Optimal Design for a Hydraulic Regeneration Propel System of the Hydraulic Hybrid Bus

LIU GUO-QING[1*], CHEN JIE[1], XU FANG-FANG[2]
1. Institute of Automotive Engineering, Shanghai Jiao Tong University
2. Changzhou College of Technology
*No.800 Dongchuan Road, Shanghai
CHINA
* guoqingliu@sjtu.edu.cn or xff122@163.com

Abstract: Hydraulic regeneration propel system (HRPS) has been proven to be able to enhance the fuel economy and reduce the emission by regenerating the braking energy when it is installed in the existing conventional bus. The rebuilt bus is normally termed as the Hydraulic Hybrid Bus (HHB). This paper aims at exploiting full potential of the HRPS, so that the HHB is able to acquire minimum fuel consumption and maximum dynamic performance with minimum cost. The ideal point and weight square sum method is used to construct the evaluate function so as to consider the tradeoff among the objectives. The generation algorithm (GA) is used to solve the multi-objective optimization. The results of the optimization show that the improving of the dynamic performance combined with reduced cost and equivalent fuel-saving is possible. The optimal designing approach presented by this paper provides the theoretical directions for the design of HRPS for a HHB.

Key Words: Regeneration, HRPS, HHB, Fuel economy, Multi-objective optimization, GA

1 Introduction

Although the internal combustion engine remains the dominant prime mover for technological and cost reasons, hybrid systems have been under consideration for some time and have become available recently in commercial products [1-7]. Hydraulic hybrid powertrain(one branch of the hybrid systems) is a critical technology to improve vehicle fuel economy in frequent stop-and-start driving cycles such as urban conditions for the high power density character. The hydraulic hybrid powertrain can be divided into two key configurations: series and parallel, and the parallel one is more attractive to the urban buses [8]. The parallel hydraulic hybrid powertrain was previously demonstrated on buses in Japan and US in late 1980s and early 1990s, currently being developed by Eaton and Permo-Drive. However, in most cases, engine is downsized and the drive line is reconfigured.

There are a great number of buses above 350,000 in China, with large contribution to fuel consumption and emissions. An applied regeneration system is critical to improve the fuel economy of the in-use bus with little reconfiguration of the original driveline. In this paper a hydraulic regeneration propel system (HRPS) is presented to meet the demand.

This paper proposes a methodology for designing a new HRPS to a typical conventional bus, which aims at exploiting full potential of the HRPS, so that the HHB is able to acquire minimum fuel consumption and maximum dynamic performance with minimum cost. The ideal point and weight square sum method is used to construct the evaluate function so as to consider the tradeoff among the objectives. The accumulator volume and the pump/motor cubage are determined as the variable parameters in the optimization process. The generation algorithm (GA) is used to solve the multi-objective and multi-parameter optimization problem.

Realistic assessment of vehicle fuel economy depends critically on driving conditions [9,10], so the typical urban driving schedule in China (TUDS) is proposed for the fuel economy study, the gear shifting rule is based on the driving habit for obtaining more exact results. As to the dynamic performance, the critical criterion is also extracted from the TUDS in that the dynamic performance is decided by the accumulator pressure which is related to the driving cycle. The cost model is built on the data offered by the manufacturers. All the vehicle parameters in the study come from a conventional bus in China.

The configuration and the control strategy are described in section 2. The simulation model is built and validated in section 3. Section 4 addresses the optimization process and the results analysis. The conclusion is drawn in section 5.

2 Configuration and Control Strategy

2.1 HRPS configuration

HRPS regenerates the braking energy to save the fuel consumption, the configuration is shown in Fig.1. A hydraulic pump/motor, linked to the second shaft of the transmission through the power take-off window by HRPS clutch and transfer case. High-pressure and low-pressure accumulator both connect with the hydraulic pump/motor. Engine clutch and HRPS clutch, controlled by a controller unit, engage or disengage engine and pump/motor to select bus power split. In Fig.1, '-' means power transmission, '-.-' means signals links and '--' means hydraulic transporting. It can be seen that when the HRPS clutch is separated, the rebuilt vehicle has little difference from the conventional one.

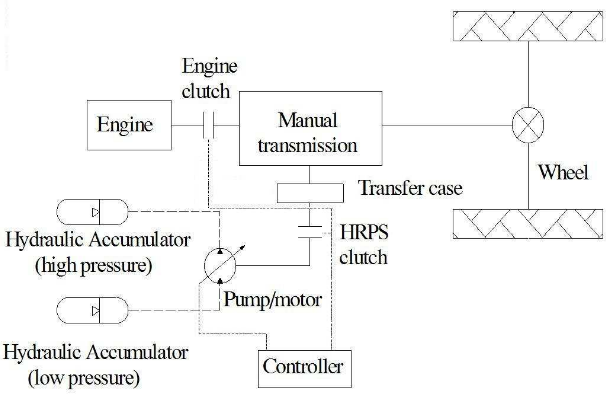

Fig.1 HRPS configuration coupled to a conventional vehicle

The HRPS operates as follows: (a) in the course of decelerating or braking, the pump/motor (pump mode) pushes the hydraulic fluid from the low-pressure accumulator to the high-pressure accumulator converting kinetic energy of the vehicle to hydraulic energy stored in the high-pressure accumulator; (b) returning the stored energy to the vehicle to accelerate the vehicle by pushing the hydraulic fluid in the reverse direction through the pump/motor (motor mode) in the propelling stage. Fig.2 shows the bidirectional power transfer route: the motor mode route is from the left to the right, the reverse is the pump mode.

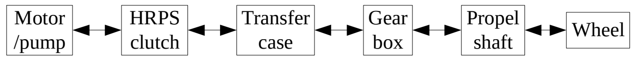

Fig.2 Power transfer route

Table 1 shows the key parameters of the conventional vehicle and the HRPS. The diesel engine speed performance is shown in Fig.3.

Table 1 Key parameters of the conventional vehicle and HRPS

Parameter	Value or Model
Gross weight m	15500kg
Tire	11R22.5
Gearbox gear ratio i_i	6.9/3.93/2.32/1.49/1
Final gear ratio i_f	4.84
Transfer case gear ratio i_t	1.66
Diesel engine	V6,6.5,170kW @2500r/min
Accumulator volume V	100L
Pump/motor cubage q	100cc/rev

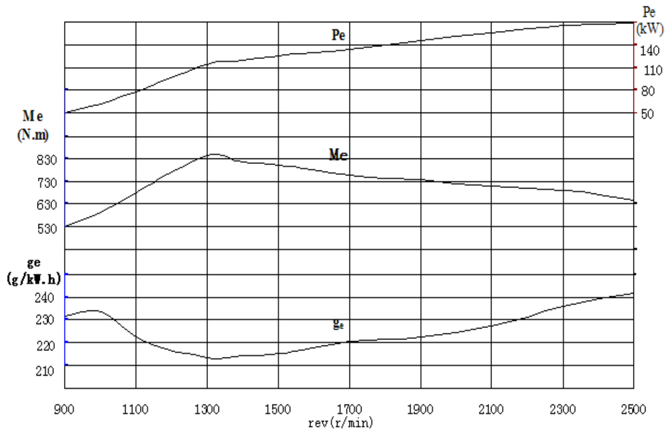

Fig.3 Engine speed performance
(Where rev, engine speed; ge, the ratio of engine fuel consumption; Me, the engine torque; Pe, the engine power)

2.2 Control strategy

Control strategy affects the system potential and the operation scheme. Fig.4 shows a conventional urban bus 0-50 km/h accelerating process and the corresponding computed fuel consumption history. In Fig.4, the red '-' curve is the vehicle accelerating history and the blue '--' curve shows the corresponding computed fuel consumption per 100km. The results state that the computed fuel consumption per 100km at the launch period is six times of the one in the succedent accelerating period. That is to say solving starting process is the key to save fuel in the urban traffic [10].

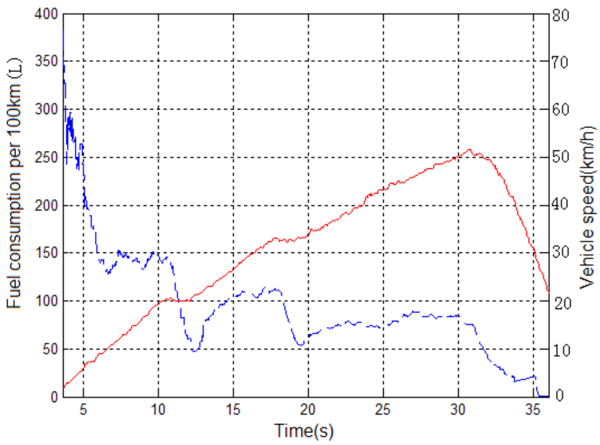

Fig.4 Conventional urban bus accelerating process and the corresponding computed fuel consumption history

Before discussing the control strategy for the HRPS, define a term"SOC" first. SOC is the state of charge of the accumulator and is defined as the ratio of instantaneous fluid volume in the accumulator of the over maximum fluid capacity, thus SOC=0 corresponds to the accumulator being empty, and SOC =1 to the accumulator being full [10]. The control rules for the HRPS are divided into two parts, i.e. the driving mode and the braking mode.

The rules for the driving mode ($T_{demand} > 0$):

IF SOC $> x$, $v < 30$km/h and $T_{motor} \geq T_{demand}$,

$$T_{motor} = T_{demand}$$

Else,

$$T_{engine} = T_{demand}$$

The rules for the braking mode ($T_{demand} < 0$):

IF SOC < 1,

$$T_{pump} = -min(-T_{demand}, T_{pump\ max})$$

$$T_{friction} = T_{demand} - T_{pump}$$

Else,

$$T_{friction} = T_{demand}, T_{pump} = 0$$

where x , a parameter that is relative to the accumulator pressure when the other parameters are settled; v , vehicle speed; T , the symbol of the torque, and the subscript shows the source of the torque, for example, T_{pump} is the pump torque and T_{demand} is the vehicle torque demand.

To the driving mode, only the motor launching the vehicle if SOC satisfies the torque requirement at lower vehicle speed ensures that the motor frequently operates at high load; The low speed limit 30km/h is based on the vehicle gear shifting rule listed in table 2, the control strategy ensues that the HRPS operates at II and III gearshift so that the engine generally falls on the better work region. What is more important is that the control strategy avoids the dynamic coupling of the two power source, which significantly simplifies the control process.

Table 2 Vehicle speed versus gearshift

Gearshift	Speed scope(km/h)
II	0~20
III	20~30
IV	30~40
V	>40

To the braking mode, the control rule ensures the pump is the preference in the braking stage. If the pump can't satisfy the brake demand, the friction brake device will act to ensure the vehicle safety.

In the control strategy, charging the hydraulic accumulator with engine power is prohibited due to the low energy density characteristic of the accumulator and the SOC of the accumulator can be allowed to vary from 1 to 0.

3 Simulation Model and Validation

3.1 Simulation model

A simulation model is built in AMESim environment to promote the study. The model includes four blocks: the mission, the conventional vehicle, the HRPS and the controller. The conventional vehicle module includes the diesel, the engine clutch, the transmission and the vehicle body; The HRPS module includes the transfer case, the hydraulic accumulator, the hydraulic pump/motor and the valves etc. In Fig.5, the arrows show the signal transmission direction, the tags near the arrow lines show the exchanging signals between the two modules. The HRPS model consults the reference [11, 12]. In case of the conventional vehicle block, the foundation is the "Diesel Vehicle with Clutch" model included in the demo of the AMESim, the parameters are replaced by the new ones. The controller is designed according to the control strategy to harmonize the blocks.

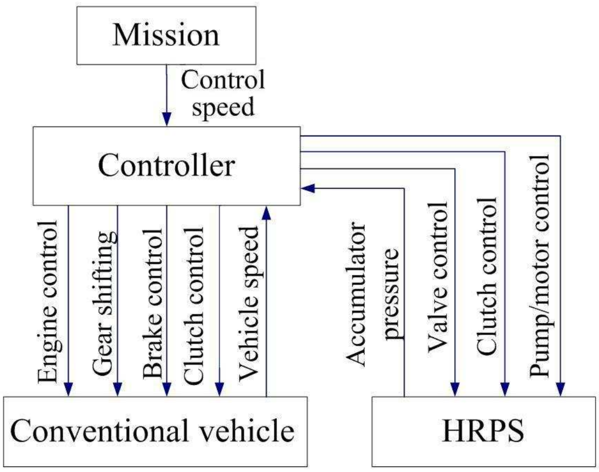

Fig. 5 Frame diagram for the simulation model

3.2 Model validation

Contrast study on the fuel economy between the test and simulation is done to validate the accuracy of the simulation model. The operating cycle is extracted from the TUDS. In the cycle, four stations are settled and the distance between stations is 400 meters. The operating rules are: accelerating the vehicle to 45km/h through changing appropriate gearshift and braking the vehicle until stop. Before launching the vehicle with HRPS at the jumping-off point, the accumulator pressure is raised to the maximum work pressure 315bar, which is to reuse the energy ahead of schedule recovered at the fourth braking period so as to acquire rational fuel saving result. Fig.6 shows the test vehicle speed and fuel economy history under two conditions: with and without HRPS, the simulation results are shown in Fig.7.

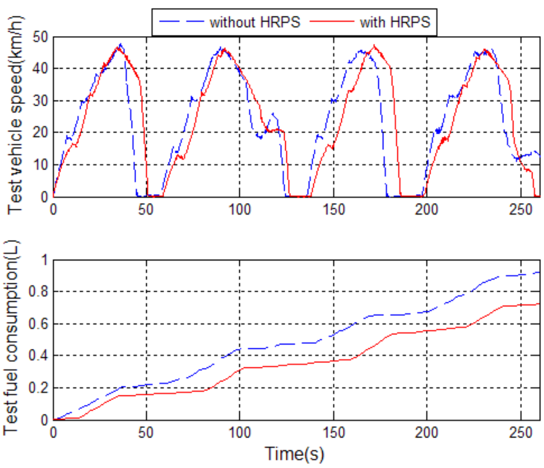

Fig.6 Test vehicle speed and fuel consumption history

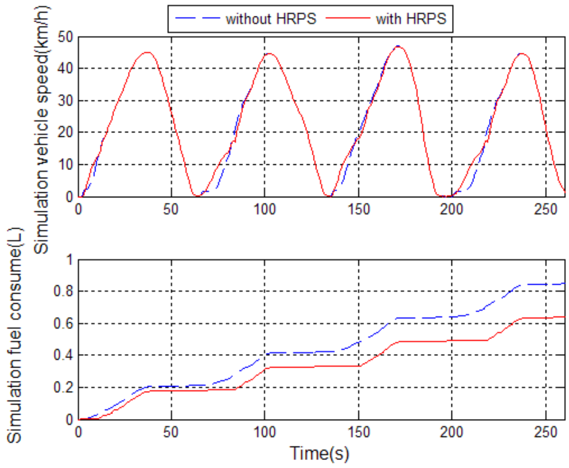

Fig.7 Simulation vehicle speed and fuel consumption history

The results show that fuel economy improvement of HRPS was tested on real road by 25% compared to the original bus. Through simulation, fuel economy improvement is obtained by 27% compared to the original bus. The results show that the model accuracy is acceptable.

4 Optimization and Analysis

The application of the HRPS offers the opportunity for engine downsizing, but it is not adopted here for two reasons: one is to prevent any adverse effect on vehicle mobility and drivability when the high pressure accumulator is empty, the other is when HRPS is installed in an in-use bus, downsizing the engine is impractical. The optimization aims at exploiting full potential of the HRPS, so that the HHB is able to acquire minimum fuel consumption and maximum dynamic performance with minimum cost. The accumulator volume and the pump/motor cubage are determined as the optimization parameters.

4.1 Evaluation function

In engineering practice, the objectives are always inconsistent when the parameters change, so to optimize the parameters, the trade off between the objectives must be considered. Ideal point and weight square sum method is used to construct the evaluation function (1),

$$U(\boldsymbol{X}) = \sum_{i=1}^{l} W_i \left[\frac{f_i(\boldsymbol{X}) - f_i^{\Delta}}{f_i^{\Delta}} \right]^2 \quad (1)$$

where $f(\boldsymbol{X})$, objective function of evaluate index; $\boldsymbol{X}=[V\ q]$; f^{Δ}, optimal value of the objective function; W_i, weight coefficient; l, number of the objective function.

The evaluation function not only makes each objective approach their ideal value as much as possible, but also reflects the importance of each objective in the multi-objective problem for the participation of the weight coefficient. To solve the evaluation function, the quantification of the evaluate indexes and the optimal value searching of the objective functions are the precondition.

4.2 Evaluate indexes and the quantification

Fuel economy, dynamic performance and the system cost are the three indexes for evaluation of HRPS.

This paper proposes to use γ, fuel saving rate, to evaluate the fuel economy of the HRPS. The γ can be expressed with equation (2),

$$\gamma=\frac{\alpha-\beta}{\alpha}\times 100\% \qquad (2)$$

where α, fuel consumption of the original vehicle; β, fuel consumption of the rebuilt vehicle with HRPS. The γ is a greatest-type index, so the maximum fuel saving rate is optimal.

The vehicle dynamic performance is always evaluated with the accelerating performance, the gradeability and the maximum speed. This paper only takes the accelerating performance of HRPS as the study object. The vehicle gradeability and the maximum speed are ensured by the original engine.

As shown in Fig.9, the accumulator pressure is various at the jumping-off point of different run-and-stop cycle under TUDS. This paper proposes using the average acceleration $\bar{a}$ at the above defined point to evaluate the HRPS dynamic performance, $\bar{a}$ is described in equation (3),

$$\bar{a}=(\frac{\bar{P}q}{2\pi}i/r-F_f)/\delta m \qquad (3)$$

where i, total transmission ratio; r, tire radius; F_f, rolling resistance; δ, vehicle rotary mass conversion coefficient; $\bar{P}$, average accumulator pressure, is obtained by equation (4),

$$\bar{P}=\frac{1}{\tau}\sum_{j=1}^{\tau}P_j \qquad (4)$$

where P_j, accumulator pressure at the jumping-off point of different run-and-stop cycle under TUDS; τ, the number of the jumping-off point under TUDS.

In terms of system costs, the low-pressure accumulator and the hydraulic oil costs vary with the change of the high-pressure accumulator; the hydraulic valves, the HRPS clutch and the hydraulic hoses costs are related to the pump/motor cubage. The system cost C_s can be expressed with equation (5),

$$C_s=C_{constant}+C_a+C_{p/m} \qquad (5)$$

where $C_{constant}$, constant part; C_a, cost that is relative to the accumulator volume; $C_{p/m}$, cost that is relative to the pump/motor cubage.

Based on the data referred by the manufactures, the empirical formula (6) can be used to express the system costs.

$$C_s=2100+20V+0.4(q-90)^2 \qquad (6)$$
$$(20\le V\le 200 \quad 90\le q\le 200)$$

4.3 Optimal value of the objective functions

In HRPS the pump/motor acts as two roles: propelling and braking the vehicle. How to design the component is a difficult problem. Fortunately, the control strategy doesn't require the HRPS to afford overfull brake torque due to the existence of the friction brake device.

The vehicle running equation:

$$F_t=F_f+F_w+F_i+F_j \qquad (7)$$

Where F_t, driving force; F_f, rolling resistance; F_w, wind resistance; F_i, gradient resistance; F_j, acceleration resistance.

Neglecting the windage resistance(which is small for the lower speed) and the road grad takes zero. Basing on the power transfer route shown in Fig.2, the cubage of the pump/motor q is restricted by the equation (8) when the motor propels the vehicle independently,

$$\frac{P_1 q i_t i_i i_f}{2\pi}=(F_f+\delta ma)r \qquad (8)$$

where P_1, minimum work pressure; i_t, transfer case gear ratio; i_i, gearshift gear ratio; i_f, final gear ratio; δ mass conversion coefficient; m, vehicle gross weight; a, maximum acceleration in TUDS; r tire radius.

And the vehicle must satisfy the road adhesive condition, shown as inequation (9),

$$F_t < F_\varphi \tag{9}$$

where F_φ, vehicle maximum adhesive force. When the motor propels the vehicle independently, the inequation (9) can be expressed as inequation(10),

$$\frac{P_2 q i_f i_i i_t}{2\pi} < mg\varphi r \tag{10}$$

where P_2, maximum work pressure; φ, vehicle adhesion coefficient;

The calculation results derived from (8) and (10) show that the pump/motor cubage should be bigger than 100 cc/rev and smaller than 253cc/rev. However, due to the restriction of the spatial arrangement, the cubage of the pump/motor must be smaller than 160cc/rev.

Hydraulic accumulator, the storage device, is characterized by much higher power density and has the ability to accept the high rates and high frequencies of charging/discharging. However, the energy density of the hydraulic accumulator is relatively low, the parameter requires carefully designed so that the fuel economy potential can be realized to its fullest. The accumulator volume V is restricted by the energy inequality (11),

$$E_a \le E_k - E_f \tag{11}$$

that is,

$$\frac{P_0 V}{n-1}\left[\left(\frac{P_2}{P_0}\right)^{\frac{n-1}{n}} - \left(\frac{P_1}{P_0}\right)^{\frac{n-1}{n}}\right] \le \frac{1}{2}mv^2 - F_f s \tag{12}$$

where E_a, energy stored in accumulator; E_k, vehicle kinetic energy; E_f, energy consumed by rolling resistance; P_0, pre-charge gas pressure; v,vehicle speed; n, polytropic index; s, braking distance.

Equation (11) states that the accumulator capability should be less than the difference between the kinetic energy and the friction energy. The highest speed of different run-and-stop cycle under TUDS varies from 16km/h to 60km/h/, ignoring the windage resistance and the road grade, the volume of the accumulator should be bigger than 14.2L and smaller than 201L. However, considering the spatial arrangement and the product in being, the accumulator volume scope is determined as 20L to 160L.

The GA is used to find the optimal value of the evaluate indexes. The optimal total fuel consumption is 1.12L when the accumulator and the pump/motor respectively take 160L and 160cc/rev. Comparing with the original fuel consumption of 1.82L, the optimal value of the fuel saving rate is 38.4%. As far as the dynamic performance is concerned，the optimal $\overline{a}$ is 2.71m/s^2 when the accumulator and the pump/motor respectively take 60L and 160cc/rev. To the system cost index, the optimal value can be found easily because the objective function is monotone. When the accumulator and the pump/motor take the low limit value, the optimal value of the system cost is 2540＄.

4.4 Weight determining

Based on the project experience and the expert opinion, the Decision Alternative Ratio Evaluation System (DARE) law is used to decide the weight distribution of the indexes, the method can be divided into the following five steps:

(1) Arranging the indexes from top to bottom on the principle that the index with bigger weight is on top.

(2) Deciding the chain radix S_0, which usually takes 10.

(3) Evaluating the chain evaluation S_i， S_i is got by quantifying the importance of the second index relative to the first index, the third index relative to the second index, the rest may be deduced by analogy. S_i varies from 0~10, 10 is of same importance, 0 means insignificant.

(4) Calculating the weight score A_i，

$$A_i = \begin{cases} A_{i+1}\dfrac{S_0}{S_{i+1}} & (i = 1,2,...,\text{k-1}) \\ S_i & (i = \text{k}) \end{cases} \tag{13}$$

where k, the number of the indexes.

(5) Calculating the weight W_i,

$$W_i = \frac{A_i}{\sum_{i=1}^{k} A_i} \qquad (i = 1,2,...,\text{k}) \tag{14}$$

the three indexes weight are shown in table 3.

Table 3 Index weight

No.	Indexes	S_0	S_i	A_i	W_i
1	Fuel economy	10		12.5	0.455
2	Dynamic performance	10	8	10	0.364
3	System cost	10	5	5	0.181

4.5 Optimization results and analysis

GA is used to solve evaluation function (1), Fig.8 shows the operation history. The results show that the evaluation function achieves the minimum when the accumulator and the pump/motor respectively take 72.8L and 125.8cc/rev. The parameter values involved in the optimization are listed in table 4.

Although the weight for the fuel economy is the biggest in the index weights, GA does not select the biggest accumulator as the optimal parameter to save more fuel. Fig.9 shows the fuel consumption and the accumulator pressure history with the change of the accumulator volume. When the accumulator is small, from 20L to 60L, the fuel consumption decreases rapidly with the increase of volume, but the effect weakens when the volume is bigger. The phenomenon is due to two points: one is the vehicle kinetic energy is limited, the other is the bigger accumulator adds the vehicle weight decreasing the fuel economy.

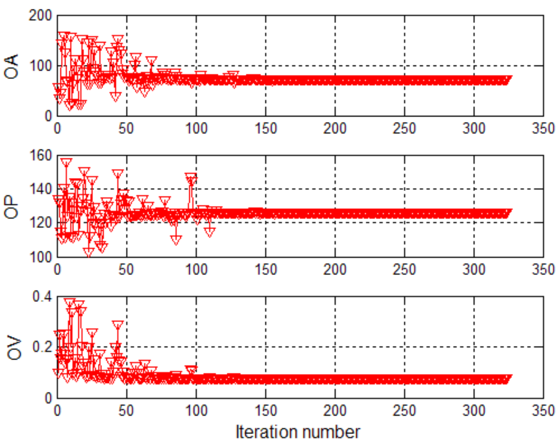

Fig.8 Optimal results of the optimization function (where, OA, optimal accumulator volume; OP, optimal pump/motor cubage; OV, optimal value)

Table 4 Value-taking in the optimization

Parameter	Value	Parameter	Value
a	0.914 m/s^2	P_0	100bar
i	31.6	P_1	130bar
i_i	3.93	P_2	315bar
l	3	r	0.505m
m	15500kg	φ	0.5
n	1.4	δ	1.01

As shown in Fig.10, under the same operation cycle, the parameter $\overline{P}$ varies with the change of accumulator. The results show that the value falls with the increase of the accumulator volume. It is to say that the $\overline{a}$ falls with the increase of the accumulator, and when the volume is bigger, the trend is obvious.

The above discussion shows that the increase of the accumulator volume will improve the fuel economy with sacrificing the dynamic performance, and the fuel economy improvement is weak when the accumulator is bigger.

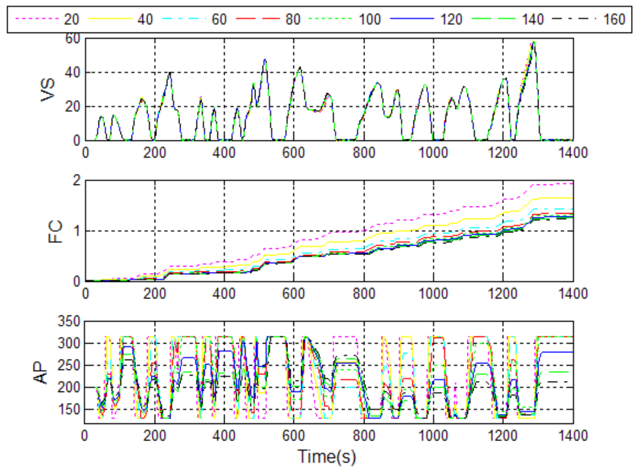

Fig.9 Operation history derived from different accumulator
(where VS, vehicle speed; FC, fuel consumption; AP, accumulator pressure)

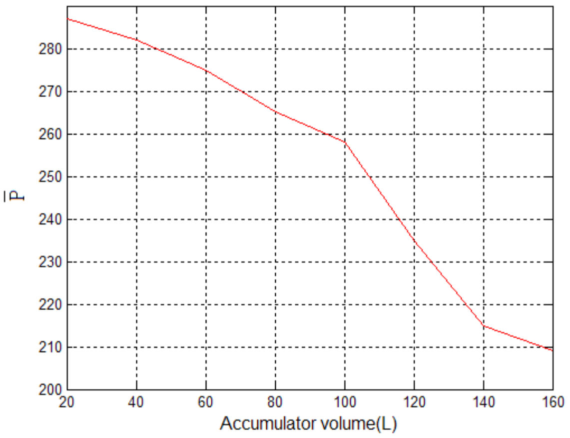

Fig.10 Parameter $\overline{P}$ obtained by different accumulator

Fig.11 shows the fuel consumption and the accumulator pressure history with the change of the pump/motor cubage. The fuel consumption falls with the increase of the pump/motor cubage, but the effect is weak. The phenomenon is due to that the smaller pump/motor can also relative effectively absorb the vehicle kinetic energy.

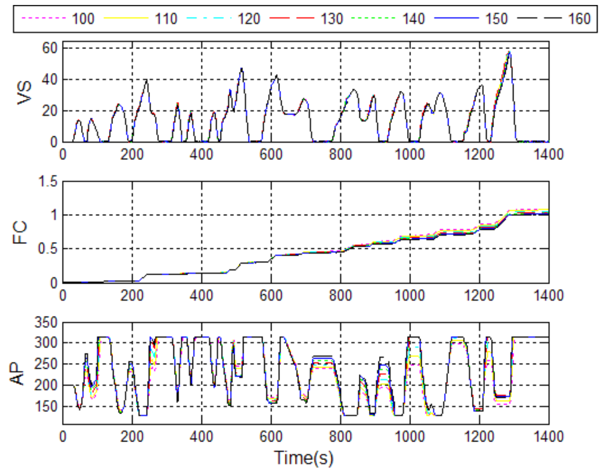

Fig.11 Operation history derived from different pump/motor
(Where VS, FC and AP are same to Fig.8)

Fig.12 shows the variation law of the parameter $\overline{P}$ and the corresponding torque of motor with the change of the pump/motor. The blue '-' curve is the parameter $\overline{P}$, the green '--' curve is the torque of the motor. Fig.12 shows that $\overline{P}$ rises with the increase of the pump/motor cubage but the amplitude is small. The torque of motor varies approximately linearly with the increase of pump/motor cubage.

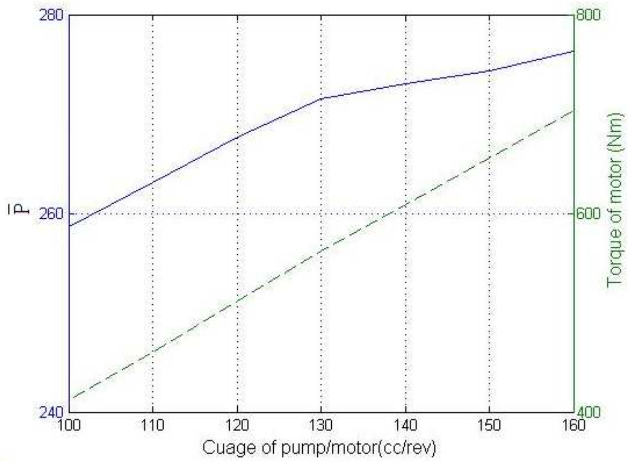

Fig.12 Parameter $\overline{P}$ and the corresponding torque of motor obtained by different pump/motor

When the accumulator is settled, the increase of the pump/motor will improve the dynamic performance but significantly adds the system cost and the effect to the fuel consumption is weak.

Table 5 shows the comparative results between the original and the optimal HRPS. The results state that the acceleration of the optimal HRPS improves greatly almost without adding system cost and sacrificing the fuel economy to the original one. The optimal system excels the original one greatly and the optimal designing approach presented by this paper is feasible.

Table 5 Compare of the original and optimal HRPS

Parameter and Evaluate index	Original	Optimal
Accumulator volume(L)	100	73
Pump/motor cubage(cc/rev)	100	125
Fuel saving ratio(%)	32.3	34.6
Acceleration(m/s2)	1.44	1.99
System cost($)	4071	4036
Evaluate function value	0.1183	0.0756

5 Conclusion

The coupling station of the pump / motor enables the HRPS to propel the vehicle independently with a smaller hydraulic motor. The control strategy avoids the power coupling between the engine and the motor, which simplifies the control process and is applied in engineering. The bigger accumulator can improve the fuel saving ratio but will lower the dynamic performance and add the system cost. The effect of the bigger pump/motor is limit to improve the fuel economy, but the increase of the cubage of the pump/motor can obviously improve the dynamic performance of the HRPS. The design considers the trade-off when optimize the HRPS parameters. The optmization design greatly improves the dynamic performance of the HRPS combined with reduced cost and equivalent fuel-saving. The study provides theoretical direction for the design of the HRPS for a HHB.

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References:

[1] B. H. Wang, J. W. Zhang and Y. G. Luo, The rapid development of a parallel hybrid propulsion control system by an online calibrating system, Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering, v221, n12, 2007, pp.1555-1565.

[2] G. L. Berta, E. Durelli and L. Nymann, Simulation models for hybrid buses, Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering, v212, n1, 1998, pp.59-72.

[3] K. Ahn and S. Cho, Optimal operation of the power-split hybrid electric vehicle powertrain, Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering, v222, n5, 2008,pp.789-800.

[4] C. P. Quigley, R. J. Ball and R. P. Jones, Fuzzy modelling approach to the prediction of journey parameters for hybrid electric vehicle control, Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering, v214, n8, 2000, pp.875-885.

[5] H. Wang, Y. Y. Liu, Z. D. Ma and H. Wang, Sub-frame design for a hydraulic hybrid vehicle system, American Society of Mechanical Engineers, Design Engineering Division, v118A, n1, 2005, pp.193-200.

[6] Q. Y. Zhang and S. Q. Chang, Modeling and simulation on hydraulic system of hydraulic hybrid vehicles, Journal of Nanjing University of Science and Technology, v32, n6, 2008, pp.701-706.

[7] Y. C. Yan, G. Q. Liu, J. Chen and T. M. Na, NLPQL of control rules for improving fuel economy of a parallel hydraulic hybrid bus, International Journal of Modelling, Identification and Control, v7, n4, 2009, pp.315-320.

[8] O. Hewko and R. Weber, Hydraulic energy storage based hybrid propulsion system for a terrestrial vehicle, Proceedings of the Intersociety Energy Conversion Engineering Conference, Vol. 4, 1990, pp.99-105.

[9] H. Sun, J. H. Jiang and X. Wang, Optimal torque management strategy for a parallel hydraulic hybrid vehicle, International Journal of Automotive Technology, Vol. 8, Nos. 6, 2007, pp. 791-798.

[10] B. Wu, C. C. Lin, Z. Filipi, H. Peng and D. Assanis, Optimal power management for a hydraulic hybrid delivery truck. Vehicle System Dynamics, Vol. 42, Nos. 1-2, 2004, pp.23–40.

[11] A. Pourmovahed, N. H. Beachley and F. J. Fronczak, Modeling of a Hydraulic Energy Regeneration System—Part I: Analytical Treatment, Journal of Dynamic systems, Measurement and Control, ASME 114 (1), 1992, pp.155-159.

[12] A. Pourmovahed, N. H. Beachley and F. J. Fronczak, Modeling of a Hydraulic Energy Regeneration System-Part II: Experimental Program, Transactions of the ASME, Vol. 114, 1992, pp.160-165.

[13] S. A Imagine, AMESim user manual, 2003.

[14] A. Pourmovahed and D. R. Otis, An Experimental Thermal Time-Constant Correlation for Hydraulic Accumulators, ASME Journal of Dynamic Systems, Measurement and Control, Vol. 112, No. 1, 1990, pp.116-121.

[17] J. M. Sun, Mechanical Optimization Design, China machine press, the second edition, 1998, pp.167-173.

[18] B. H. Wang, Y. G. Luo and J. W. Zhang, Simulation of city bus performance based on actual urban driving cycle in China, International Journal of Automotive Technology, 9 (4), 2008, pp.501-508.

Notation:

a	maximum acceleration in TUDS(m/s^2)
$\overline{a}$	average acceleration at the jumping-off point of different run-and-stop cycle under TUDS (m/s^2)
A_i	weight score value in DARE law(null)
AP	accumulator pressure(bar)
C_a	cost relative to accumulator volume(＄)
$C_{constant}$	constant part of system cost(＄)
$C_{p/m}$	cost relative to pump/motor cubage(＄)

C_s	system cost(＄)
DARE	the Decision Alternative Ratio Evaluation System law
E_a	energy stored in accumulator (J)
E_f	energy consumed by rolling resistance(J)
E_k	vehicle kinetic energy(J)
FC	fuel consumption(L)
F_f	rolling resistance(N)
F_i	gradient resistance（N）
F_j	acceleration resistance(N)
F_t	driving force(N)
F_w	wind resistance(N)
F_φ	vehicle maximum adhesive force(N)
f^{Δ}	objective function optimal value(null)
ge	the ratio of engine fuel consumption (g/kW.h)
i	total transmission ratio(null)
i_f	final gear ratio(null)
i_i	gearbox gear ratio(null)
i_t	transfer case gear ratio(null)
l	objective function number(null)
m	vehicle gross weight(kg)
Me	engine torque(N.m)
n	polytropic index
OA	optimal accumulator volume(L)
OP	optimal pump/motor cubage(cc/rev)
OV	optimal value(null)
P_0	accumulator pre-charge gas pressure
P_1	lowest work pressure(bar)
P_2	highest work pressure(bar)
$\overline{P}$	average accumulator pressure at the jumping-off point of different run-and-stop cycle under TUDS(bar)
Pe	engine power(kW)
q	pump/motor cubage(cc/rev)
r	wheel radius(m)
rev	engine speed(r/min)
s	braking distance(m)
S_0	chain radix in DARE law (null)
S_i	chain evaluation in DARE law(null)
SOC	state of charge of accumulator(null)
T_{engine}	engine torque(Nm)
T_{demand}	vehicle demand torque (Nm)
$T_{friction}$	friction brake torque(Nm)
T_{motor}	motor torque(Nm)
T_{pump}	pump torque(Nm)
$T_{pump\ max}$	maximum pump torque(Nm)
TUDS	typical urban driving schedule in China
v	vehicle speed(km/h)
V	accumulator volume(L)
VS	vehicle speed(km/h)
W_i	weight coefficient(null)
x	SOC value(null)
α	original vehicle fuel consumption(L)
β	rebuilt vehicle fuel consumption(L)
γ	fuel saving rate(null)
φ	adhesion coefficient(null)
δ	vehicle rotary mass conversion coefficient