Control Strategy Development and Optimization for a Series Hydraulic Hybrid Vehicle

Chih-Keng Chen, Tri-Vien Vu, and Chih-Wei Hung

***Abstract* – This paper is an extension of our previous work [1]. In this work, the series hydraulic hybrid model of a Truck Class II is established for both forward and backward simulation. At first, dynamic programming (DP) methodology was applied to estimate an optimal-benchmark solution for the proposed system over a pre-selected driving cycle. The optimal control trajectories include the engine power threshold, the operating range and the initial value of hydraulic accumulator pressure was inherited to establish the improved rule-based control strategies. The improvement of system fuel economy with different control strategies were assessed and compared.**

***Index Terms*- Dynamic-Programming, Hydraulic-Hybrid-Vehicle, Optimal-Control, Rule-Based.**

I. INTRODUCTION

GASOLINE depletion risks forecasting and environmental concerns are the main reasons that stimulate developing a higher efficient transportation system nowadays. In 2009, more than 70% of U.S oil consumption was consumed by transportation sector in which light-duty vehicles accounted for 45% [2] . Hence improving the efficiency of light-duty vehicle fleet is one of the most effective approaches to reduce the dependency on oil.

As a short and mid-term solution, recently, hybrid vehicles have aroused attention of researchers and manufacturers all over the world. In the recentpast, researches and studies have indicated that further advantage can be realized with hydraulic hybrid vehicle. It is observed that hydraulic pump/motor (P/M) units achieve a higher magnitude of power density than that of electric generator/motor [3]. With the development of modern technology, hydraulic bladder accumulators can accept high charging/discharging rates with above 95% of round-trip efficiency [4]. This feature allows hydraulic hybrid vehicles (HHVs) to achieve higher fuel economy through the regenerative braking ability. However, the low energy density property of hydraulic accumulator requires more space and significant mass reserving for energy storage component. Hence, a compromise should be taken into account.

In all types of hybrid vehicle, the main function of supervisory controller is to coordinate multiple power sources to satisfy the power demand of the driveline with minimum fuel consumption in the most convenient way. In general, control strategies of the hybrid propulsion systems can be classified into three categories: rule-based, semi-optimal, and global optimal. Among these, the rule-based control strategy is a real-time implementable power management. This control strategy uses several rules that would consider the vehicle load level and maintain the engine at its efficient operating range. Most papers regarding rule-based control strategy for HHV used the state-of-charge (SOC) of accumulator as the sole state variable [5]-[8].

Dynamic Programming (DP) is one of methodologies to find a global optimization solution for sequential or multi-stages decision problems. The algorithm searches for optimal decision at discrete points in a time sequence with chosen cost functions. In the hybrid system, the power split between the two energy sources is selected as the control variable and the total equivalent fuel consumption over a given driving cycle is the cost function of DP technique. DP has been shown to be a powerful tool for optimal control in automotive applications. It can be used to optimize powertrain parameters, gear shifting strategy in conventional vehicles [9]-[11] and for the control variables replied on torque split or power split factors in hybrid vehicles [5] [12] [13]. DP also has been successfully applied to replace the Rule-based control strategy for hybrid electric vehicle [14] or to optimize the Rule-Based control strategy for hydraulic-electric hybrid vehicle [15].

In this work, a series hydraulic hybrid propulsion system tailored for a light-duty delivery truck has been established. The remainder of this paper is organized as follows. The analysis and modeling of the system will be discussed in the second section of this paper. Then, the fundamental formulation of DP technique will be introduced in the first part of section three. The simulation results of DP application and the improved rule-based control strategies will be present and discussed in the second part of that section. The main contributions of this work will be concluded in the last section.

II. MODELING OF A SHHV

A. System Description

This paper proposes a configuration of SHHV for a Rear-Wheel-Drive 3.5 ton Light-Duty Class II Truck. The schematic and control signal path of the system is shown in

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Fig. 1. The specifications of the vehicle are listed in Table 1. In this configuration, Diesel Engine *E* is connected to hydraulic pump *P1;* P/M unit *P2* is connected to the rear wheels through differential DF; high-pressure accumulator *Acc* functions as a secondary power source of SHHV system. When the vehicle is in the traction mode, P/M *P2* operates as a motor to propel the wheels. Hydraulic power is provided by hydraulic accumulator *Acc* and/or pump *P1*. When the brake event occurs, *P2* will work as a pump to convert the vehicle kinetic energy into hydraulic power. Thus, the braking energy thus will be captured and stored by *Acc*. The mechanical braking system will be activated in emergency event or in case the pressure of *Acc* exceeding its maximum working pressure during vehicle braking mode. The required driving/braking torque is satisfied by adjusting the displacement of *P2*. Since the engine is decoupled from the wheel loads, it can be easily controlled to operate at desired location. The desired torque and speed of the engine can be achieved by control the displacement of P1 or/and the opening percentage of the engine throttle. In order to control the system, it is assumed that the measurements of engine speed, vehicle speed, and hydraulic system pressure are available.

B. SHHV discretized model

Many excellent textbooks and papers have been published on the subject of DP theory and its applications. They provide evidence that DP is a powerful numerical method for solving optimal control problems. Main advantage of DP above other methods is that global optimal solution is guaranteed with any type of problem. The serious drawback of DP is the computational effort grows exponentially with the number of state variables and inputs of the underlying dynamic system. Bellman called this difficulty the “curse of dimensionality”. Since implementation of DP algorithm on a computer arises many numerical issues and requires much effort, a general dynamic programming Matlab function, *dpm*, is used in this work. The *dpm* function was presented by Sundström and Guzzella [16]. The program was applied successfully for the optimal hybridization of hybrid electric vehicle [17].

This paper proposes a backward approach to estimate the system fuel consumption at each stage. From a given driving cycle, the desired vehicle speed and acceleration is defined. Using a simplified longitudinal vehicle dynamic model, the desired angular speed and desired torque of hydraulic pump/motor P2 are estimated. With a nonlinear hydraulic pump/motor model, the required driving/braking power of P2 is specified. This required power will be provided/absorbed by accumulator Acc and pump P1 due to the selected power split factor u. Using accumulator and pump models, the new stage of SOC and engine power will be estimated. From the desired power of the engine, the fuel consumption of the system can be estimated.

In this work, Japan 1015 driving cycle is selected as the driving test for proposed SHHV system. The speed profile and key statistical information of this driving cycle is shown in Fig.2 and Table I below.

With a given driving test cycle, at stage k, the reference speed v_k and acceleration a_k are available. The desired torque and speed of P/M unit P2 are estimated by (1) and (2) respectively.

$$T_{P2,k} = r\left[\delta m a_k + \left(C_d A_f v_k^2/2 + f_r m g\right)\right]/i_{df} \quad (1)$$

$$\omega_{P2,k} = i_{df} v_k / r \quad (2)$$

The definitions and values of above symbols are listed in Table 1. The actual displacement and flow-rate requirement of P/M P2 are estimated by equations (3) and (4).

$$D_{P2,k} = T_{P2,k} / p_{acc,k} \quad (3)$$

$$q_{P2,k} = D_{P2,k}\omega_{P2,k} \quad (4)$$

where $p_{acc,k}$ is the pressure of accumulator at stage *k*. At each stage, when the value of x is chosen, the pressure of the accumulator can be estimated by (5) below.

$$p_{acc,k} = x_k\left(p_{\max} - p_{\min}\right) + p_{\min} \quad (5)$$

Total efficiency of hydraulic pump/motor is a function of its displacement, angular speed, pressure different and its

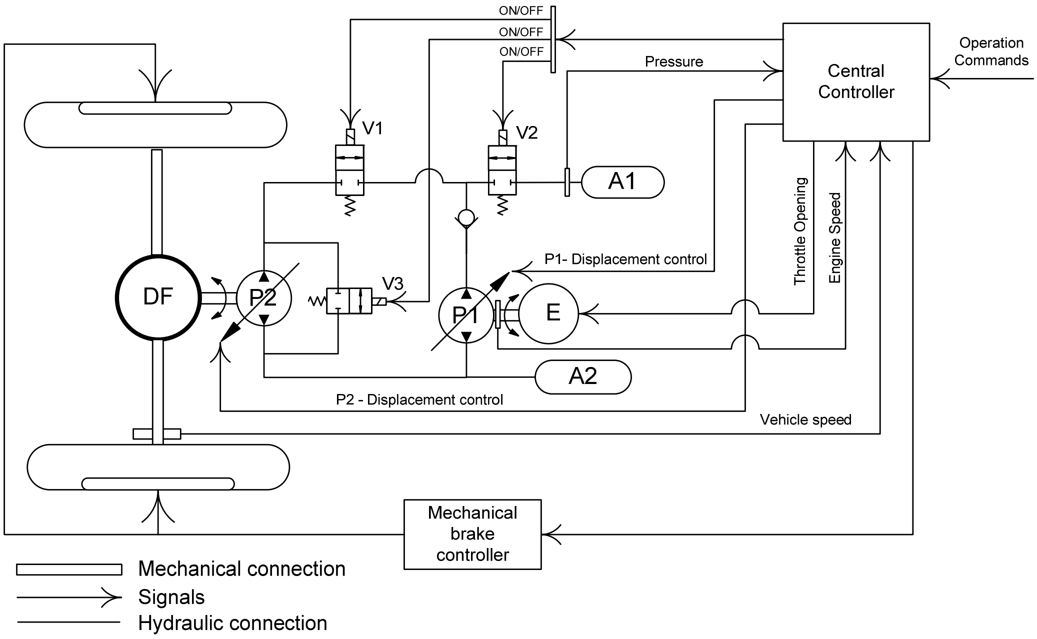

Fig. 1 Schematic and Control Signal Paths of SHHV.

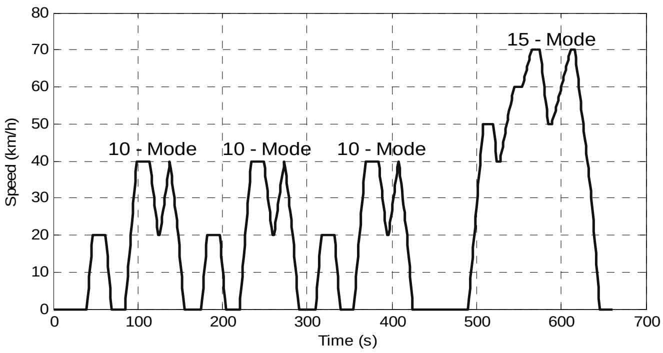

Fig.2 Japan 1015 driving cycle.

TABLE I
JAPAN 1015 DRIVING CYCLE INFORMATION

Symbol	Quantity	Value
	Vehicle	
Ts	Time	660 sec.
S	Distance	4.16 km
v_{max}	Max. Speed	70 km/h
v_a	Average Speed	22.68 km/h
	Max. Acceleration	0.79 m/s^2

working mode. In this work, the updated version of Wilson's P/M theory with the constants and calibrated coefficients roughly matched the model used by EPA [18].

$$\eta_{P2,k} = f\left(D_{P2,k}, \omega_{P2,k}, p_{acc,k}\right) \tag{6}$$

The desired power provided/absorbed by *P2* then be obtained by (7).

$$P_{P2,k} = T_{P2,k}\omega_{P2,k}\eta_{P2,k} \tag{7}$$

Assuming that the gas compression is determined based on the thermodynamics of ideal gas and the process is adiabatic, the volume of fluid at stage *k* can be estimated approximately by following equation.

$$V_{f,k} = V_0\left[1-\left(p_0 / p_{acc,k}\right)^{1/n}\right] \tag{8}$$

where V_0 is accumulator size, p_0 is air-charge pressure, *n* is specific heat ratio. For adiabatic process *n* is 1.4.
The fluid volume of accumulator is updated as

$$V_{f,k+1} = V_{f,k} - \text{sgn}(P_{acc,k})q_{acc,k}\Delta t \tag{9}$$

where Δt is the time step, $P_{acc,k}$ is the desired accumulator power which will be shown later. The result of the sign function is *+1* when $P_{acc,k} > 0$ corresponding to the dis-charging mode and *-1* for $P_{acc,k} < 0$ in the charging mode of the accumulator, respectively. The charged/discharged flow-rate of the accumulator depends on the desired accumulator power as

$$q_{acc,k} = P_{acc,k} / p_{acc,k} \tag{10}$$

The power of the accumulator P_{acc} and the engine-pump *P1* are determined from the desired power of *P2* by using the power split factor concept. This concept is selected as the control input variable of the model and can vary in a wide range, $u \in (-\infty, 1]$.

$$P_{acc,k} = u.P_{P2,k} \tag{11}$$

and

$$P_{P1,k} = \left(1-u\right)P_{P2,k} \tag{12}$$

The pressure and corresponding SOC of accumulator at next stage will be estimated as following

$$p_{acc,k+1} = p_0 / \left[1-\left(V_{f,k+1}/V_0\right)^n\right] \tag{13}$$

and

$$x_{k+1} = \left(p_{acc,k+1} - p_{\min}\right)/\left(p_{\max} - p_{\min}\right) \tag{14}$$

In this optimization, total efficiency of pump P1 is assumed a constant. Hence, the desired engine power is obtained as follow.

$$P_{E,k} = P_{P1,k} / \eta_{P1} \tag{15}$$

When the engine power demand is given, the desired engine toque and speed will be interpolated from the engine experiment data as shown in Fig. 3. Two PID controllers are used to control the engine torque and speed following reference values as presented in Fig.4. The outputs of the controller are engine throttle opening and pump displacement respectively.

C. Simulink Model of the SHHV

In order to assess the performance and benefit of different control strategy, a nonlinear model of the SHHV is established in the MATLAB/SIMULINK environment. The system is modeled as shown in Fig. 5. The parameter of the system using in this simulation is listed in Table II. Other vehicle specification, component selection, and parameter setting for the proposed model can be found in [19].

III. CONTROL STRATEGY OPTIMIZATION AND SIMULATION RESULTS

A. Fundamental formulation of DP

In the SHHV system, when the system configuration, component parameters, and driving cycle are defined, the fuel economy strongly depends on the coordinating of two

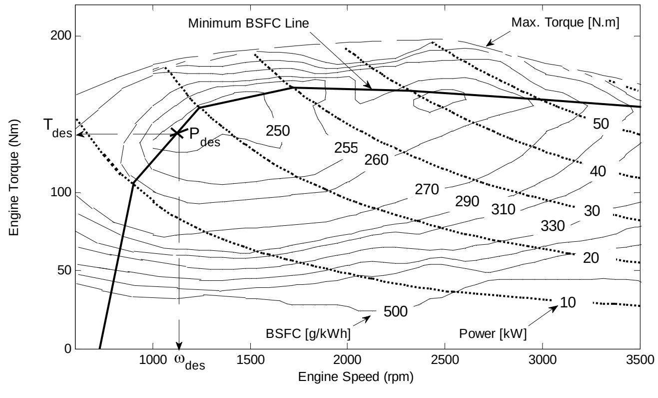

Fig. 3 Diesel Engine Experimental BSFC Map of the SHHV System.

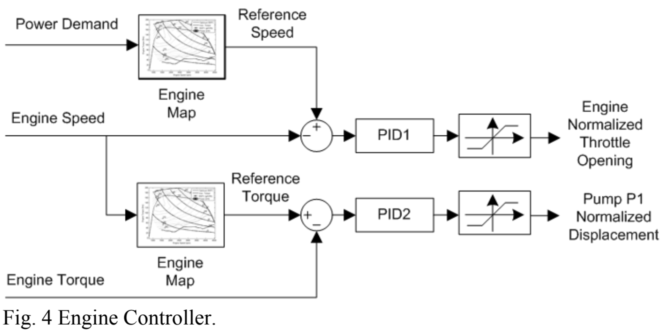

Fig. 4 Engine Controller.

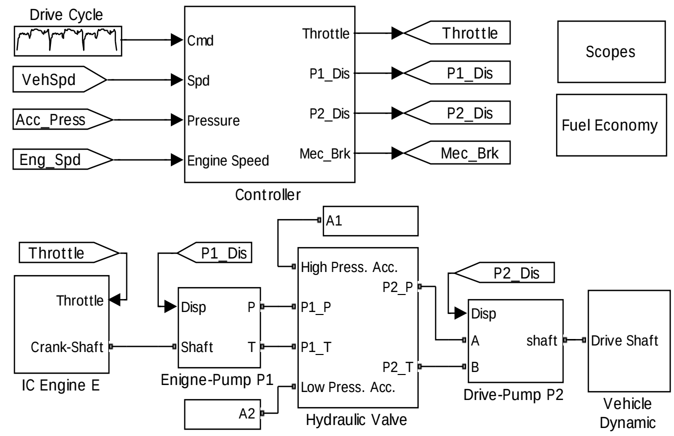

Fig. 5 Simulink Model of SHHV.

TABLE II
VEHICLE SYSTEM PARAMETERS

Symbol	Quantity	Value
	Vehicle	
m	Mass (Gross Weight)	3490 kg
δ	Equivalent Rotation Mass Ratio	1.15
A_f	Front Area	2.5 m^2
C_d	Drag Coefficient	0.3
ρ	Air Density	1.2 kg/m^3
i_{df}	Differential Ratio	4.875
f_r	Rolling Resistance	0.008
	Tire	195/75R16
	Engine	
	Model	4M42-4AT2
	Piston Displacement	2977cc
$P_{e,max}/\omega_{e,pmax}$	Max. Output	92 kW /3200rpm
$T_{e,max}/\omega_{e,tmax}$	Max. Torque	294 Nm/1700 rpm
$\omega_{e,max}$	Max. Speed	3700 rpm
	Cylinders	In-line 4 Cylinder/4 Stroke
	Compression ratio	17:1

power sources to propel the system. The objective of DP application is to find out optimal power split factor u that will minimize the fuel consumption Δm_f of the engine. The SOC of hydraulic accumulator is selected as the model state variable. Since there is no power split device available for SHHV. The purpose of applying *dpm* is actually to estimate the optimal power threshold, throttle opening of the engine, and the displacement of pump the hydraulic pump. The optimal control trajectories will be adopted to establish the implementable rule-based control strategies. The model equations are presented in previous part and can be summarized as

$$x_{k+1} = f_k\left(x_k, u_k\right) + x_k, \quad k = 0,1,...,N-1. \tag{16}$$

The optimization problem of minimizing the total fuel mass consumed over Japan 1015 driving cycle can be stated as the discrete-time optimal control problem.

$$\min_{u_k \in U_k} \sum_{k=0}^{N-1} \Delta m_f\left(u_k, k\right) \tag{17}$$

$$x_{k+1} = f_k\left(x_k, u_k\right) + x_k \tag{18}$$

$$N = 660 / T_s + 1 \tag{19}$$

where $\Delta m_f.T_s$ is the fuel mass consumption at each time step. When using the SOC of accumulator, a low power density energy storage device, as a state variable, if the time step is not small enough there may not exit any feasible solution. However, decreasing time step will increasing the time consumption of the DP program. Hence, it should be compromised. In this work, the time step is of 0.01s.

Physical constraints are given by equations from (20) to (24) below. If any constraint is violated, an infinite penalty is given to the cost function.

$$0 \le x_k \le 1 \tag{20}$$

$$\omega_{E\_\min} \le \omega_{E,k} \le \omega_{E\_\max} \tag{21}$$

$$P_{E,k} \le P_{E\_\max} \tag{22}$$

$$-D_{P2,\max} \le D_{P2,k} \le D_{P2,\max} \tag{23}$$

$$0 \le D_{P1,k} \le D_{P1,\max} \tag{24}$$

In order to apply DP algorithm, the control variable u and the state variable x are needed to discretize. Unlike in hybrid electric vehicle system, the state variable of the HHV system can be varied from zero to one. The lower bound of the control variable depends on the size of the engine. The smaller lower bound the larger capable of engine maximum power is required. The lower bound also affects the fuel consumption of the system due to the low power density characteristics of accumulator. Due to the limitation of the engine power of our current experiment platform, the lower bound of the control variable is select as zero. This prevents the engine from charging the accumulator during braking event to avoid unnecessary and inefficient energy conversion and storing the regenerative brake energy effectively. The state variable and the control variable are gridded into 50 points Detail of structure and syntax of the *dpm* function can be found in [16].

To estimate the fuel economy of the system, the final constraint of the SOC must be considered, the final value must be equal to the initial one. The example of applying this constraint to the system can be seen from Fig. 6. This constraint implied that there is no pre-charged energy in the accumulator during the test. The engine does not provide any unused energy stored in accumulator also.

The fuel economy of the system with different final constraint on the SOC is estimated and fitted as shown in Fig. 7. The results show that when the initial SOC is small the energy assistant from accumulator is small then the engine must operate at its high rate power which apart from its optimal region. However, if the SOC is too high, there is not enough space storing braking energy; some braking energy cannot be recovery hence the fuel economy of the system is reduced. The best fuel economy of the system is of 21.5 Km/L if the initial and the final value of the SOC is about 90%.

The behaviors of the system under 480 to 660 second interval of Japan 1015 driving cycle with DP applying are shown in Fig.8. Interestingly, the SOC can go down closely to zero, it means that the stored energy can be fully used at each vehicle start and the accumulator has enough space for

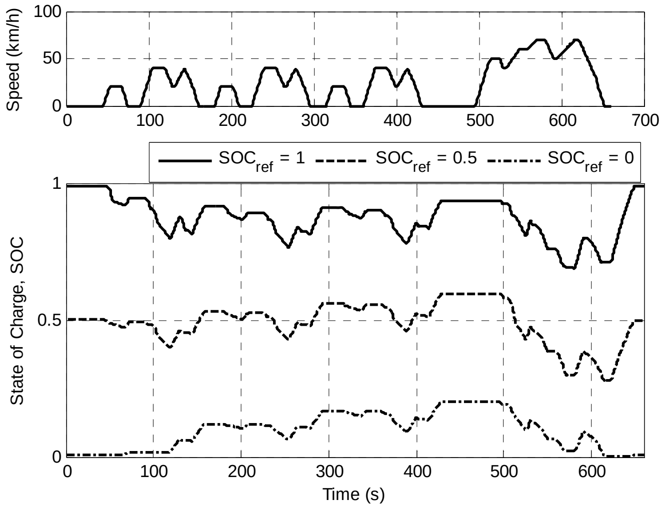

Fig. 6 Several SOC Trajectories with Different Final Constraint

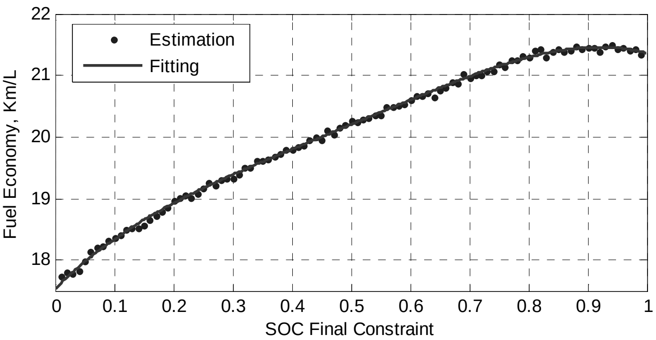

Fig. 7 The Relationship between Fuel Economy and Final Constraint on System SOC

effective regenerative braking. Besides, the whole braking energy over the driving test schedule is captured and reused.

It also can be seen that the operating power of the engine remain nearly constant even when the demand power is low or high. When the power demand is low, the exceeded power from engine is absorbed by the accumulator. When the power demand is high, extra power is assisted by the accumulator. As a result, the engine can operate at its high fuel efficiency region. In addition, avoiding transient operating of the engine and concerning its low energy density characteristic, the accumulator is only used to absorb the exceeded engine power during tracking phase but not charging directly. Hence, the power split factor is limited within zero and one.

B. Improved Rule-Based Control Strategy

One of the advantages of hydraulic accumulator in comparing with electric battery is that the accumulator can charge and discharge with a very high frequency. Besides, the accumulator can be charged to the full state or discharged to the zero state. The full state is at which the pressure of the accumulator reach its highest working value and the zero state is at which there is no available fluid stored in accumulator.

In thermostatic control strategy, the first controller either uses the SOC of accumulator A1 or the driver's command for determining engine ON/OFF state. If the controller only uses the SOC as the state variable, it is named Pure-thermostatic mode. In this case, whenever the accumulator is depleted it will be charged by the engine until reaching the full state. In the Acceleration-thermostatic mode, the accumulator will be charged whenever it is depleted and the acceleration of the vehicle is greater than zero. In the last case, the charging of accumulator will occur if the condition of SOC is met and the acceleration of the vehicle is greater or equal to zero. This case is named With-idle-thermostatic mode.

The benefit of the Pure-Thermostatic mode is the pressure in the accumulator being high most of time. The high acceleration driving condition can be met under any situation. However, higher pressure also means smaller available space reserved for regenerative braking. Hence, the efficiency of the system may be reduced. On the other hand, in the Acceleration-Thermostatic mode the energy buffer capable of accumulator may not be used, the engine operating points will be shifted apart from its optimal region due to the high power demand from driver. Inherently, With-Idle-Thermostatic mode is the best candidate since the engine can charge the accumulator when the vehicle is standstill to ensure that the accumulator can function as a secondary power source for a hard acceleration and have efficient space for braking energy recovery later. The engine state and SOC of the system under the first 160 Seconds of Japan 1015 driving cycle with different mode of thermostatic control strategy are illustrated as shown in Fig. 9.

The results of *dpm* application indicated that the optimal SOC initial condition is about 90%, the optimal power threshold is about 20 kW, and the optimal range of accumulator pressure is from 150bar to 250bar. From above analysis, the primary rule-based control strategy is derived by extracting the optimal control trajectories from *dpm* results. The behaviors of the system with the improved control strategy under Japan 1015 drive cycle is shown in Fig. 10. When the high power demand occurs, the pre-charge energy in accumulator will be used to provide the exceeded power, hence the engine will not necessary to be shifted to the higher power region. In addition, the pressure of accumulator is kept at low value before each hard deceleration event.

A summary of the effects of different control strategies over the performance and the fuel economy improvement of proposed SHHV system is given in Table III. The fuel economy improvement of the system was estimated with two different criteria. In the first criterion, the fuel economy of the system working on Hydrostatic was selected as the baseline to estimate the fuel economy improvement of the system with other control strategies; in the second criterion, the fuel economy of *11.42 Km/L* was selected.

TABLE III
SUMMARY OF THE SHHV PREDICTED PERFORMANCE AND FUEL ECONOMY IMPROVEMENT

No.	Name	Fuel Economy [Km/L]	Fuel Economy Improvement [%]	
			Criterion 1	*Criterion 2*
1	Hydrostatic	10.15	-	-
2	Thermostatic			
2.1	Acceleration	12.01	18.33	5.17
2.2	Pure	12.71	25.22	11.30
2.3	With-Idle	13.28	30.84	16.29
3	Optimal-Thermostatic			
3.1	Acceleration	16.36	61.18	43.26
3.2	Pure	17.79	75.27	55.78
3.3	With-Idle	18.13	78.64	58.76
4	DP	21.5	112	88

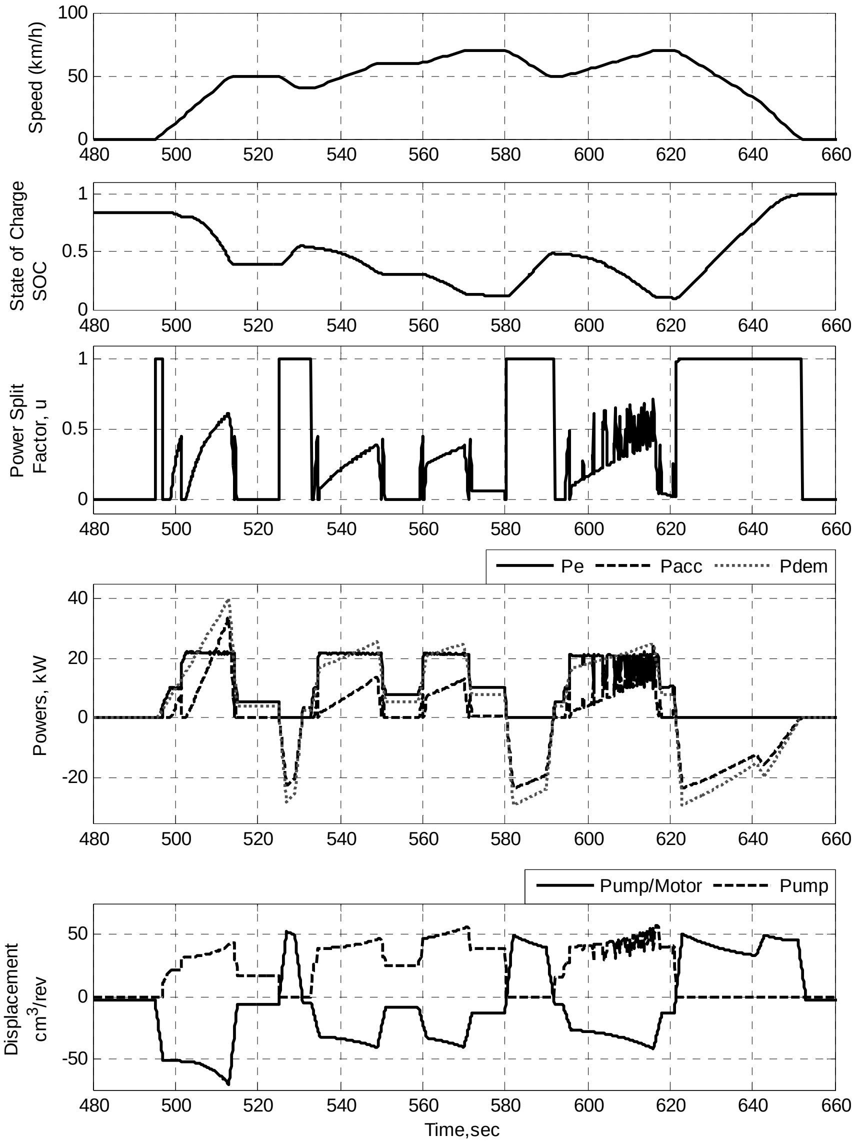

Fig. 8 SHHV DP results from 480 to 660sec. of Japan 1015 Driving Cycle

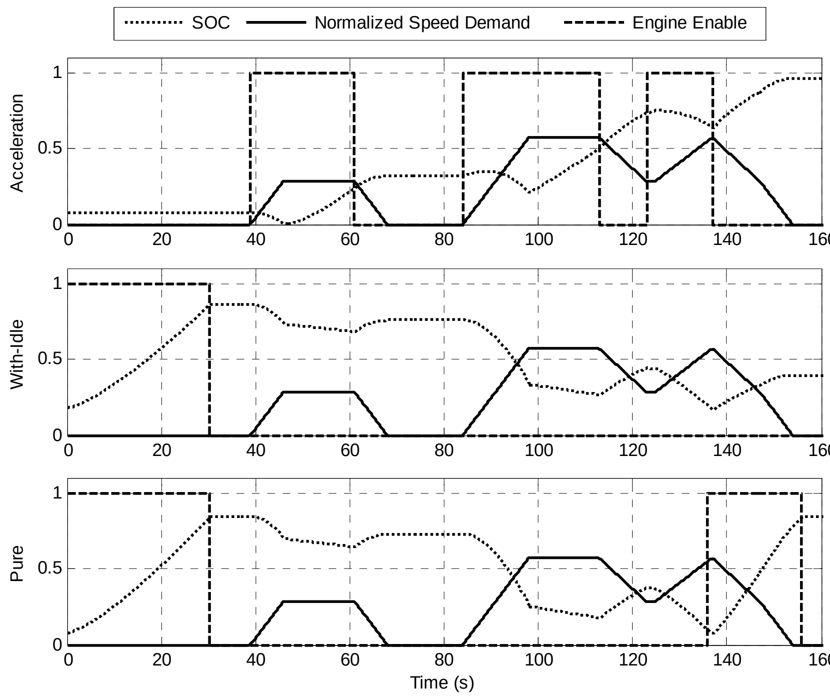

Fig. 9 The Engine State and Accumulator SOC of the System under the First 160 Seconds of Japan 1015 Driving Cycle with Different Mode of the Rule-based Control Strategy

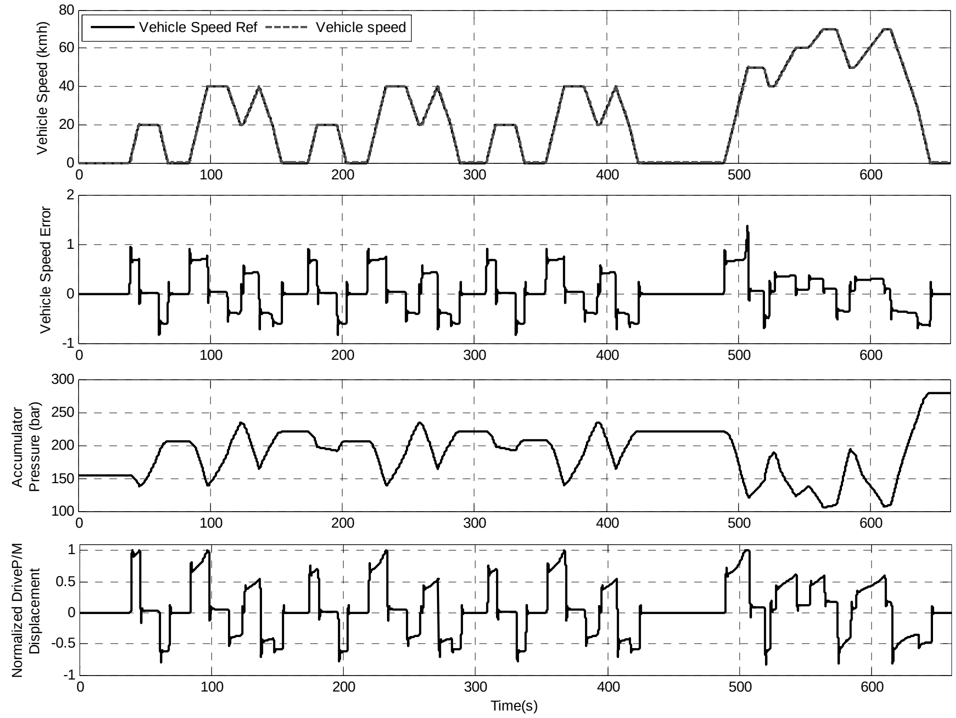

Fig. 10 System Behavior of the Optimal-Thermostatic Control Strategy.

IV. Conclusion

In this work, DP optimal control technique has been applied successfully for the SHHV system. The optimal trajectories has been studied and adopted to establish implementable rule-based control strategy. The control strategy has been simulated in the MATLAB/Simulink environment to predict the improvement of fuel economy of the proposed system in different modes.

Simulation results show that the fuel economy improvement of proposed system using rule-based control strategy can be up to 80% in comparing with a traditional hydrostatic control strategy and up to 60% in comparing with MYs 2012-2016 standards. With DP technique, the results can be up to 112% and 88% respectively.

In the future, since the power split device in SHHV has not been available, it is necessary to take the engine speed, engine-pump displacement and pump/motor displacement as control variables into account for more reasonable and accuracy problem.

References

[1] Chen, C.K., Vu, T.V., Hung, C.W., System Modeling and Control Strategy Development for a Series Hydraulic Hybrid Vehicle, Lecture Notes in Engineering and Computer Science: Proceedings of The International MultiConference of Engineers and Computer Scientists 2013, 13-15 March, 2013, Hong Kong, pp 177-183.

[2] Curtis, C., (2011, July 29). President Obama Announces New Fuel Economy Standards [Online]. Available: http://www.whitehouse.gov/blog/2011/07/29/president-obama-announces-new-fuel-economy-standards.

[3] Backe, W., "Present and future of fluid power", in *Proc. Inst. Mech. Eng. I J. Syst. Control Eng.* Vol. 207, No 4, 1993, pp. 193-212.

[4] Energy Protection Agency, "World's First Full-Size Hydraulic Hybrid SUV," *presented at 2004 SAE World Congress*, EPA420-F-04-019, 2004.

[5] Wu, B., Lin, C.C., Filipi, Z., Peng H., and Assanis,D., "Optimal Power Management for a Hydraulic Hybrid Delivery Truck," *Vehicle Syst. Dyn.,* Vol. 42, No. 1-2, pp. 23-40, 2004.

[6] Filipi, Z., Loucas, L., Daran, B., Lin, C-C., Yildir, U., Wu, B., Kokkolaras, M., Assanis, D., Peng, H., Papalambros, P., Stein, J., Szkubiel, D., and Chapp, R., "Combined Optimization of Design and Power Management of the Hydraulic Hybrid Propulsion System for the 6x6 Medium Truck," *Int. J. Heavy Veh. Syst.*, Vol. 11, pp. 371-401, 2004.

[7] Kim, Y. and Filipi, Z., "Series Hydraulic Hybrid Propulsion for a Light Truck - Optimizing the Thermostatic Power Management," SAE Technical Paper 2007-24-0080, (2007).

[8] Tavares, F., Johri, R., Salvi, A., and Baseley, S., " Hydraulic Hybrid Powertrain-In-the-Loop Integration for Analyzing Real-World Fuel Economy and Emissions Improvements," SAE Technical Paper 2011-01-2275, (2011).

[9] Kolmanovsky, I.V., Sivashankar, S.N., and Sun, J., "Optimal Control-Based Powertrain Feasibility Assessment: A Software Implementation Perspective," in *ACC 2005*, pp. 4452-4457, USA, 2005.

[10] Lin, C.C, Peng, H., and Grizzle, J.W., "Control System Development for an Advanced-Technology Medium-Duty Hybrid Electric Truck," SAE Technical Paper 2003-01-3369, 2003.

[11] O'Keefe, M.P., and Markel, T., "Dynamic Programming Applied to Investigate Energy Management Strategies for a Plug- in HEV," presented at the *Conf. EVS-22*, Yokohama, Japan, 2006.

[12] Kim, Y.A., "Integrated Modeling and Hardware-In-The-Loop Study for Systematic Evaluation of Hydraulic Hybrid Propulsion Options," PhD. dissertation, Dept. Mech. Eng., Michigan Univ., MI, 2008.

[13] Shan, M., "Modeling and Control Strategy for Series Hydraulic Hybrid Vehicles," PhD. dissertation, Col. Eng. Toledo Univ., OH, 2009.

[14] Lin, C.-C., Kang, J.-M., Grizzle, J.W., Peng, H., "Energy Management Strategy for a Parallel Hybrid Electric Truck", Proceedings of the American Control Conference, v 4, p 2878-2883, 2001.

[15] Lin, X.K., Ivanco, A., Filipi, Z., "Optimization of Rule-Based Control Strategy for a Hydraulic-Electric Hybrid Light Urban Vehicle Based on Dynamic Programming.

[16] Sundström, O. and Guzzella, L., " A Generic Dynamic Programming MATLAB Function," in *2009 Proc. IEEE Control Appl. Intell. Control,* pp. 1625–1630.

[17] Elbert, P., Ebbesen, S., and Guzzella, L., "Implementation of Dynamic Programming for n-Dimensional Optimal Control Problems with Final State Constraints," *IEEE Transactions on Control Systems Technology*, vol.PP, no.99, 2012.

[18] Pourmovahed, A., Beachley, N.H., and Fronczak, F.J., "Modeling of a Hydraulic Energy Regeneration System - Part I. Analytical Treatment," *AEME Journal of Dynamic Systems, Measurement, and Control*, Vol. 114, No. 1 pp.155-159, 1992.

[19] Vu, T.V., "Simulation and Design of Hydraulic Hybrid Vehicle," M.S thesis, Dept. Mech. and Eng. Da-yeh Univ., Changhua, Taiwan, 2011.