

Modeling and Simulation of Power Coupling System in Hydraulic Hybrid City Bus

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Abstract

In order to solve the coupling problem of power in Hydraulic Hybrid City Bus (HHB), a hydraulic hybrid power coupling system based on planetary gear transmission principle is proposed in this paper. The system consists of diesel engine, power coupler, hydraulic pump/motor, etc. The realizable operating modes of power coupling system are analyzed in this paper. Under coordination of clutches, the engine driven mode, hydraulic driven mode, hybrid driven mode, hydraulic engine-start mode and braking energy recovery mode are realizable. Based on Lagrange equation, kinetic analysis and kinematics analyses are presented. In addition, the simulation model of the power coupling system is proposed, which includes diesel engine model, power coupler model, hydraulic pump/motor model, etc. The example simulation analysis is proposed under the hybrid driven mode; the results show that the power coupling system proposed in this paper can realize power coupling of hydraulic hybrid city bus. Compared with traditional city bus, the hydraulic hybrid city bus can choose small-displacement engine so as to improve fuel economy and dynamic property.

Keywords

HHB; Power Coupling System, Planetary Gear Trans, Hydraulic Starting

1. Introduction

The hydraulic hybrid utilizes a high pressure accumulator for energy storage which has more superior power density than conventional battery technology. This makes fluid power attractive for urban driving applications in which there are frequent starts and stops and large startup power demands [1]. The hydraulic hybrid city bus has great application value, and this is an effective way for energy conservation and emission reduction. The power coupling is the key technology of the hydraulic hybrid city bus and the planetary gear transmission mechanism is used most frequently.

Many scholars have researched the power coupling problem of the hydraulic hybrid bus, and some achievements were achieved. Wojnarowski, Sherman and Wojnarowski J. proposed the bond graph model of epicyclical gear transmission and planetary gear transmission. The methodology of gear system kinetic analysis and kinematics analysis are developed based on graph theory [2]-[4]. Guoqi He and L. W. Tsai proposed power coupling system of hybrid bus based on planetary gear transmission, and the lever analogy was used to analyze the speed characteristic and torque characteristic of power coupling system [5] [6]. The results show that many operating modes of hybrid bus can be realized by planetary gear transmission system.

A power coupling system of hydraulic hybrid bus is proposed in this paper, based on epicyclical gear transmission. Many operating modes of hydraulic hybrid city bus were realized by cooperation of many clutches. The power coupling system was used on the Yu Tong ZK6126HG city bus, and the coupling property under hybrid driven mode was analyzed. Diesel engine model, hydraulic accumulator model and hydraulic pump/motor model were established; in addition, the mathematic model of coupler based on Lagrange equation was established. All the models were simulated under Matlab/Simulink simulation environment. The results show that the power coupling system can solve the power coupling problem of the hydraulic hybrid bus effectively, and the vehicle dynamic property improved obviously under the same fuel consumption.

2. Design of Power Coupling System

This paper studies a power coupling system applied to hydraulic hybrid city bus. **Figure 1** shows the schematic diagram of power coupling system. The power coupling system is made up of engine and epicyclical gear transmission integrated with a hydraulic motor that can also function as a hydraulic pump. In the figure, the power coupler is set between engine crankshaft and engine flywheel. The engine crankshaft is connected to the sun gear of power coupler, and the sun gear can be integrated or separated with planet carrier by clutch C1. The output shaft of hydraulic motor is connected to the gear ring of power coupler by clutch C2. The output shaft of power coupler or the output shaft of planet carrier of power coupler is connected to the transmission of the vehicle. In follows, the component models are built and the characteristic of the power coupling system is analyzed.

3. Operating Modes of Power Coupling System

The hydraulic hybrid city bus drives on certain mode based on different driving cycle, and realizes energy conservation and emission reduction with braking energy regeneration.

3.1. Engine Driven Mode

To launch the vehicle from a standstill and for driving in city traffic, the clutch C1 is engaged and the clutch C2 is disengaged. At this time, the sun gear integrated with planet carrier, and the transmission ratio between engine and transmission is one. The engine is used to drive the city bus, individually, and the dynamic property of the city bus is not influenced compared with the traditional city bus. The power transmission line is shown in **Figure 2**. In the follow figures of this paper, the clutch of black color indicates that the clutch is engaged and the clutch of white color indicates that the clutch is disengaged.

3.2. Hydraulic Driven Mode

Hydraulic accumulator has the advantage of higher power density and the ability to accept the high rates and high frequencies of charging and discharging, both of which are not favorable for batteries [7]. As a result, the hydraulic hybrid city bus stores the braking energy in the hydraulic accumulator as a form of hydraulic energy. When the hydraulic hybrid city bus works on the starting or climbing conditions, the hydraulic energy is translated to the kinetic energy of the bus to reach the purpose of energy conservation and emission reduction. **Figure 3** shows the power transmission line of hydraulic driven mode.

Under hydraulic driven mode, both clutch C1 and clutch C2 are engaged, the energy of hydraulic accumulator is translated into mechanical energy of hydraulic motor, and the hydraulic hybrid city bus is droved by the hydraulic motor. At the same time, the engine crankshaft also be droved, the engine is started at the end of releasing of hydraulic accumulator energy. The clutch C2 is disengaged after the engine is started, and the working mode turns into the engine driven mode as mentioned above. The hydraulic driven mode adopts the low speed

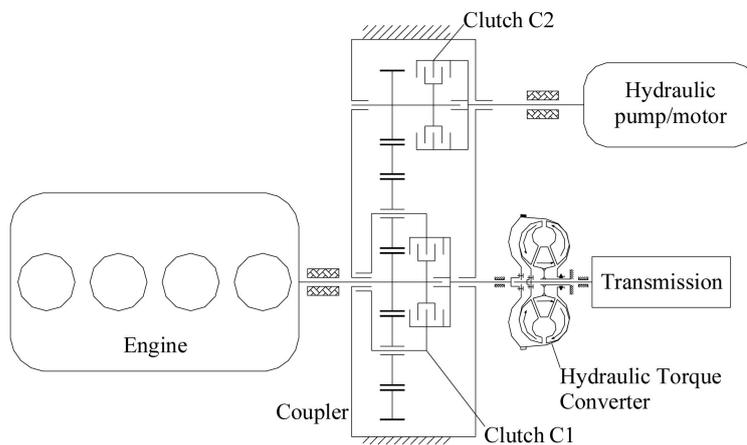


Figure 1. The schematic diagram of power coupling system.

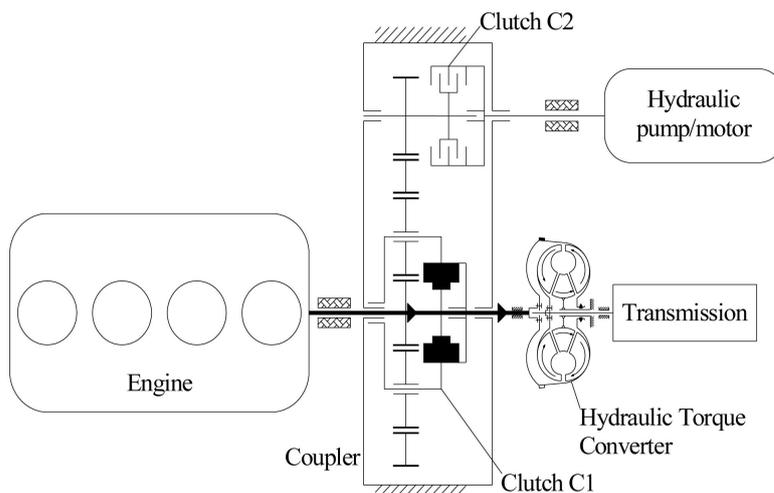


Figure 2. Power transmission line of engine driven mode.

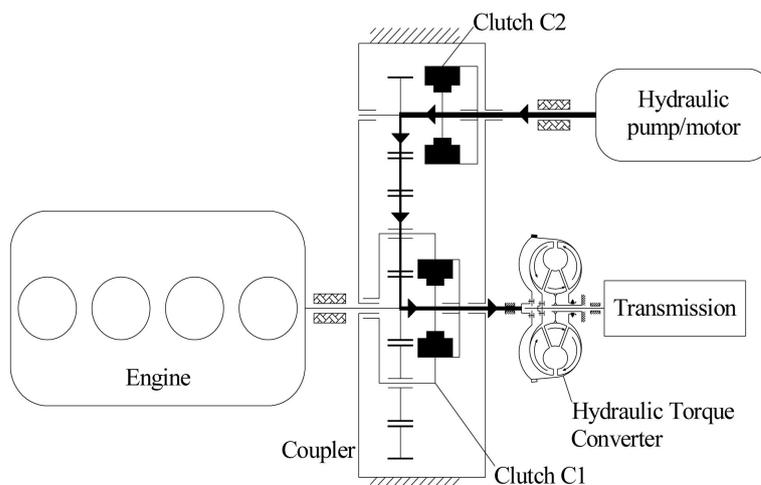


Figure 3. Power transmission line of hydraulic driven mode.

high torque characteristic of hydraulic motor, and the engine working conditions of light load, low efficiency and high emissions are avoided [8]. In addition, the purpose of the engine always working on the economy re-

gion is reached.

3.3. Hybrid Driven Mode

Simpson planet gear mechanism belongs to 2K-H type epicyclical gear transmission, which has two degree of freedom when all elements are flexible [9]. Two input torques can be synthesized to one output torque at advantage of this feature. If the hydraulic hybrid city bus running in hybrid driven mode, clutch C2 should be engaged while clutch C1 should be disengaged. The power transmission line of hybrid driven mode is shown in **Figure 4**. The engine power and the hydraulic motor power synthesized in the power coupler, and then output to transmission to drive the city bus. Hydraulic energy storage system features high frequency response, high power density, simple system structure, compatible technology and remarkable energy-saving and emissions reduction effect [10]-[13]. Under the working condition of starting or climbing, some auxiliary power can be added to the hydraulic hybrid city bus to improve the dynamic property, and ensure that the engine works in the economy region, which is helpful to reduce fuel consumption and emission.

3.4. Hybrid Driven Mode

Two engine hydraulic start modes are proposed by the power coupling system: standstill hydraulic start and dynamic hydraulic start. The standstill hydraulic start is that the bus hydraulically starts when the hydraulic accumulator pressure is low. The hydraulic energy is released from the hydraulic accumulator and translated to mechanical energy by hydraulic motor, and then drive the engine to idling speed. The dynamic hydraulic start works when the hydraulic accumulator pressure is high. At this time, both clutch C1 and C2 should be engaged, and the transmission is in a non-neutral state. The city bus and engine are droved by hydraulic motor in the meantime. Clutch C2 is disengaged, and the engine is started by the inertia of the city bus, when the end of the hydraulic energy release. The power transmission line of engine hydraulic start mode is shown in **Figure 5**.

The engine hydraulic start mode reduces fuel consumption and emission during the engine start, and then the engine works in an economy region. Compared with the traditional way of engine start, higher torque can be output, and engine cannot be influenced by temperature at the engine hydraulic start mode.

3.5. Braking Energy Recovery Mode

Hydraulic braking force and mechanical friction braking force can be distributed reasonably, depending on the demand of braking intensity of power coupling system.

Under the condition of low braking intensity, clutch C2 is engaged and clutch C1 is disengaged. The hydraulic braking works alone, all of the braking energy is recovered into the hydraulic accumulator.

Under the condition of middle braking intensity, clutch C2 is engaged and clutch C1 is disengaged. The hydraulic braking and the mechanical braking combine to work, and most of the kinetic energy is recovered into the hydraulic accumulator. This mode can not only recycle the braking energy but also reduce the braking intensity, which prolongs the life of the braking. **Figure 6** shows the power transmission line of braking energy recovery mode under low and middle braking intensity.

Under the condition of high braking intensity, both clutch C1 and clutch C2 are engaged. Hydraulic pump, engine and mechanical braking work to decelerate the city bus. The power transmission line of braking energy recovery mode under high braking intensity is shown in **Figure 7**. When the bus is driving on slippery roads or steep roads, the braking energy recovery mode can avoid using the mechanical braking frequently [14]. Obviously, this mode is helpful to prolong the life of braking discs, slow down heat fade and water fade, and improve the braking efficiency and safety of the city bus.

4. Characteristic Analysis of Power Coupling System

4.1. Kinematics Analysis of Power Coupling System

1) Speed calculation of planet carrier of the power coupler

The conversion of the epicyclical gear train is the ordinary fixed axis gear train, and the method solving transmission ratio of fixed axis gear train can be used to calculate transmission ratio of epicyclical gear train [15]. According to the transmission ratio equation of the epicyclical gear train, the transmission ratio of the gear

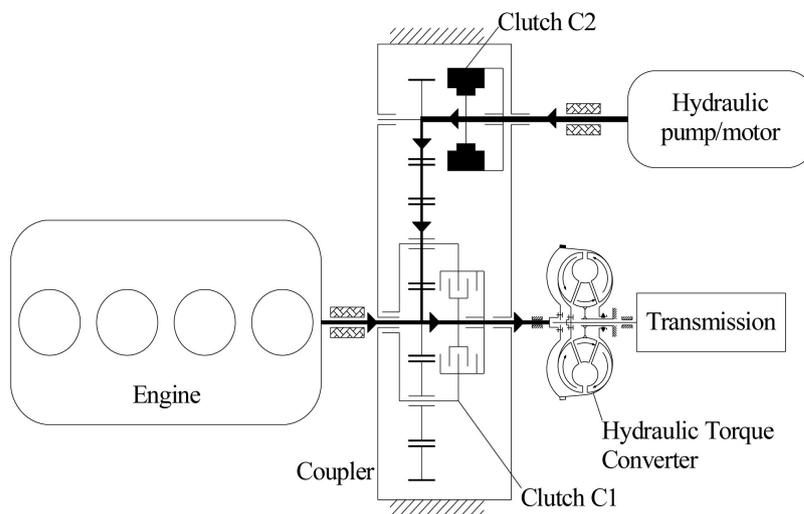


Figure 4. Power transmission line of hybrid driven mode.

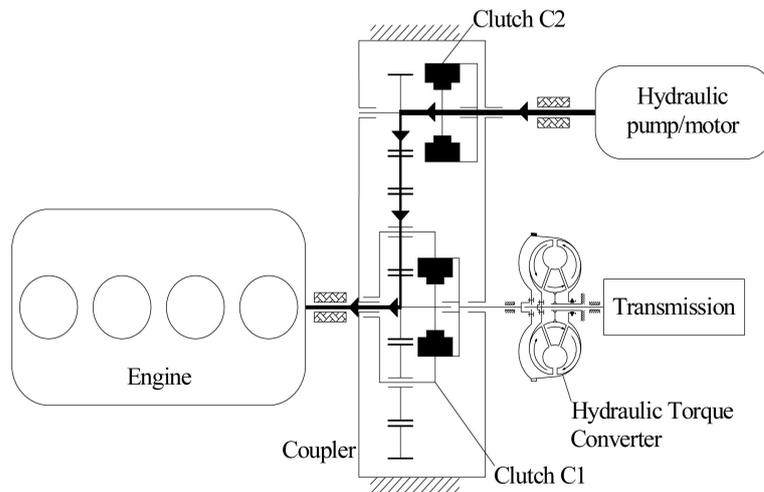


Figure 5. Power transmission line of engine hydraulic start mode.

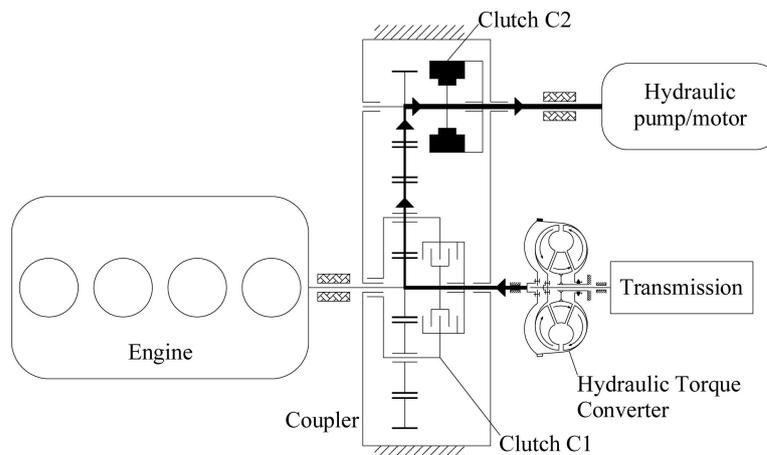


Figure 6. Power transmission line of braking energy recovery mode under low and middle braking intensity.

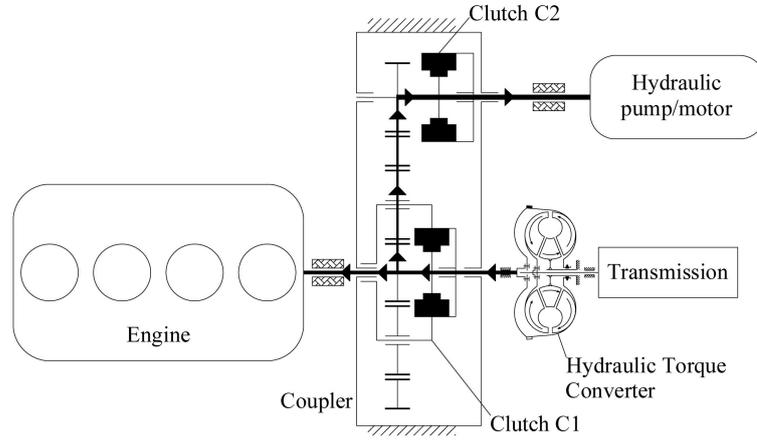


Figure 7. Power transmission line of braking energy recovery mode under high braking intensity.

ring and the sun gear is shown as Equation (1).

$$\frac{n_s^H}{n_r^H} = \frac{n_s - n_H}{n_r - n_H} = -\beta \quad (1)$$

where β is the transmission ratio between sun gear and ring gear, relative to the planet carrier; n_s^H is the speed of sun gear, relative to the planet carrier; n_r^H is the speed of gear ring, relative to the planet carrier; n_r is the actual speed of gear ring; n_s is the actual speed of sun gear; and n_H is the actual speed of planet carrier.

From Equation (1), we have

$$\omega_s R_s + \beta \omega_r R_r - (1 + \beta) \omega_H R_H = 0 \quad (2)$$

where ω_s is the angular speed of sun gear; ω_r is the angular speed of gear ring; ω_H is the angular speed of planet carrier; R_H is the radius of planet carrier; R_s is the radius of sun gear; and R_r is the radius of gear ring.

From Equation (2), we have the angular speed of planet carrier of the power coupler:

$$\omega_H = \frac{R_s}{(1 + \beta) R_H} \omega_s + \frac{\beta R_r}{(1 + \beta) R_H} \omega_r \quad (3)$$

2) Speed calculation of planet gear of the power coupler

As shown in **Figure 8**, assume the gear ring of the power coupler holds still, calculate and analyze the relation between the sun gear and the planet gear.

The motion of any point in the plane graphics can be regard as synthesis of the two movements, and the speed of any point can be calculated depend on speed synthesis theorem [16]. Angular speed is a vector and the definition of plus and minus are described below: from the positive end of the shaft, if the rigid body is turning counterclockwise, the angular speed takes the positive value, otherwise, the negative value. Assume A as the basic point, the speed relation between the sun gear and the planet gear is:

$$u_C = u_A + u_{CA} \quad (4)$$

where μ_C and μ_A is the linear speed of point C and A , respectively; μ_{CA} is the speed of point C relative to point A .

From Equation (4), we have the angular speed relation of sun gear, planet carrier and planet gear:

$$\omega_s \cdot R_s = \omega_H \cdot R_H - \omega_p \cdot R_p \quad (5)$$

where ω_p is the angular speed of planet gear; R_p is the radius of planet gear.

As shown in **Figure 9**, assume the sun gear of the power coupler holds still; calculate the speed relation between the gear ring and the planet gear.

The sun gear of the power coupler holds still, takes A as the basic point. The speed relation between the gear ring and the planet is:

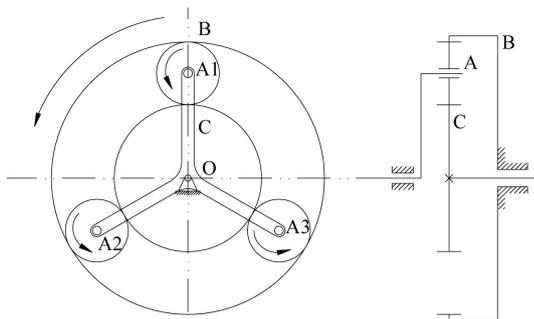


Figure 8. The schematic diagram of the sun gear driving the planet carrier.

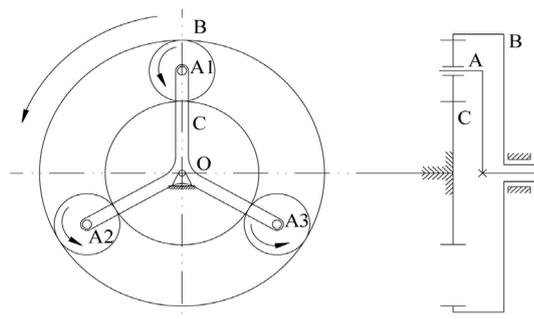


Figure 9. The schematic diagram of the gear ring driving the planet carrier.

$$u_B = u_A + u_{BA} \tag{6}$$

where μ_B is the linear speed of point B ; μ_{BA} is the speed of point B relative to point A .

From Equation (6), the angular speed relation of gear ring, planet carrier and planet gear is:

$$\omega_r \cdot R_r = \omega_H \cdot R_H + \omega_p \cdot R_p \tag{7}$$

From Equation (5) and Equation (7), the angular speed of planet gear is:

$$\omega_p = \frac{R_r}{2R_p} \omega_r - \frac{R_s}{2R_p} \omega_s \tag{8}$$

4.2. Kinetic Analysis of Power Coupling System

Take the angular displacement of crankshaft (θ_s) and the angular displacement of the hydraulic motor output shaft (θ_r) as the generalized coordinates. The relative angular speed is the generalized angular speed, that is $\dot{\theta}_s = \omega_s$, $\dot{\theta}_r = \omega_r$. The kinetic energy of the power coupling system is:

$$E = \frac{1}{2} J_s \dot{\theta}_s^2 + \frac{1}{2} J_H \dot{\theta}_H^2 + \frac{1}{2} J_r \dot{\theta}_r^2 + \frac{1}{2} J_p \dot{\theta}_p^2 + \frac{3}{2} m_p \cdot R_H^2 \cdot \omega_H^2 \tag{9}$$

where J_s is the moment of inertia of the engine output shaft and its pontes; J_H is the moment of inertia of the transmission output shaft and its pontes; J_r is the moment of inertia of the pump motor output shaft and its pontes; J_p is the moment of inertia of the planet wheel of the power coupling system; m_p is the mass of the planet gear.

From Equation (3), (8), (9), we have Equation (10):

$$E = \frac{1}{2} J_{11} \omega_s^2 + J_{12} \omega_s \omega_r + \frac{1}{2} J_{22} \omega_r^2 \tag{10}$$

where

$$J_{11} = J_s + J_H \left[\frac{R_s}{(1+\beta)R_H} \right]^2 + J_p \left(\frac{R_s}{2R_p} \right)^2 + 3m_p R_H^2 \left[\frac{R_s}{(1+\beta)R_H} \right]^2 \quad (11)$$

$$J_{12} = J_H \frac{\beta R_s R_r}{[(1+\beta)R_H]^2} - J_p \frac{R_s R_r}{(2R_p)^2} + 3m_p R_H^2 \frac{\beta R_s R_r}{[(1+\beta)R_H]^2} \quad (12)$$

$$J_{22} = J_r + J_H \left[\frac{\beta R_r}{(1+\beta)R_H} \right]^2 + J_p \left(\frac{R_r}{2R_p} \right)^2 + 3m_p R_H^2 \left[\frac{\beta R_r}{(1+\beta)R_H} \right]^2 \quad (13)$$

The generalized force corresponded to θ_s and θ_r is torque, the power of all external force is:

$$P = T_s \cdot \omega_s + T_H \cdot \omega_H + T_r \cdot \omega_r \quad (14)$$

where T_s is the output torque of engine; T_r is the output torque of hydraulic motor; T_H is the input torque of transmission.

From Equation (3) and Equation (14), the power of external force is:

$$P = \left(T_s + \frac{R_s \cdot T_H}{(1+\beta)R_H} \right) \cdot \omega_s + \left(T_r + \frac{\beta R_r T_H}{(1+\beta)R_H} \right) \cdot \omega_r \quad (15)$$

The generalized force corresponded to θ_s and θ_r is:

$$Q_r = T_r + \frac{\beta R_r}{(1+\beta)R_H} T_H \quad (16)$$

$$Q_s = T_s + \frac{R_s}{(1+\beta)R_H} T_H \quad (17)$$

5. Mathematical Model of Power Coupling System

5.1. Model of the Power Coupler

The Lagrange equation [17], named after Joseph Louis Lagrange, is an important equation of Lagrange mechanics. The Lagrange equation can be used to describe the movement of object. Put the generalized force into the Lagrange equation, we have:

$$\begin{bmatrix} J_{11} & J_{12} \\ J_{12} & J_{22} \end{bmatrix} \begin{bmatrix} \ddot{\theta}_s \\ \ddot{\theta}_r \end{bmatrix} = \begin{bmatrix} Q_s \\ Q_r \end{bmatrix} \quad (18)$$

Aliased as

$$\mathbf{J}\mathbf{a} = \mathbf{T} \quad (19)$$

where \mathbf{J} is the moment of inertia of the components of the planet gear; \mathbf{a} is the acceleration matrix of the sun gear and the gear ring; \mathbf{T} is the generalized force matrix.

Equation (19) is the model of the power coupler, solve Equation (19), we have the relation of the engine torque, the hydraulic motor torque and the output torque of the power coupler.

From Equation (5) and (7), the coupling model of the power coupler is:

$$2\omega_H R_H = \omega_s R_s + \omega_r R_r \quad (20)$$

5.2. Diesel Engine Model

In this work, the engine simulation model is simplified to accommodate the need of top-level control strategy design. A lookup table of engine brake torque versus engine speed is directly derived from the engine brake specific fuel consumption map (Figure 10), which was generated from experimental data obtained in the Hybrid

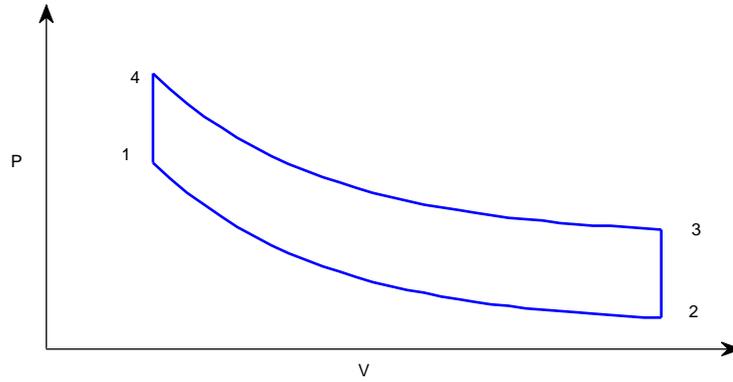


Figure 11. The working cycle of the hydraulic accumulator thermodynamics process.

accumulators, so we should assume that the compression and expansion of gas is adiabatic. The process 2 to 3 and the process 4 to 1 are the isometric process. Gas temperature returns to ambient temperature. The dynamic accumulator model is based on the following equation:

$$pV^n = C \quad (24)$$

Here, p is supply pressure; V is gas volume; n is polytrophic index within a range between 1 (isothermal process) and 1.4 (adiabatic process); C is constant and depends on pre-charge pressure.

Derivative the Equation (24) on both sides, we get:

$$\frac{dV}{dt} = -\frac{V^n}{npV^{n-1}} \cdot \frac{dp}{dt} \quad (25)$$

From the Equation (25) and Equation (26) we get:

$$\frac{dV}{dt} = -\frac{C^{1/n}}{n} \cdot \frac{1}{p^{1+1/n}} \cdot \frac{dp}{dt} \quad (26)$$

where, the minus indicates that when the pressure of hydraulic accumulator increases, the gas volume in the inner bladder of the hydraulic accumulator decreases.

For a hydraulic accumulator, if we ignore the change of the quality and the volume of the hydraulic oil, the fluid volume that into and out of hydraulic accumulator equal to the change rate of the gas volume of bladder in the hydraulic accumulator. We get:

$$Q_a = \frac{dV}{dt} \quad (27)$$

6. Simulation and Analysis of the Results

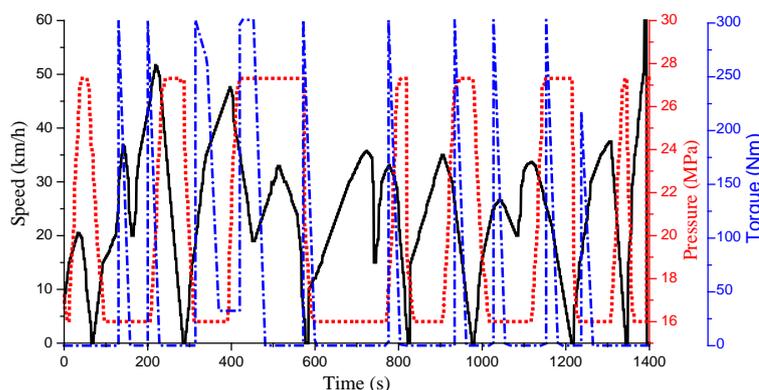
The simulation model can be established according to the mathematical model of the power coupling system under Matlab/Simulink. Copy relative functional modules to the simulation environment; connect each block diagram according to the flow direction of the signal, create a signal source, and then assign simulation results to relative variables, and finally draw curves in workspace.

In the simulation model, the input torque of the power coupler is the engine output torque under China typical city bus driving cycle. To keep the high efficiency, the solver is the key. Variable step size 4 order 5 order Runge-Kutta algorithm (ode 45) is employed according to the simulation model of the coupler. The parameters of the test vehicle are shown in [Table 1](#).

[Figure 12](#) shows China typical city bus driving cycle, which is established depending on Recommended Practice for Measuring Fuel Economy and Emissions of Hybrid-Electric and Conventional Heavy Duty Vehicles (SAEJ2711). The pressure change of hydraulic accumulator under China typical city bus driving cycle, and the hydraulic motor output torque while the releasing its energy under China typical city bus driving cycle are also

Table 1. Parameters of the test hydraulic hybrid city bus.

Engine parameters	Rated speed/r·min ⁻¹	2300	Hydraulic pump/motor parameters	Flow /L·min ⁻¹	266
	Rated power/kW	185		Displacement/ml	160
	Maximum torque/Nm	1000		Maximum torque/Nm	889
Vehicle parameters	Loaded quality/kg	165,00	Hydraulic accumulator parameters	Capacity/L	96
	Frontal area/m ²	6		Rated pressure/MPa	16
	Tire radius/m	0.51		Maximum pressure/MPa	31.5

**Figure 12.** Driving cycle curve, accumulator pressure curve, and accumulator pressure curve.

should in the **Figure 12**. From the curves, we know that hydraulic accumulator and hydraulic motor can supply high instantaneous torque, which can be used to improve the dynamic property while the hydraulic hybrid city bus under the conditions of acceleration and climbing.

Figure 13 shows the desired torque. The torque coupling property, speed coupling property and power coupling property of the power coupling system is shown in **Figure 14**, **Figure 15** and **Figure 16**, respectively. As shown in the curves, the power coupling system, proposed in this paper, can solve the coupled problem of the hydraulic hybrid city bus effectively. The low-powered engine can be taken while the hydraulic motor is the auxiliary power, then the engine work in the economic fuel consumption region to improve the fuel economy and emission performance.

From these figures, one can see that the power coupling system played an important role when the vehicle in the climbing condition. At this time, stored accumulator energy is used to supply much of the power demand, with the engine being used to supply short bursts of power. For the certain power demand, the large displacement engine can be replaced by a little one, and the remaining power added by the hydraulic motor. Therefore, the purpose of energy conservation and emission reduction can be achieved.

Furthermore, the power of hydraulic motor is supplied by the hydraulic accumulator, when the accumulator at the low energy, all power should be present by engine, and there will be high fuel consumption. To solve this problem, a cooperation strategy between engine and hydraulic motor should be designed.

7. Conclusions

1) This paper proposed a power coupling system of the hydraulic hybrid city bus, and analyzed the operating modes of the system.

2) Kinetic analysis and kinematics analysis of the power coupler under hybrid driven mode were presented. The models of engine, hydraulic accumulator and hydraulic pump/motor were established, and the mathematical model of power coupler was built, based on Lagrange equation.

3) Simulation was proposed under Matlab/Simulink. The results show that the power coupling system proposed in this paper could solve the coupled problem of the hydraulic bus effectively. Compared with the traditional city bus, the low-powered engine can be taken into the hydraulic hybrid city bus to improve the fuel economy and emission performance.

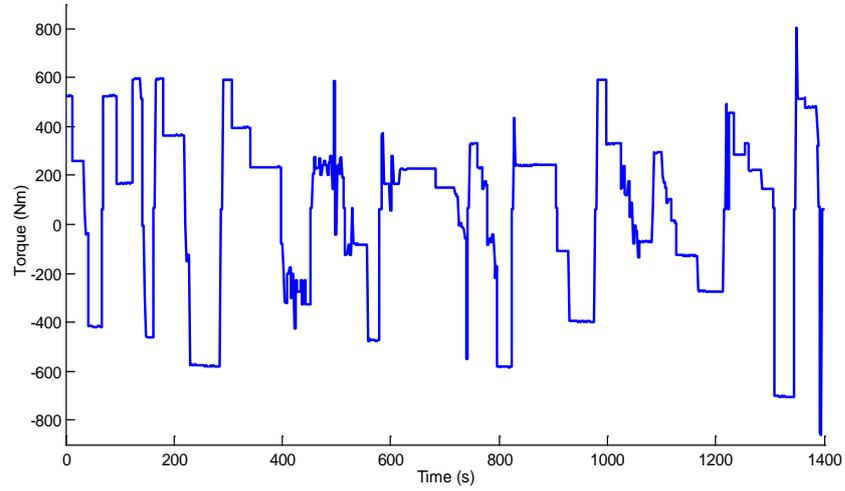


Figure 13. Vehicle demand torque curve.

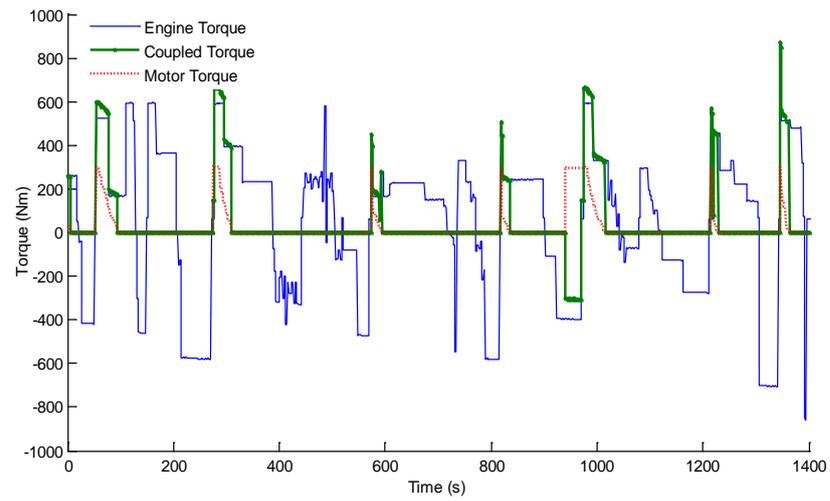


Figure 14. Coupling torque characteristic curve.

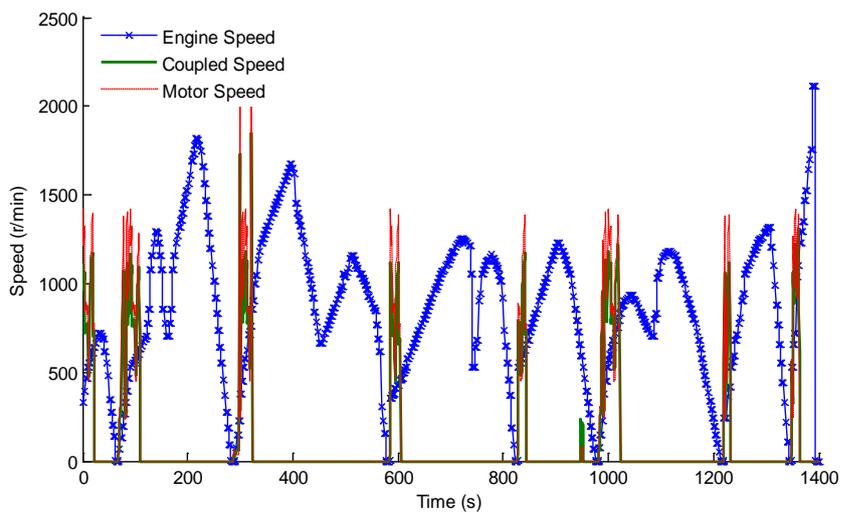


Figure 15. Speed coupling characteristic curve.

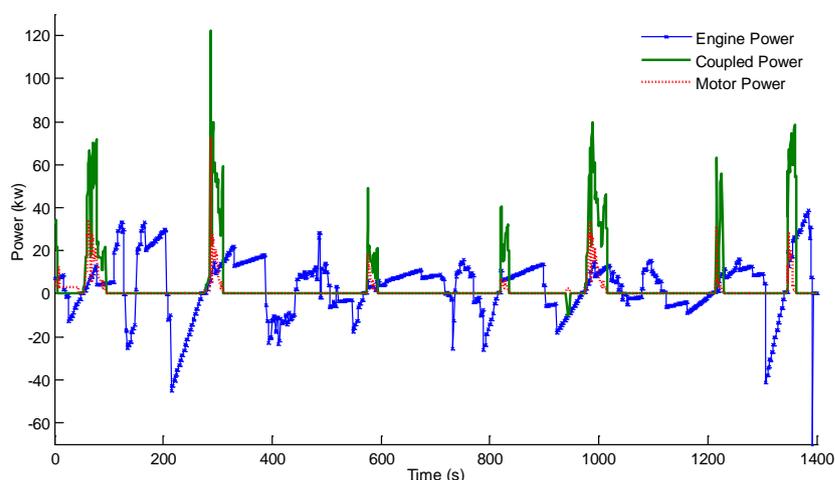


Figure 16. Power coupling characteristic curve.

Acknowledgements

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