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Study of Hydraulic Regenerative Braking System in Hydraulic Hybrid Vehicles

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ABSTRACT

This study investigates a regenerative braking system for hydraulic hybrid vehicles because regenerative braking is one of the greatest advantages of hybrid vehicles. The regenerative braking system can capture and recycle the normally wasted braking energy during vehicle drive cycle. This paper presents a model of a novel configuration for hydraulic regenerative braking systems based on the SimScape toolbox in the MATLAB/Simulink environment. The influences of key component parameters on the performance of braking, the rate of recovered energy and system efficiencies are analysed. The simulation results indicate that, although the percentage of stored energy and the rate of energy recovery of the system vary and depend on numerous factors, the total efficiency is reasonable and has a high potential for implementation in real automobiles.

***Key Words*:** hydraulic hybrid, regenerative braking, SimHydraulic, energy recovery

液壓煞車能再生系統於液壓混合動力車之研究

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摘　要

本論文針對液壓混合動力車的煞車再生系統進行研究，煞車動能的再生利用是混合動力車最主要的優點之一，藉由煞車能再生系統的設計，能將一般車輛行駛煞車時浪費掉的能量儲存下來並再次利用。文中將介紹利用商用軟體 MATLAB/Simulink 中的 SimScape 工具列所建立之新型煞車再生系統架構與模型，並分析能量回收效益與關鍵元件參數對系統的影響，雖然能量儲存和回收再利用的比率會受到許多因素影響，經由模擬結果顯示出整體系統具有極高效益，且適合應用在實際車輛運輸產業上。

關鍵詞：液壓混合動力，再生煞車，SimHydraulic，能量回收

I. INTRODUCTION

In conventional vehicles, when a deceleration is took place, the sufficient braking torque will be applied to quickly reduce the vehicle speed to meet the requirement of brake performance. As a consequence, the kinetic and mass potential energy of vehicle is dissipated by mechanical friction braking. Among of wasted braking energy is significant, especially in a frequently braking pattern driving vehicles such as city buses and garbage collecting cars. For example, in typical urban driving cycles such as LA92, New York City, the percentage braking energy to traction energy are 58.01 and 81.9 percent, respectively [4]. The hybridization application for heavy trucks, which possess high mass potential energy, also helps recapturing and reusing significant amounts of braking energy.

In hydraulic hybrid vehicles, the hydraulic pressurized fluid power is utilized as alternative power source to the engine power which used diesel or gas as fuel source. The classification of hydraulic hybrid systems are normally based on the power flows between the engine, the hydraulic and the mechanic components of vehicles. There are two types of hydraulic hybrid system – parallel and series. In the first one, hydraulic power is co-operated with engine power to drive the wheels, while in the second one, the engine is completely decoupled from the wheels, most of mechanical transmission is eliminated, and the wheels are propelled by hydraulic power system which used the engine as a primary power source.

The common and important part of both kinds of hydraulic hybrid vehicles is the hydraulic regenerative braking system. In which, the Pump/Motor units can work as motor to drive the wheels and as a pump to recover wasted braking energy under the pressurized fluid stored in the high pressure accumulator. Investigating the influences of some parameters to the performance and the efficiency of the above system is the main objective of this paper.

In the hydraulic regenerative braking system, the brake energy is used as prime moving source for hydraulic pump; the low pressure fluid will be pumped from the reservoir to the high pressure accumulator which functions as a high pressure hydraulic energy storage device. Based on the state of charge of hydraulic accumulator, the required acceleration demanded, the stored energy will be used to propel the vehicle later on via hydraulic motor. The advantages of this system are high power density, high torque available, high rate and efficiency of energy recovery, and lower cost potential.

The efficiencies of the system have been measured experimentally and analytically. In Ref [10], the results indicate that the average efficiency of energy recovery of the system was 66%. The overall regeneration efficiency of the proposed hydraulic system (consists of three 2.5 litter accumulators, a Vickers fixed displacement 4.11 cm3/rev, an axial piston pump/ motor, and two 0.847 kg.m^2 moment of inertia flywheels) is 73% [3]. Results from studies of Pourmovahed et al. [9] showed that for a high and moderate pump/motor swivel angle, the round-trip efficiency of the hydraulic regeneration system varied from 61 to 89% [9]. Recently, Ho et al. [6] proposed a hydrostatic transmission composed with hydraulic accumulators. Regarding to the energy recovering potential of the system, simulation results indicated that 32 to 66% of braking energy can be recuperated [6]. The variation is due to the losses of load variation.

Although the result is different, yet it indicates that hydraulic regenerative braking system possesses a high potential of energy recovery. It is worth to apply this kind of system in real vehicle applications.

In this paper, a new model of configuration for hydraulic regenerative braking system is established. Based on simulation, the regenerative braking process will be investigated with regarded to the effects of main component parameters. The rate of recovery energy during regenerative braking and efficiencies of system also calculated.

After the introduction, the characteristics and working circumstances of the system will be analyzed in part two, the simulation will be described and the results are going to discussed in part three, and then, some conclusions are given in part fourth.

II. ANALYSIS OF HRB SYSTEM

The hydraulic regenerative braking system is made up of many components. The key components of the system are accumulators, pumps, motors, and the mechanical flywheel. The function of each component, the method to determine the specifications, and the working circumstances of the system will be described and analyzed in the following sections.

1. System Description

Focusing on the behaviors and the implementations of hydraulic regenerative braking system before applied for specific hydraulic hybrid vehicles, a hydraulic energy regeneration platform, as in Fig. 1, has been established. In this system, flywheel FW was used to stimulate reflected the mass moment inertia of a vehicle. Hydraulic motor M2 converts pressurized hydraulic fluid into rotating power to drive flywheel FW. High pressure accumulator A1 was used to store energy which was captured by hydraulic pump P2 during the regenerative braking process. Low pressure accumulator A2, functions as a reservoir, was used to provide low pressure fluid for the system in braking mode; it also holds the spent fluid that

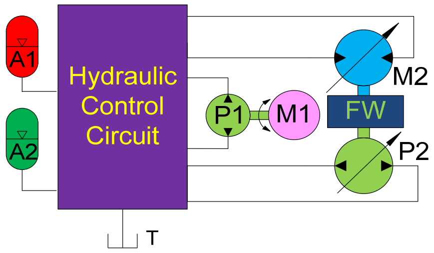

Fig. 1. Hydraulic regenerative braking system configuration

has been used by motor M2 in propulsion mode. Hydraulic pressure compensated pump P1 is used to provide additional constant high pressure fluid needed to drive flywheel. Electric motor M1 was used to provide prime moving source for pump P1. Hydraulic control circuit consists of several different hydraulic components and a controller to monitor and control the operating of system under various situations.

Accumulators A1 and A2 can be charged to reach specific pressure, considered as low working pressure whose effects will be discussed later, by using pump P1. The displacement of hydraulic motor M2 and pump P2 are variable. This characteristic allows controlling the tractive and braking torque which will determine the acceleration and deceleration of the flywheel.

To study the performance of the system, plausible operating modes of the system are considered, included pre-charging, propelling, regenerative braking, and regenerative propelling.

In the real applications, the preload of gas accumulator is obtained by pre-pressurized the gas in the gas chamber. After assembling into a specific system, this kind of parameter is not easy to change. The objective of the pre-charging mode is to find the required fluid volume, which will be used by the system, of A2, and find out the optimal initial pressure, that affect the braking torque as well as deceleration of the system, for A1. In addition, with the same initial pressure target, the higher preload of gas inside results a lighter accumulator since the gas has very low mass density characteristic in comparison with fluid one.

Different working conditions have been conducted to investigate the effect of component parameters. In the propelling mode, pump P1 will be used to drive motor M2 which will rotate the flywheel FW to reach specific steady state speeds at which the regenerative braking will be engaged. The displacement of P2 will be set at different values during braking mode to study the effect of that parameter to the braking performance as well as the rate of available stored energy. The important feature of the system will be validated through the regenerative propelling. After this mode, the rate of energy recovery will be calculated, the efficient of the system as well as of components also evaluated.

The flowing directions of fluid in each mode will be discussed below.

In the pre-charging mode, as shown in Fig. 2, the hydraulic system is connected to an external fluid tank. The electric motor M1 is activated to drive hydraulic pump P1. The hydraulic circuit will control the direction of fluid flow to charge A1 and A2. This process is monitored by pressure gauges mounted at the inlet port of accumulators.

After finding optimal initial pressure values of A1 and A2 for a certain vehicle application, the fluid tank can be removed from the system. Then a stand-alone hydraulic regenerative braking system is established. Hence the tank will not be mentioned in other modes thereafter.

The flow direction and components interplaying in the propelling mode is shown in Fig. 3. In this mode, hydraulic pump P1 converted mechanical energy into hydraulic fluid energy to drive hydraulic motor M2 which in turn converts hydraulic energy back to mechanical energy to propel the flywheel FW. P2 will be set at zero displacement in order to reduce its rotational resistance. In this case, A1 is isolated from the system while A2 function as a fluid tank. This process is monitored by an angular speed meter mounted at the shaft of motor M2. The steady state angular speed of flywheel is control by adjusting the displacement of M2.

In the braking mode, hydraulic pump P2 is activated, A1 is engaged into system. Kinetic energy from flywheel will be

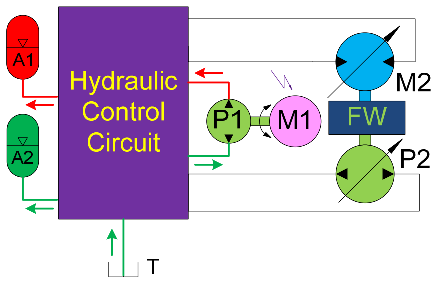

Fig. 2. Pre-charging

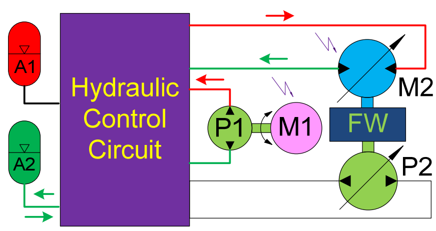

Fig. 3. Propelling

converted into hydraulic fluid energy by P2 and stored under high pressurized fluid in the fluid vessel of A1. In this case, A2 acts as low pressure reservoir. The fluid flow directions and activated components in this mode are shown in Fig. 4.

In the regenerative propelling mode, motor M2 uses hydraulic energy from A1 tor popel flywheel as shown in Fig. 5.

The states of components in different mode are summarized and listed in Table 1.

2. Component Specifications Determination

The specifications of components in the proposed system were considered to reflect the characteristic of a full series hydraulic hybrid implementation for a 2.5 ton medium truck class.

The required energy E_{rq} for a vehicle full stop from an initial speed V_0 is

$$E_{rq} = \frac{mV_0^2}{2} \quad (1)$$

Where m is vehicle mass, V_0 is longitudinal velocity at which the brake system is activated.

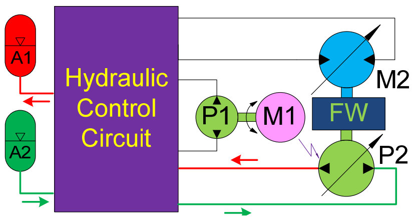

Fig. 4. Braking

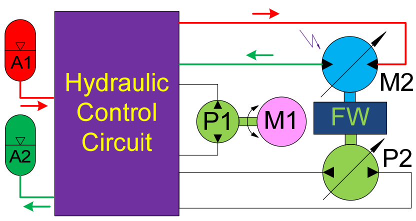

Fig. 5. Regenerative propelling

Table 1. Components Interplay in Each Mode

Mode	M1	M2	P2	A1	A2
Pre-charging	A(1)	D(2)	D	C(3)	C
Propelling	A	A	D	I(4)	T(5)
Braking	D	D	A	C	D.C(6)
Reg. Propelling	D	A	D	D.C	C

Note: (1) Activated; (2) Deactivated; (3) Charged; (4) Isolated; (5) Tank; (6) Discharged.

Since the braking time is normally short and the braking torque is very large in comparing with air drag and rolling resistance which can be neglected, it can be assumed that E_{rq} is the total energy that the ideal regenerative braking system might recover.

In this system, the mechanical flywheel is used to stimulate reflect the characteristic of mass moment of vehicle. To achieve that target, the flywheel must have enough mass moment inertia and must be accelerated to a certain angular speed. These parameters of the flywheel are determined from the principle of energy equivalent law.

$$E_{FW} = \frac{J\omega_0^2}{2} = \frac{mV_0^2}{2} = E_{rq} \quad (2)$$

Where J is mass moment of inertia and ω_0 is angular speed of flywheel FW. The angular speed of flywheel is limited by the maximum angular speed that hydraulic pump, motor can afford. It is obvious that the higher angular speed is used, the smaller inertia is required.

In hydraulic hybrid vehicle systems, hydraulic accumulator is normally bladder type which consists of pressure vessel with an internal elastomeric bladder. The bladder separates the accumulator into two sides, one with pre-charged nitrogen gas, and the other is hydraulic fluid as shown in Fig. 6 Neglecting the compression of fluid, and considering the nitrogen gas is an ideal gas. The relationship between pressure and volume of gas is following the thermodynamics principle as below

$$p_i V_i^k = pV^k = p(V_i - V_f)^k = C \quad (3)$$

where p_i, V_i is the initial pressure and volume of gas, V_f is the fluid volume, C is a constant depended on the initial condition of accumulator, k is the gas poly-index. For an adiabatic process, k = 1.4. States of accumulator are simplified described in Fig. 7.

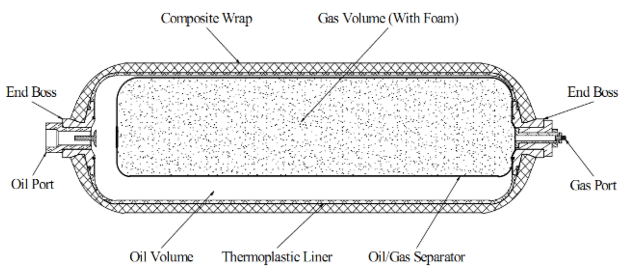

Fig. 6. Bladder accumulator components [2]

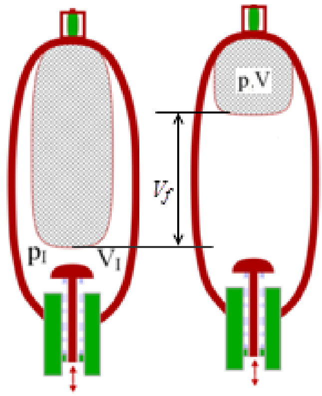

Fig. 7. Accumulator states

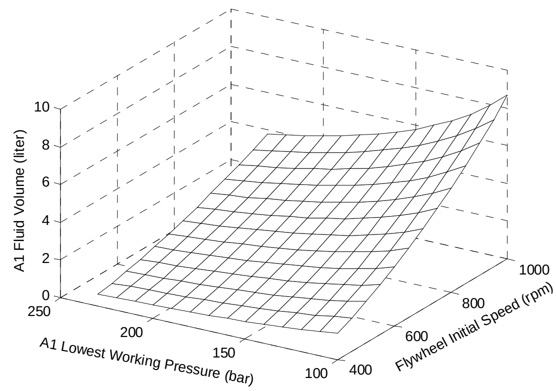

Fig. 8. Fluid volume flow into A1

Assumed when the braking is took place, the pressure of fluid inside A1 is of $p_{L,A1}$, the energy that accumulator stored during braking can be expressed as

$$E_{acc} = p_{L,A1} V_f \tag{4}$$

where V_f is the total volume of fluid that flows into A1 during braking.

To ensure that the system could recover kinetic energy of the flywheel at specific speed as much as possible and the pressure at inlet of A1 is not exceed its maximum working pressure, hydraulic accumulator must have enough space and sufficient maximum flow rate. From Eq.(2) and Eq.(4), we have

$$V_{f,A1} = \frac{J\omega_0^2}{2p_{L,A1}} \tag{5}$$

The capacitor of A1 is established from following equation

$$V_{A1} = \frac{V_{f,A1}}{\left(\frac{p_{pr,A1}}{p_{L,A1}}\right)^{\frac{1}{1.4}} - \left(\frac{p_{pr,A1}}{p_{H,A1}}\right)^{\frac{1}{1.4}}} \tag{6}$$

where $p_{L,A1}$, $p_{H,A1}$ are the lowest and highest working pressure of A1 during braking process.

The total fluid volume flow into A1 and the requirement capacitor of A1 due to the change of initial brake speed and lowest working pressure is shown in Fig. 8 and Fig. 9, respectively.

The inertia of the flywheel in this case is 15.83 kg.m^2 and the highest working pressure is set at 350 bar.

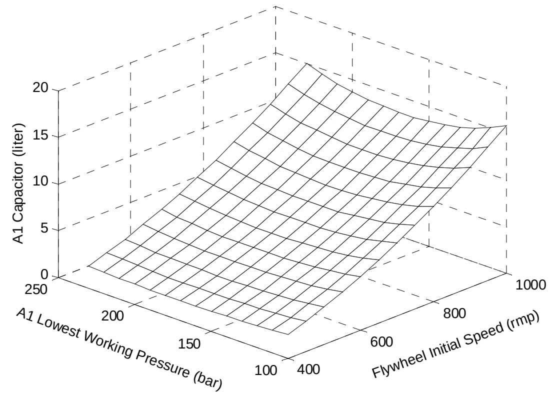

Fig. 9. Requirement capacitor of A1

Fig. 10 presents a simplified schematic of the system during braking period. Applied the Newton second law for this system, the equation of motion of the flywheel is

$$\frac{d\omega}{dt} = -\frac{T_{P2}}{J}. \tag{7}$$

where $d\omega/dt$ is the derivation of flywheel angular speed during braking period and T_{P2} is actual torque of pump P2.

The required torque to operate pump is

$$T_{P2} = \frac{x_{P2} D_{\max,P2} \Delta p}{\eta_{t,P2}} \tag{8}$$

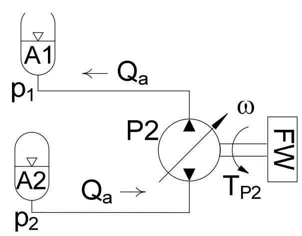

Fig. 10. Schematic diagram of HRB system in braking mode

where x_{P2} is the displacement fraction, $D_{max,P2}$ is the maximum displacement, $\eta_{t,P2}$ is the torque efficiency, and Δp is the differential pressure across of P2, respectively.

According to Fig. 10, the pressure differential across pump P2 is established as

$$\Delta p = p_1 - p_2 + \frac{fL\rho Q_a^2}{2d_0 A^2} \tag{9}$$

where p_2 is pressure at the inlet of A2, f is friction coefficient of fluid, L is effective hose length between A1 and A2, ρ is oil density, d_0 and A are internal diameter and cross section area of hose, respectively.

Pressure at the inlet of gas charged accumulators A1 and A2 during braking is determined from equation below

$$p_1 = p_{pr,1}\left(\frac{V_{a,1}}{V_{a,1-}\int Q_a dt}\right)^{1.4} \tag{10}$$

$$p_2 = p_{pr,2}\left(\frac{V_{a,2}}{V_{a,2-}\int \frac{Q_a}{\eta_{v,P2}} dt}\right)^{1.4} \tag{11}$$

where Q_a is the actual volumetric fluid flowrate generated by pump P2, $\eta_{v,P2}$ is the volumetric efficiency of P2.

The actual volumetric flow rate of oil through pump P2 is given by,

$$Q_a = x_{P2}\omega D_{\max,P2}\eta_{v,P2} \tag{12}$$

During regenerative propelling mode, the stored energy from accumulator A1 is used to propel the motor M2. And then, the hydraulic energy will be converted into mechanical energy to rotate the flywheel FW. Fig. 11 is simplified schematic diagram of system in regenerative propelling working mode.

The actual driving torque T_{M2} that the motor can generate is given,

$$T_{M2} = x_{M2}D_{\max,M2}\Delta p\,\eta_{t,M2} \tag{13}$$

where $\eta_{t,M2}$ is torque efficiency, x_{M2} is displacement factor, $D_{max,M2}$ is maximum displacement, and Δp is difference pressure across of motor M2.

The difference pressure across of the motor M2 is determined in the same way as of pump P2, but the pressure at the inlet port of A1 and A2 is determined as below

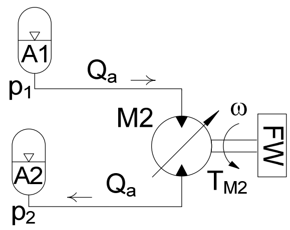

Fig. 11. Schematic diagram of HRB system in regenerative propelling mode

$$p_1 = p_{pr,1}\left(\frac{V_{a,1}}{V_{a,1-}\int Q_a \eta_{v,M2} dt}\right)^{1.4} \tag{14}$$

$$p_2 = p_{pr,2}\left(\frac{V_{a,2}}{V_{a,2-}\int Q_a dt}\right)^{1.4} \tag{15}$$

The fluid flowrate flow through M2 is given

$$Q_a = x_{M2}\omega D_{\max,M2}/\eta_{v,M2} \tag{16}$$

where x_{M2}, $D_{max,M2}$, and $\eta_{V,M2}$ are displacement factor, maximum displacement and volumetric efficiency of motor M2, respectively.

The rate of recovery energy of hydraulic regenerative braking system is defined as ratio of regenerative braking of hydraulic accumulator energy over kinetic energy of flywheel.

$$\varepsilon = -\frac{\sum_{i=0}^{\Delta t} p_i Q_{ai}\Delta t_i}{\frac{1}{2}J\left(\omega_0^2 - \omega_{\Delta t}^2\right)} \tag{17}$$

where Δt is braking period, $E_a = -\sum_{i=0}^{\Delta t} p_i Q_{ai}\Delta t_i$ is regenerative braking energy and $E_f = J\left(\omega_0^2 - \omega_{\Delta t}^2\right)/2$ is kinetic energy of flywheel corresponding to the speed reduction from ω_0 to ω_t. We can see that ε depends on the fraction of braking time over total driving time as well as the speed reduction during braking. As a consequence, the rate of energy recovery of hydraulic regenerative braking system will be varied based on the driving profiles that hydraulic hybrid vehicles work on.

From above equations, we can see that there are some main factors influence the braking performance as well as energy regenerative recovery rate. For instance, the displacement of pump and motor, the pressure, that is depended on the selection of its parameters such as capacity, pre-load, and fluid initial volume, at the inlet of accumulators. Others that have a significant influent are volumetric and mechanical efficiencies of pump and motor. Typically, the range of efficiency of hydrostatic pump and motor is from 80% to 90% [5]. By using principle of Floating Cup, the efficiency of hydraulic pump and motor can have significant improvement. Peak total efficiency could be up to 98% [5].

III. SIMULATION AND RESULTS

In order to investigate the performance of the system during braking as well as to verify the rate of energy recuperation of the system under a typical driving cycle, a hydraulic regenerative braking system model is established. The model is implemented in Simulink using corresponded hydraulic, mechanical, and other fundamental blocks from Simscape toolbox [8] as shown in Fig. 12. In this model, P1 is a hydraulic variable displacement pressure compensated pump block. M1 is an electric motor block. P2 and M2 are hydraulic variable displacement pump and motor blocks, respectively. A1 and A2 are hydraulic accumulator blocks. The Hydraulic Control Circuit composes from several hydraulic directional valve and pipe blocks. The Controller is used to generated control signal for the solenoids of hydraulic vavles V1, V2, V3, V4; the speed control signal for electric motor M1; the displacement control signal M2_D, P2_D of M2 and P2, respectively.

There are some assumptions of SimHydraulic blocks that used to build the model. For example, the volumetric efficiency of pump and motor, which is specified at nominal pressure, angular velocity and fluid viscosity, is considered as constants. The displacements of pump and motor are proportional to value of the control signal. The charging and discharging processes of accumulators are assumed to be polytrophic.

To investigate the effect of pump displacement, the value of this parameter is increased graduate for each simulation steps while other parameters of the system were kept as constants. The same approach was used to study the effect of

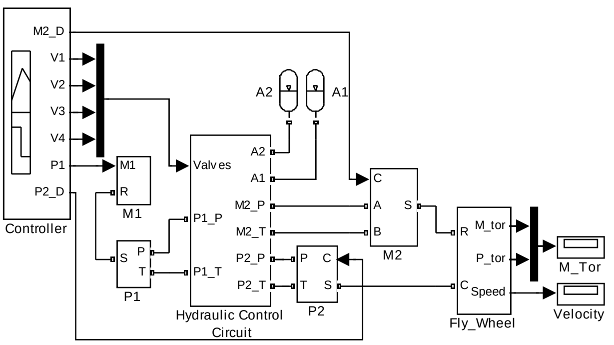

Fig. 12. Simulink model of HRB system

preload of high pressure accumulator A1. The value of parameters, the measurement data, and the numerical results are showed in Table 2. In which, the maximum pressure is the highest pressure of fluid stored in high pressure accumulator during braking for a given set of parameters value. A little amount of fluid was pre-stored in A1 under the form of initial volume to prevent the lack of fluid state during working processes. During energy recovery period, the braking torque is large enough to pump fluid flowing into accumulator. The percentage of stored energy is defined as a ratio of fluid energy stored in accumulator over the available braking energy.

Simulation results for the effect of pump displacement are showed from Fig. 13 to Fig. 15. We can see that the bigger pump displacement is used, the bigger braking torque we have, and the shorter braking time we get. If the displacement is not large enough, the generated braking torque is insufficient to stop the flywheel. From the calculation of measurement data, we also see that the bigger displacement also helps to get the higher percentage of stored energy. On the other hand, although the bigger displacement will cause a larger flow rate flow through the pump, the total fluid stored in accumulator increased a litter bit because of the decreasing of braking time. From these results, we can see that the braking system is easy to track the required deceleration by pump displacement controlling.

Table 2. Investigated parameters setting

Test No.	x_{P2} (%)	$p_{L,A1}$ (bar)	$p_{H,A1}$ (bar)	Braking Time (sec)
1	100	150	187.47	3.8
2	75	150	186.25	5.01
3	50	150	183.9	7.32
4	25	150	177.63	13.5
5	100	200	236.16	2.93
6	100	150	187.47	3.8
7	100	100	139.58	5.35
8	100	50	94.98	8.79

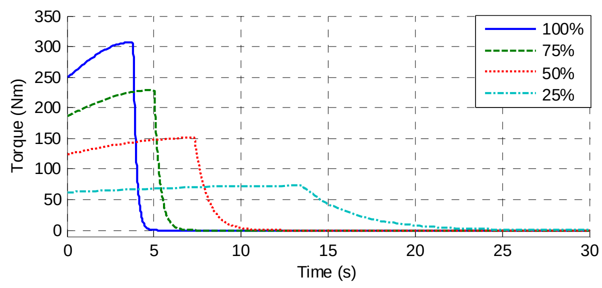

Fig. 13. The variation of braking torque due to pump displacement

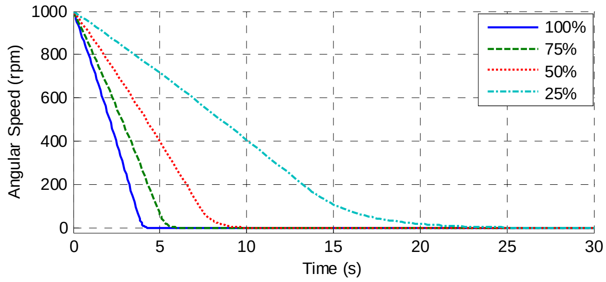

Fig. 14. The variation of flywheel speed due to pump displacement

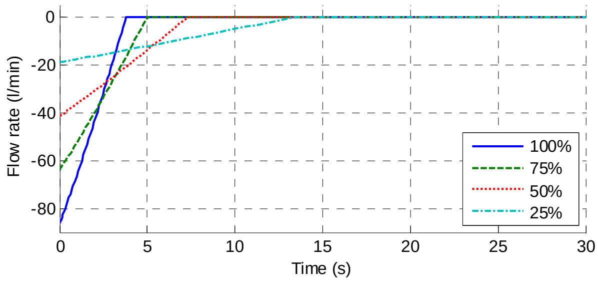

Fig. 15. The variation of flowrate eue to pump displacement

The influence of initial pressure parameter of A1 to the performance of the system is investigated through tests number five to number eight, the results are presented in Fig. 16 and Fig. 17. By increasing the initial pressure of A1, the system will be able to generate a bigger braking torque as showed in Fig. 16, and then the braking time will be shorter. The drawback of this approach is the increasing of fluid pressure in A1 as well as the maximum working pressure, which may increase the cost of hosing system, of the system. From Fig. 17 we can see that when the initial pressure of A1 is higher, the flow rate of fluid flows into A1 is reduced faster. Then, the volume of fluid stored in A1 is smaller. As a consequence, a smaller percentage of braking energy would be recaptured.

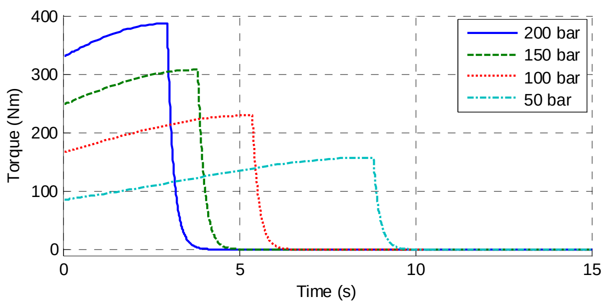

Fig. 16. The variation of braking torque due to the initial pressure of A1

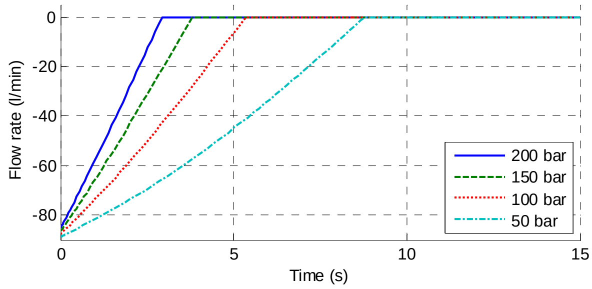

Fig. 17. The variation of flowrate due to the initial pressure of A1

To evaluate the efficiency of the components as well as the ability of energy recovering of proposed system, the system was conducted to work under following: propelling, braking, and regenerative propelling. The specifications of key components using in the simulation are listed in Table 3.

The maximum displacement of hydraulic pump is determined based on the maximum required deceleration of the flywheel while the displacement of hydraulic motor must be large enough to satisfy the required acceleration and the steady state speed of flywheel as well. The parameters of accumulators are selected based on the maximum available kinetic energy of the flywheel at the investigated braking speed.

During propelling mode, the displacement of pump P2 is kept at zero, the inlet and outlet of pump are connect together through a hydraulic valve to ensure there is no pressure different across the pump. That is, ideally, the pump loss in this case is very small. The same way is also applied for the motor M2 during braking mode to help the system recaptures the braking energy as much as possible.

Table 3. Component specifications

Hydraulic variable displacement pump P2	
- Max. displacement	75 cm^3/rev
- Volumetric efficiency	0.85
- Total efficiency	0.75
Hydraulic variable displacement motor M2	
- Max. displacement	75 cm^3/rev
- Max. speed	1180 rpm
- Max. torque	250 Nm
- Volumetric efficiency	0.85
- Total efficiency	0.75
High pressure accumulator A1	
- Capacitor	20 litter
- Initial pressure	120 bar
- Initial volume	2 litter
Low pressure accumulator A2	
- Volume	20 litter
- Initial pressure	4.3 bar
- Initial volume	10 litter

The efficiencies of hydraulic components are defined as ratios between its hydraulic power at outlet port and that ones at inlet port. The total efficiency of pump P2 defined as ratio between hydraulic power at the pump outlet and kinetic energy of flywheel. The total efficiency of motor M2 is a ratio between mechanical power at output shaft and hydraulic power at motor inlet. Simulation results for the system with different fraction of pump displacement are listed in Table 4. In which, *Eff_h1* and *Eff_h2* are total efficiencies of hydraulic components are taken into account during the braking and regenerative propelling processes, respectively. *Acc* is stand for efficiency, which defined as a ratio between stored and released energy, of high pressure accumulator A1. The total efficiency of the system is the percent of kinetic energy that the system could regenerate over the braking kinetic energy.

The numerical results from Table 4 indicate that the efficiencies of pump and motor, varied due to the fraction of displacement. With a larger displacement fraction used, the hydraulic pump has a higher efficiency. As a consequence, during braking process, we should try to use the pump displacement as large as possible to obtain high efficiency.

The angular speed of flywheel, the output shaft torque, pressure and flow rate at the inlet of A1 when the fraction of displacement is 100% are showed in Fig. 18. The negative flow rate means the fluid is flowing into A1 while the positive one means the A1 is discharging.

The efficiencies of component and the power flow in the system are presented in Fig. 19. In this case, the system can recover 86.1% of available braking energy and return 68.81 percent of that energy to the flywheel neglecting the road load friction.

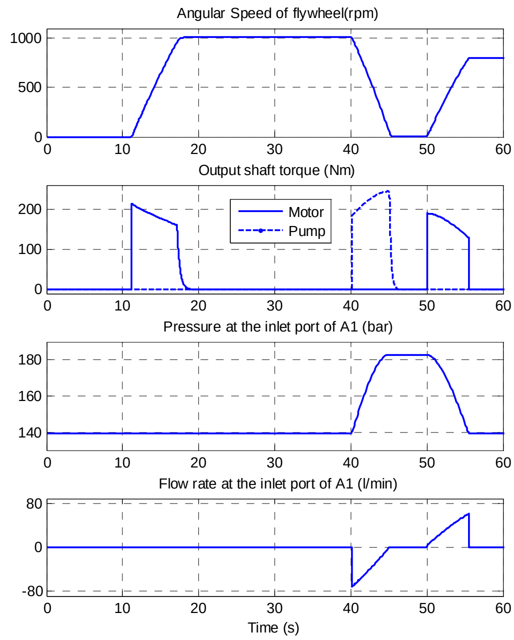

Fig. 18. Simulation results for a specific driving cycle

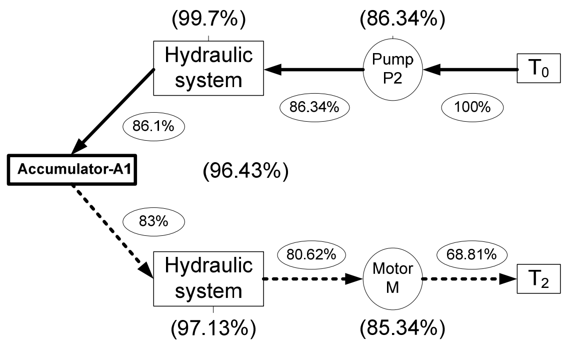

Fig. 19. Energy flow

IV. CONCLUSION

In this work, the relationship among components involved in regenerative braking and propelling operating modes has been analyzed. The Simulink model for a hydraulic regenerative braking system has been established. The influences of the main factors to the braking performance were investigated and analyzed. Simulation results indicated that more than 86% of braking energy can be captured and nearly 70% of kinetic energy can be reused. The architecture of the system allows building the experimental platform easily to implement and validate the component models as well as the merit of hydraulic hybrid system for vehicle.

Table 4. Efficiencies of components

Test No.	Displacement Fraction (%)	Efficiencies (%)					
		Pump P2	Motor M2	Eff_h1	Eff_h2	Acc	Total
1	100	86.35	85.34	99.74	97.11	96.43	68.81
2	75	83.97	83.85	99.74	97.19	97.88	60.86
3	50	79.08	77.41	99.74	97.17	97.76	55.96
4	30	69.98	69.63	99.74	97.16	97.81	50.44

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NOMENCLATURE

A	Internal cross section area of hose
A1	High pressure accumulator
A2	Low pressure accumulator
C	Accumulator initial condition constant

d_0	Internal diameter of hose
$D_{max,M2}$	Maximum displacement of M2
$D_{max,P2}$	Maximum displacement of P2
Δp	Pressure different across hydraulic components
Δt	Braking period
E_a	Regenerative braking energy
E_f	Kinetic energy of flywheel
E_{rq}	Kinetic energy of equivalent vehicle
f	Friction coefficient of fluid
FW	Mechanical fly-wheel
$\eta_{t,M2}$	Torque efficiency of M2
$\eta_{t,P2}$	P2 mechanical efficiency of P2
$\eta_{v,P2}$	Volumetric efficiency of P2
$\eta_{V.M2}$	Volumetric efficiency of M2
J	Mass moment of inertia of flywheel FW
k	Gas poly-index
L	Effective hose length between A1 and A2
m	Vehicle mass
$M1$	Electric motor
$M2$	Hydraulic variable displacement motor
$P1$	Hydraulic pressure compensated pump
$P2$	Hydraulic variable displacement pump
p_2	Pressure at the inlet of A2
$p_{H,A1}$	Highest accumulator working pressure
P_i	Initial pressure of gas
$p_{L,A1}$	Lowest accumulator working pressure
$p_{pr,A1}$	Accumulator charging pressure
Q_a	Actual volumetric fluid flowrate generated by hydraulic pump
ρ	Fluid density
T_{M2}	Mechanical torque at the output shaft of M2
T_{P2}	Mechanical torque at the output shaft of P2
V_0	Vehicle braking speed
V_{A1}	Capacitor of accumulator A1
V_f	A1 fluid volume
V_i	Initial volume of gas
ω	Flywheel angular speed
ω_0	Angular speed of flywheel FW
x_{M2}	Displacement factor of M2
x_{P2}	Displacement fraction of P2

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