on a Parallel Hydraulic Hybrid Crane

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ABSTRACT

In order to study the braking energy regeneration characteristics of the Front-mounted Parallel Hydraulic Hybrid Crane (FPHHC), the AMESim simulation models are established and analyzed by establishing the vehicle dynamics model and referencing to the actual data of the crane and physical hydraulic components, the simulation results are verified by road tests on the experimental prototype. The experiment results basically match with the simulation results. In the vehicle braking process, the hydraulic hybrid system of the experimental prototype can effectively recycle the vehicle braking energy, the energy recovery rate is up to 50.84%, and the energy-saving effect is obvious. The performance in terms of the vehicle acceleration and braking has obviously been improved, then the prototype crane, which can show that this system has a certain practical significance in saving energy.

KEY WORDS: AMESim simulation, Braking energy recovery, Hydraulic hybrid, Road tests

1. INTRODUCTION

IN recent years, with the rapid development of worldwide industrial production, transportation and other industries, the globalization of market competition is increasingly fierce, the problems of energy shortage, environmental pollution, and so on, make people think about a vehicle's fuel economy and emissions, Clarke P and Yang, et. al. (2017) has proposed that it makes the automobile industry head to the direction from the type of traditional pure fuel to the type of clean energy eventually. Simon B, et. al. (2007) and Li Jun-cheng et. al. (2011) consider that the hydraulic hybrid technology is an important branch of hybrid species; Offer G J et. al. (2010), Liu Xin-hui (2010), Zeng Zi-wei (2012) and Fu Song-biao et. al. (2010) thinks that for the hydraulic hybrid's high energy utilization ratio and large power density, environmental protection and other advantages, Zheng, et. al. (2012) have proposed that it is especially suitable for the need of frequent start-stop on large-tonnage truck, engineering vehicles, city buses and other heavy vehicles. According to the energy flow and power flow configuration relations of the driving system, Liu Tao, et. al. (2010) has proposed that the hybrid powertrain can be divided into three kinds of series, parallel and mixed type. Among them, vehicles can be independently driven by the engine or the hydraulic secondary components of the parallel hybrid system, Sun, et. al. (2009, 2010) and Zhang Qing-yong, et. al. (2011) have proposed that making it convenient to refit vehicles based on existing models.

Considering the characteristics of the hydraulic hybrid vehicle configuration synthetically, based on the FPHHC, the vehicle model is established by AMESim software and the simulation analysis is done to study the effect of energy recovery and utilization, and the validity of the simulation results is verified on experimental prototype by road test.

2. WORKING PRINCIPLE AND MODEL OF PARALLEL HYDRAULIC HYBRID VEHICLE

2.1. *Composition and Working Principle of Biaxial Parallel Hydraulic Hybrid System*

The BIAXIAL parallel hydraulic hybrid system is composed of the traditional internal combustion engine system and hydraulic secondary component, which are connected together by a torque coupler. J.E. Naranjo, et. al. (2012) has proposed that the traditional internal combustion engine system of components include the engine, clutch, gearbox, drive axle, the accelerator pedal and brake pedal. The hybrid system is composed of the hydraulic pump/motor, hydraulic

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accumulator, torque coupler and the corresponding hybrid controller and so on. Based on the traditional rear-mounted parallel hybrid vehicle (RPHHV) structure (as shown in Figure 1), the front-mounted parallel hybrid vehicle layout mode (FPHHV) is putt forward (as shown in Figure 2), the torque coupling device is put between the engine and the gearbox, the secondary component speed range can more match the range of engine speed, that can make the engine work more in the high efficient area, and then improve the fuel economy and power performance of the vehicle.

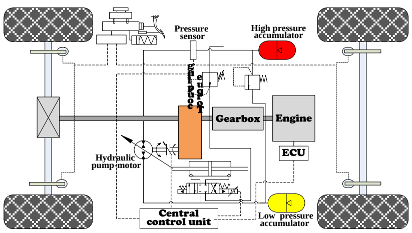


Figure 1. Structure Diagram of RPHHV.

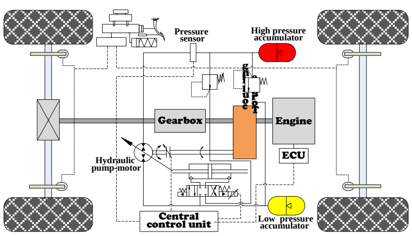


Figure 2. Structure Diagram of FPHHV.

FU, and Zhang, et.al. (2010, 2111) have proposed that the key technology of a hydraulic hybrid system is the interconversion between pump and hydraulic motor by adjusting the work quadrant of the hydraulic secondary component. In the stage of vehicle accelerating, the engine maintains the idle running, while the hydraulic pump/motor works in the condition of motor, the hydraulic accumulator release energy by a hydraulic secondary component, the hydraulic hybrid system drives the vehicle as the main source of power. When the hydraulic accumulator releases the energy completely or the vehicle achieves a certain speed, the vehicles engine begins to work, at the same time, the hydraulic hybrid system separates from the vehicle transmission system and the vehicle is not driven.

2.2. Mathematical Model of the Working Condition of Vehicle Driving and Braking

The vehicle is driven by driving forces and various resistance forces in the driving conditions. Ding Zuo-wo, Zhao Dong-biao. (2007) have proposed that the driving forces mainly include the forces provided by the engine and the hydraulic hybrid system. Resistance mainly includes; rolling resistance, air resistance, slope resistance and acceleration resistance. In the process of driving, the dynamic equation of the hydraulic hybrid vehicles is the same with the traditional vehicle, assuming that the vehicle traveling track is a straight line, and then the balance equation in the process of driving is as follows:

$$F_{dr} - F_f = m\frac{dv}{dt} \tag{1}$$

Where F_{dr} is the vehicle driving force (N); F_f is the total resistance in the process of vehicle travelling (N); v is vehicle instantaneous velocity (m/s).

$$F_f = F_{air} + F_r + F_g \tag{2}$$

Where, F_{air} is air resistance (N); F_r is rolling resistance (N); F_g is slope resistance (N). Rolling resistance is determined by the total mass of the vehicle m (kg) and the coefficient of wheel rolling friction f , usually take $f = 0.02$.

$$F_r = mg \cdot f \tag{3}$$

The air resistance equation is

$$F_{air} = \frac{C_d A v^2}{21.15} \tag{4}$$

Where, C_d is air drag coefficient; A is the area of vehicle windward. The slope resistance equation is

$$F_g = mg \cdot \sin\alpha \tag{5}$$

The acceleration resistance equation is

$$F_a = m \cdot \frac{dv}{dt} \tag{6}$$

The vehicle driving force equation is

$$F_{dr} = F_e + F_{hb/M} \tag{7}$$

Where, F_e is the driving force provided by engine (N); $F_{hb/M}$ is the driving force regenerated by hydraulic system (N).

The driving force provided by engine is

$$F_e = \frac{T_e i_0 i_g \eta_e}{r} \tag{8}$$

Where, T_e is engine torque (Nm); η_e is engine transmission efficiency. Then the total force balance equation in the working condition of driving is

$$\frac{T_e i_0 i_g \eta_e}{r} + F_{hb/m} - \frac{C_d A v^2}{21.15} - mgf - mg\sin\alpha = m \cdot \frac{dv}{dt} \tag{9}$$

The main types of forces of the vehicle in braking condition are; air drag, rolling resistance, gradient resistance, engine anti-drag resistance and wheel braking force. Then the force equation of equilibrium in the braking condition of vehicle is

$$F_b + F_{air} + F_r + F_g + F_{eb} = m\frac{dv}{dt} \quad (10)$$

Where, F_b is wheel braking force (N), F_{eb} is engine anti-drag resistance (N). Whereby, the wheel braking force includes the hydraulic regenerative braking force provided by the hydraulic hybrid system, and the friction braking force of the original vehicle provided by its braking system, whereby, the wheel braking force equation is

$$F_b = F_{hb/P} + F_s \quad (11)$$

Where, $F_{hb/P}$ is hydraulic regenerative braking force; F_s is friction braking force.

The engine anti-drag resistance equation (Yu, et.al. 2009) is

$$F_{eb} = \frac{I_e i_0^2 i_g^2}{r^2 \eta_e}\frac{dv}{dt} \quad (12)$$

Where, I_e is rotational inertia of engine flywheel ($kg \cdot m^2$). Then the total force balance equation in the working condition of braking is

$$F_{hb/p} + F_s + \frac{C_d A v^2}{21.15} + mgf + mg\sin\alpha + \frac{I_e i_0^2 i_g^2 \eta_e}{r^2 \eta_e}\frac{dv}{dt} = m \cdot \frac{dv}{dt} \quad (13)$$

2.3. Main Parameters Design of the Hydraulic System

The displacement of hydraulic pump/motor is

$$V_s = \frac{X_p}{X_{p\max}} V_{s\max} \quad (14)$$

Where, V_s is the displacement of hydraulic pump/motor (m^3/rad); $X_{p\max}$ is the maximal one-way displacement of the variable hydraulic cylinder (m); $V_{s\max}$ is hydraulic pump/motor maximum displacement (m^3/rad).

When the hydraulic pump/motor is operated in the hydraulic pump state, its driving force comes from the load. In the process of braking, the braking torque of hydraulic pump/motor is

$$T_P = \frac{pV_s}{2\pi} \quad (15)$$

Where, p is the pressure of hydraulic accumulator (MPa).

The drive balance equation of hydraulic pump/motor in pump working conditions is

$$T_{p/m} = \frac{F \cdot r}{i_b i_0 \eta_T} \quad (16)$$

Where, $T_{p/m}$ is output torque of the hydraulic pump/motor (Nm); η_T is system efficiency; i_b is speed ratio of torque coupler, it should ensure that the secondary component runs in its specified speed range and as much as possible works in the efficient area in the braking and driving process;

$i_b = \frac{0.377 r \omega_{p/m}}{v_{avg} i_0}$, i_0 is the gear ratio of the main reducer; r is wheel rolling radius (m). Then when the hydraulic pump/motor works in pump working condition, the wheel braking force is

$$F_{hb/P} = \frac{T_p i_{P/M} i_0 i_g \eta_{p,v}}{r \eta_{p,t}} \quad (17)$$

When the hydraulic pump/motor works during the motor working condition, the power comes from the oil pressure in accumulator. In the process of driving, the output torque of hydraulic pump/motor is

$$T_M \quad \frac{pV_s}{2} \quad (18)$$

Then in the working condition of hydraulic motor, the driving force of wheel is

$$F_{hb/M} = \frac{T_M i_{P/M} i_0 i_g \eta_{m,v} \eta_{m,t}}{r} \quad (19)$$

Assuming that the vehicle travels on a flat and straight road, when the vehicle brakes, the energy equilibrium equation is

$$E_r = E_{fri} + E_{acc} + E_{\text{air-res}} \quad (20)$$

Where, E_r is vehicle kinetic energy (J); E_{fri} is the loss of energy when the wheel overcome the rolling friction (J); E_{acc} is the energy of the accumulator recycling (J); $E_{\text{air-res}}$ is the loss of energy when the vehicle overcomes the air resistance (J).

$$E_r = \frac{1}{2} m (v_0^2 - v_1^2) \quad (21)$$

Where, v_0 and v_1 respectively correspond to the speed of the vehicle at the beginning of the brake and brake speed after time t.

$$E_{fri} = fGs = fmgs \quad (22)$$

Where, s is the displacement in the process of vehicles braking (m).

$$\sum \mathrm{E}_{\mathrm{acc}} = -\int_{v_1}^{v_2} pdV = \frac{p_1V_1^n}{n-1}(V_2^{1-n} - V_1^{1-n}) = \frac{p_1V_1^n}{n-1}\left[(\frac{p_1}{p_2})^{\frac{1-n}{n}} - 1\right] \tag{23}$$

Where, p_1 is the minimum working pressure (MPa); p_2 is the maximum working pressure (MPa); V_1 is the gas volume when working pressure is p_1, V_2 is the gas volume when working pressure is p_2, n is air polytropic exponent (Ding, et.al. 2007)

$$E_{\text{air-res}} = \int_0^t \frac{C_D A}{21.5} v^2 ds = \frac{C_D A}{21.5 \times 4a}[v_0^4 - (v_0 - at)^4] \tag{24}$$

Where: t is braking time (S); a is vehicle braking deceleration (m/s^2).

The recovery rate of braking energy is defined as

$$\varepsilon_r = \frac{\sum E_{\mathrm{acc}}}{\sum E_r} = \frac{\frac{p_1V_1^n}{n-1}(V_2^{1-n} - V_1^{1-n})}{\sum \frac{1}{2} m(v_2^2 - v_1^2)} \tag{25}$$

The reuse rate of braking energy is defined as

$$\varepsilon_{\mathrm{u}} = \frac{\frac{1}{2}mv_2^2}{\frac{1}{2}mv_0^2} = \frac{v_2^2}{v_0^2} \tag{26}$$

Where, v_2 is the highest speed that the vehicle can reach when it is driven forward alone by the hydraulic hybrid system after the recovery of energy is completed.

3. AMESIM SIMULATION MODE OF THE CRANE

IN order to verify the fuel economy of the front-mounted parallel hydraulic hybrid crane under complex conditions, Fu Yong-ling, Qi Xiao-ye (2011) have proposed that the vehicles backward simulation, according to the model is set up in the AMESim simulation environment. The drive cycle is inputted by means of IFP Drive. In the vehicle driving cycle, the drive and the brake torque needed by the vehicle is calculated by a different gear, and the control signal is inputted to the engine, braking system of the original vehicle and hydraulic hybrid system through the controller. According to the principle of the system and the models of main components, as shown in Figure 3, the vehicle backward simulation model of AMESim is established, which mainly includes the control module, the switch gear and hydraulic system module. Some specific parameters of the simulation model are listed in Table 1.

4. ANALYSIS OF SIMULATION RESULTS

AN integrated UDDS driving cycle is chosen and used as a simulation condition, which is put forward by the environmental protection agency (EPA), and used first for testing the fuel economy of medium and heavy trucks in the United States. The simulation time is 1500s; the simulation step size is 0.01s. When a vehicle is in a driving condition of the UDDS, the change of vehicle speed is shown in Figure 4, you can clearly see that whether driving or braking of the vehicle are all very frequent.

The pressure of the hydraulic accumulator and displacement change of the secondary component is shown respectively in Figure 5 and Figure 6. Under the simulation condition of the UDDS, the hydraulic hybrid system can effectively recycle and reuse the vehicle's braking energy, and the hydraulic secondary components quickly and effectively switch between the working conditions of pump and motor.

As seen from Figure 7, in a complete UDDS condition, the output power of a traditional vehicle engine is generally greater than the one of hybrid vehicle engine. A hybrid vehicle starts and stops frequently on a city road, by regulation of the hydraulic secondary component, the hybrid system realizes the recycling and release of power, which can effectively reduce the power output and fuel consumption of the engine.

As shown in Figure 8, in the phase from 344s to 360s, while the vehicle starts from static, the hydraulic hybrid system has the effect of auxiliary driving, which makes the engine effectively avoid the inefficient zone of low speed and large torque.

In order to clearly verify the ascension of a vehicles dynamic performance, climbing ability and starting acceleration capability, the vehicle is simulated and analyzed respectively. The slope of simulation is set at 35%, the accumulator outlet pressure is set as 30MPa; that is, and the accumulator has been filled and has reached the maximum pressure acquiescently. As seen in Figure 9, the speed of the hydraulic hybrid vehicle has improved significantly and the hydraulic hybrid vehicle can climb while the traditional vehicle' s speed is 0 all the time and it can't climb. From Figure 10, it can clearly be seen that the hybrid vehicle can accelerate faster than the traditional vehicle; it is almost twice as fast as the traditional vehicle. The simulation results show that the front-mounted parallel hybrid structure can significantly improve the vehicle's dynamic performance. The comparison of the simulation results of the engine power and fuel consumption of the prototype vehicle, which does not add the hydraulic regenerative system and FPHHV, is shown in Figure 11(a) and (b).

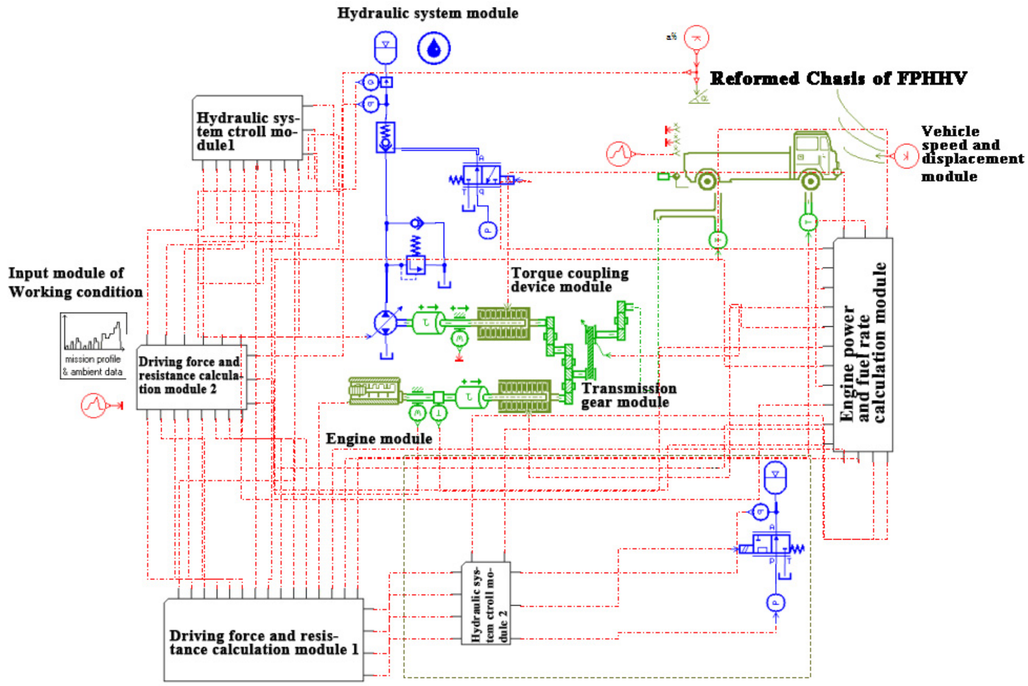


Figure 3. Simulation Model of the FPHHC Based on AMESim.

Table 1. Selection and Parameters of the Main Sub-model of a Simulation Model.

Components name	Parameters name	Value
Multiple-disc clutch	Maximum friction torque(Nm)	104
Safety valve	Opening pressure(MPa)	30
Hydraulic control one-way valve	Opening pressure(MPa)	0.1
	Pilot ratio	3.5
Hydraulic accumulator	Air polytropic exponent	1.4
Hydraulic oil	Oil temperature (℃)	40
	Density(kg/m3)	850
	Modulus of volume elasticity(MPa)	1700
	Absolute viscosity(cP)	51
Driving environment	Wind speed(m/s)	0
	Air density(kg/m3)	1.205
	Environment temperature (℃)	25

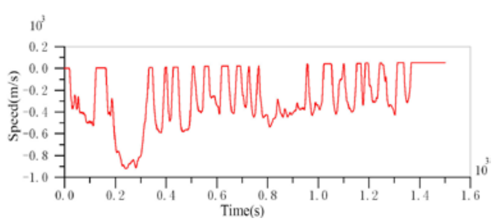


Figure 4. Vehicle Speed.

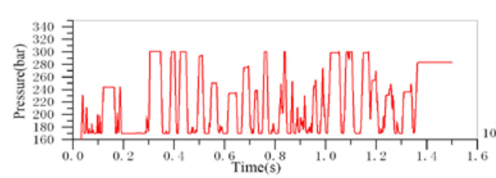


Figure 5. Accumulator Pressure.

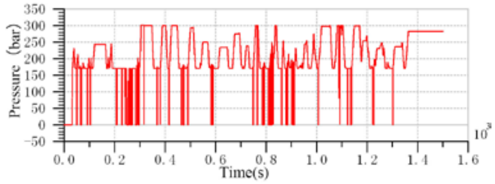


Figure 6. Pressure Change of the Hydraulic Secondary Component.

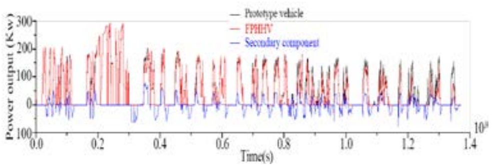


Figure 7. Power Output of the Engine and the Secondary Component.

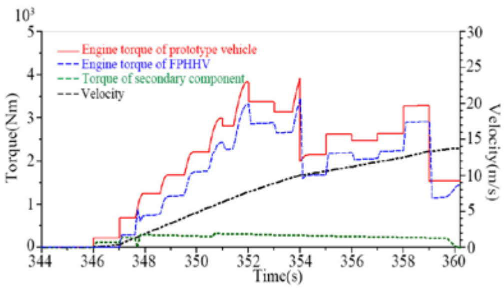


Figure 8. Output Torque Contrast.

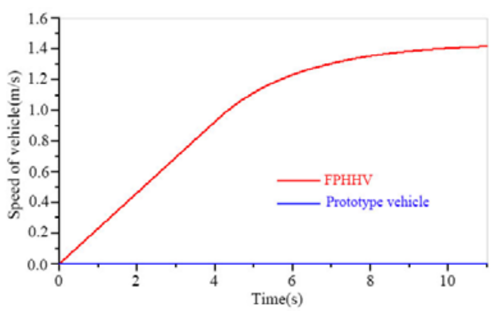


Figure 9. Gradeability Contrast.

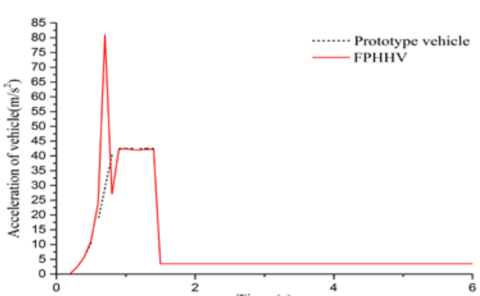


Figure 10. Contrast of Starting Ability.

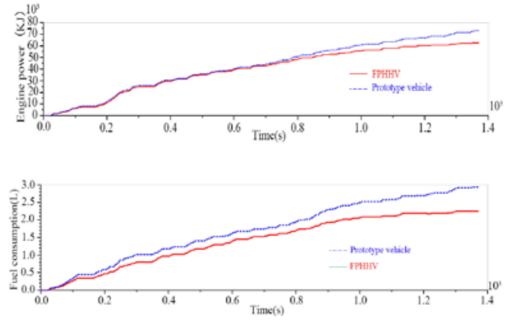


Figure 11. Contrast of Power and Fuel Consumption.

As seen from the figure, in an effective work circle, the engine of the prototype vehicle works at 72824.8 kJ, and the engine of the FPHHV works at 62510.3 kJ, the energy-saving rate is14.163%. In a full UDDS condition with the range of 11 km, the fuel consumption of the prototype vehicle is 2.95L, while the FPHHV is 2.34L, and the oil-saving percentage is about 21%.

5. EXPERIMENTS AND RESULTS ANALYSIS

5.1. Experimental Prototype and the Schematic Diagram of the FPHHC

IN order to verify the effectiveness of the hybrid configuration and the precision of the simulation model, in this article, an experimental prototype was developed based on a type of domestic crane and the actual road test and experiment were carried out. The overall appearance of the experimental prototype and the layout of hydraulic hybrid system are shown in Figure 12 and Figure 13; the experimental prototype structure is shown respectively in Figure 14.

Figure 12. Overall Appearance of the FPHHC Experimental Prototype.

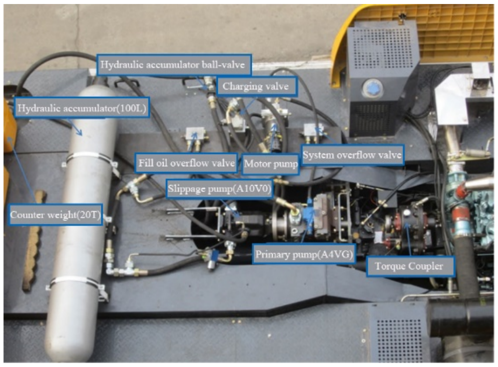


Figure 13. Hydraulic Hybrid System Diagram of the FPHHC Experimental Prototype.

5.2. Test and Result Analysis of the Experimental Prototype

Considering the road conditions and the actual situation of the vehicle, in this paper, the corresponding performance experiments of the experimental prototype were conducted under the conditions of a flat and climbing road. A series of experimental curves were analyzed through the way of a real vehicle test.

1) Test and result analysis of energy recovery efficiency

When the crane was accelerated to a certain speed, the crane was braked by the fixed displacement of the hydraulic secondary component (A4VG). The crane kinetic energy was calculated based on the initial speed of the start braking; the hydraulic energy under the same accumulator pressure change was calculated according to the AMESim equivalent model. As is shown in Figure 15, the crane starts to brake at the initial speed of 16.687 km/h, when the speed gets to 1.833 km/h; the accumulator pressure begins to decline. The charging pressure of accumulator is 7 MPa, the accumulator volume is 100L, braking time is about 18.8s, and braking distance is about 47m. When the accumulator pressure begins to decline, the revolving speed of the secondary component is about 154 rpm. Then the rate of energy recovery is 50.84%. Braking energy recovery efficiency varies depending on the pump, the control current is different. Considering the influence of the hydraulic pump being noisy and oil make-up insufficiently, the control current was given at 550mA (max: 650mA), the recovery efficiency is on the average above 50%, so the recovery efficiency is higher relatively.

2) Test and result analysis of energy recycling efficiency

Utilize the hydraulic system to brake the crane, and then the braking energy is stored in the accumulator and used for starting again. The energy regeneration efficiency is calculated based respectively on the vehicle speed at the time of braking and the speed at which the vehicle is started again. As is shown in Figure 16, wide fluctuations in the accumulator pressure proves that there is braking energy regeneration and is reused effectively. According to 16 MPa of the initial pressure of accumulator to calculate the energy recycling efficiency, when relied completely on the hydraulic system to brake and drive, the energy recycling efficiency was about 13.1%. If the accumulator pressure is not lost during the waiting of the vehicle, the energy regeneration efficiency can be further improved.

3) Test and result analysis of acceleration performance

The experimental prototype is respectively under the condition of only relying on the original vehicle transmission system and the aid of the hybrid system, by the same throttle opening. The vehicle is accelerated from the speed of 0 km/h to the speed of 30 km/h, and the time and speed was recorded and cleared up as shown in Figure 17.

As seen from the test curve the experimental prototype shifted gear twice in the process of being accelerated to the speed of 30 km/h. When only on the original vehicle transmission system, the experimental prototype took 27.2 seconds to accelerate to target speed. And under the auxiliary working condition of hybrid system, it took only 20.4 seconds for the experimental prototype to achieve the target speed, so that the vehicle's start time was significantly shortened.

4) Test and result analysis of climbing performance

In the vehicle climbing performance test, the road slope is 2.8%, and is chosen to test. By judging the maximum climbing gear of the experimental prototype in the test slope, the vehicle climbing performance could be evaluated. In the form of engine power, with the 8th gear climbing, the vehicle speed and engine speed decreased significantly, as shown in Figure 18a), the vehicle cannot fulfill speeding up and climbing; while in the hybrid system, a prototype could smoothly start and finish climbing, as shown in Figure 18(b). This shows, with the aid of a hybrid system, the power performance of experimental prototype is improved significantly.

5) Test and result analysis of fuel economy

The vehicle's fuel economy is closely related with its running condition. A specific road segment is chosen as a test road segment. Under the original transmission structure and the mode with a hybrid system, the experimental prototype was tested separately. In order to ensure that the experimental results are more representative, and to minimize the impact of traffic factors on the experimental results, the same test period is selected and used, and multiple tests are conducted. The vehicle speed and the average fuel consumption curves are shown in Figure 19. The fuel consumption recorder is installed on the experimental prototype, which is used to measure and record the vehicle's fuel consumption.

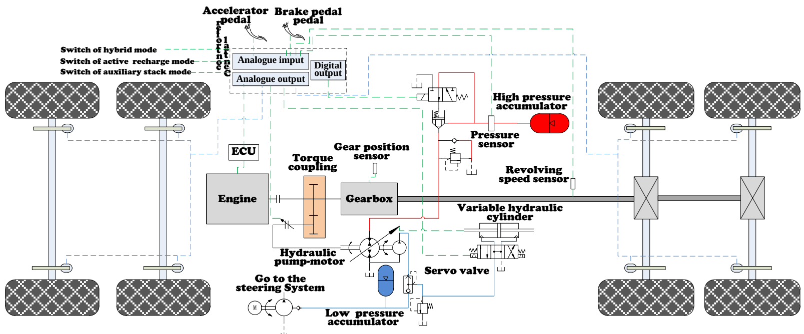


Figure 14. The Structure Diagram of the FPHHC Experimental Prototype.

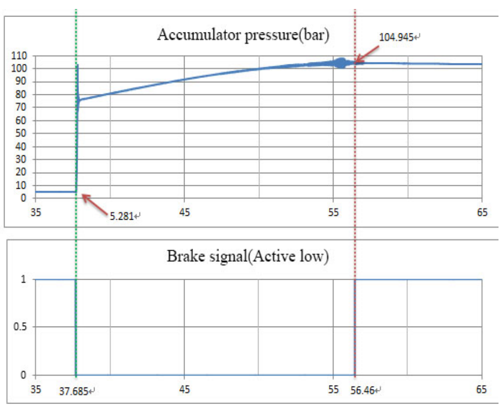


Figure 15. System Parameters Change in the Process of Energy Recovery.

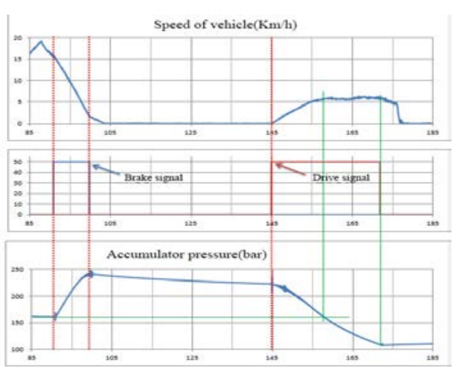


Figure 16. System Parameter Changes in the Process of Energy Utilization.

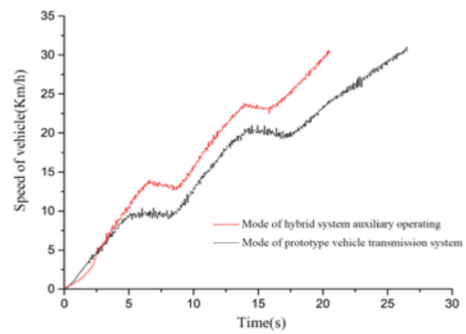
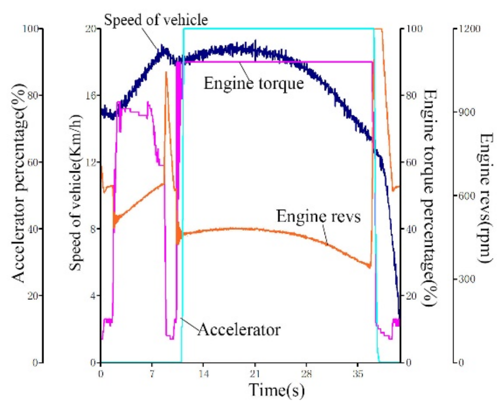
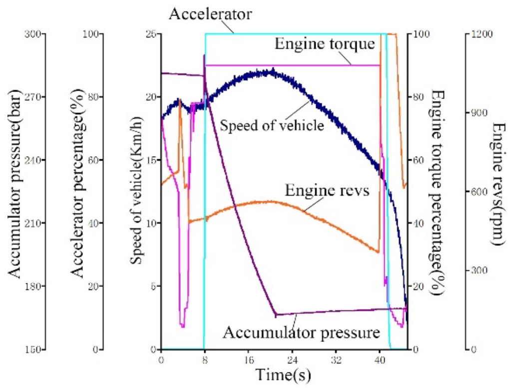


Figure 17. Speed Curves of the Vehicle is Accelerated from Idle State to the Speed of 30 Km/h.



(a) Engine Power Alone



(b) Engine Power Assisted by Hybrid System

Figure 18. The 8th Gear Climbing Situation of the Crane in the Form of Two Kinds of Power.

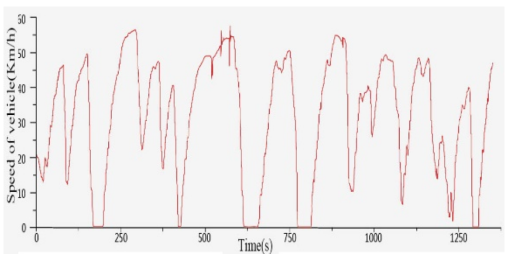


Figure 19. The Vehicle Speed.

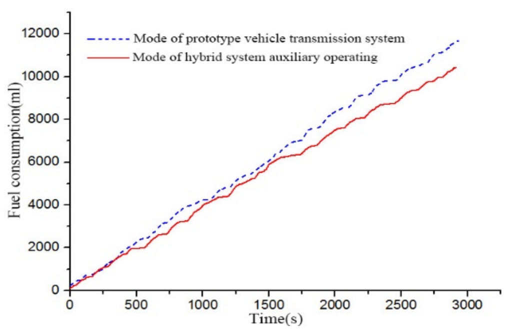


Figure 20. The Vehicle Speed and the Average Fuel Consumption in Test Conditions.

In general, the vehicle's fuel consumption is closely related to its running condition, in the frequent start-stop condition. The fuel consumption of original vehicle increased significantly, while the fuel consumption of hybrid vehicle was basically unchanged from the one when the smooth flow of traffic, so the oil-saving rate of hybrid vehicle increased significantly. According to the experimental data, it can be concluded that the average fuel efficiency of hybrid vehicle is 15.2%.

6. CONCLUSIONS

THROUGH the establishment of AMESim simulation model combined with the experimental prototype, the braking energy recovery characteristics of front-mounted parallel hydraulic hybrid crane (FPHHC) are analyzed. The theoretical basis for modification of the experimental prototype is provided through the simulation model, and the correctness of the theory model is verified by the road test of hydraulic hybrid experimental prototype, and the simulation model, which is established earlier in this article and the alignment, is good. The comparative analysis of two kinds of vehicles shows that, in the process of vehicle braking, the hydraulic hybrid system of the FPHHC can effectively recycle the braking energy of a vehicle, and the energy-saving effect is obvious. In the aspects of vehicle acceleration and braking performance, the FPHHC has an obvious improvement more than the original vehicle, which shows that the system has a certain practical significance in the aspect of energy conservation. But as a result of the leakage of cartridge valve components, the efficiency of energy regeneration efficiency is affected, so the issue should be payed attention to later. In addition, this paper's analysis method and the experimental results lay the foundation and has a certain reference value for the development and optimization of late hybrid vehicles.

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7. DISCLOSURE STATEMENT

NO potential conflict of interest was reported by the authors.

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