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A Hydraulic Regenerative Braking System: Some Modeling and Experimental Results

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Abstract

This paper presents a principle diagram of the proposed kinetic recovery system using the hydraulic accumulator. The proposed system installed on a 2.5 tone garbage specialized dump truck made in Việt Nam. The system is connected to the power take-off unit of the transmission system on the vehicle, could brake to recover kinetic energy of the vehicle. The dynamic equations of a braking process using hydraulic accumulator have been obtained and simulated using Matlab - Simulink program. The system has been tested, the experimental results are well in agreement. The influence of the accumulator initial pressure values on kinetic energy recovering process and efficiency of the system have been looked into and estimated. The system could be used to study characteristics of the hydraulic regenerative braking system extensively.

Keywords: HRBS; HRB with PTO; Braking to recover kinetic energy.

1. Introduction

Conventionally, during a vehicle braking process, the kinetic energy turns to heat at the braking shoes [1]. In our study, we aim to turn the same to hydrostatic energy and store it in a hydraulic accumulator for other possible uses such as crane lifting, winching, vehicle initial accelerating.... A system for the purpose has been proposed, fabricated and installed in a 2.5 tone specialized dump truck in Việt Nam. A series of experiments have been successfully carried out. Some of the obtained results will be presented hereunder.

2. The system description

The principle diagram of the proposed hydraulic regenerative braking system (HRBS) is shown in Fig. 1. The main components are: 1- oil tank; 2 - pump; 3- control valve; 4- return line; 5- check valve; 6- pressure transducer; 7 – pressure switch; 8- hydraulic accumulator; 9- oil reusable line; 10 – pressure indicator; 11- power take-off unit; 12- proximity sensor (electromagnetic); Br – signal from braking pedal; P_{acc} – signal from pressure transducer; V_1 – valve control signal; CL-clutch-off control signal; CLPTO-PTO-on control signal.

The operating of the regenerative braking system includes idle mode and regenerative braking mode [2]. The entire braking process of the proposed system is controlled by an Arduino Uno card [3] operating as per the flowchart is shown in Fig. 2. The program runs and all the concerned output parameter values are recorded automatically.

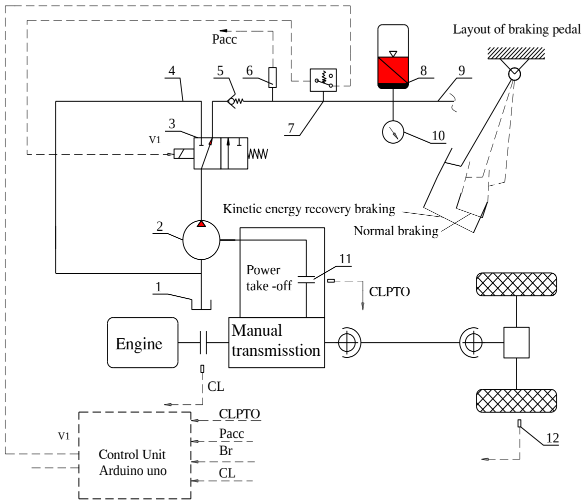

Fig. 1 The principle diagram of the proposed HRBS

As it can be seen from Fig.s 1 and 2, working of the system is very simple, the initial stroke of the braking pedal is accompanying with energy recovery process. In case of emergency, the remained part of the braking pedal stroke will activate the normal brake to stop the vehicle.

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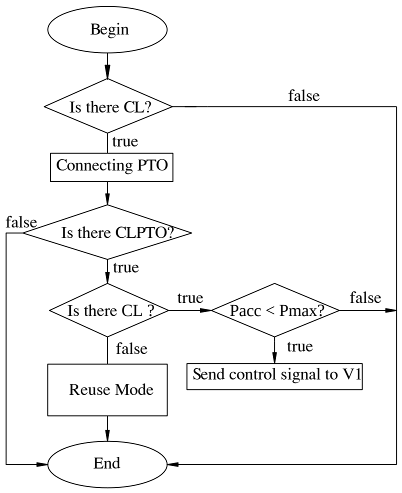

Fig. 2 The flow chart showing working of the proposed HRBS

3. The system modeling

Limiting the system of the vehicle operating on the delta. Considering a case shown in Fig. 3: the test vehicle runs on a straight and horizontal way. When the vehicle is decelerated, the vehicle is subjected by a system of forces as shown:

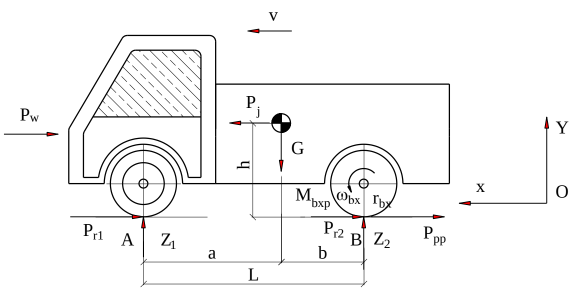

Fig. 3. Forces acting on the test vehicle

Balancing equation of forces acting on the test vehicle when brakes in a horizontal way have the following form:

$$P_j - P_w - P_{r1} - P_{r2} - P_\eta - P_{pp}=0 \tag{1}$$

where:

P_w is air resistance [4]:

$$P_w = KFv^2 \tag{2}$$

K- Coefficient of air resistance;

F - The vehicle cross-sectional area [m^2];

v - The vehicle braking speed [5] which is defined as:

$$v = v_0 - \int_0^t a_v dt \tag{3}$$

where: v_o – initial braking speed [m/s] and

a_v – vehicle acceleration [m/s^2]

P_{r1}, P_{r2} - rolling resistance of front and rear wheels [4]

$$P_r = P_{r1}+P_{r2} = f.G \tag{4}$$

P_r - total rolling resistance of wheels [N]

f- rolling resistance coefficient.

P_η – Friction Force of Transmission System [N].

P_j - vehicle's inertia force [4] which is defined as:

$$P_j = \delta \frac{G}{g}\frac{dv}{dt} \tag{5}$$

where:

G – weight of the vehicle [N];

t – time [s];

g - the acceleration of gravity,

g = 9,81[m/s^2];

δ - is a proportionality constant taking care of all rotating masses in the moving vehicle [4]. To simplify the calculation, an assumption has been made namely only vehicle wheels are to be taken into account, neglecting the contribution of the others rotating parts, hence:

$$\delta = 1 + \frac{\sum_i^e J_{bxi}}{m.r_{bx}^2} \tag{6}$$

For a vehicle wheel, it's angular and linear velocities depend on each other by:

$$\omega_{bx} = \frac{v}{r_{bx}} \tag{7}$$

P_{pp} - The braking force acting on the rear wheels by the system

$$P_{pp} = \frac{M_{bxp}}{r_{bx}} \tag{8}$$

where:

M_{bxp} - Braking torque at the wheels by the system [N.m]

r_{bx} – The working radius of the wheels[m]

a relation between the braking torque at wheels and the pump driving torque i.e., at the power take-off (PTO)'s output shaft, is defined as:

$$M_{bxp} = \frac{M_p i_{trp}}{\eta_{tr}} \tag{9}$$

where:

M_p - torque at the shaft of pump [Nm]

η_{tr} - the efficiency of the force transmission

i_{trp} - transmission ratio from pump shaft to the wheel shaft of the vehicle:

$$i_{trp} = \frac{\omega_p}{\omega_{bx}} \tag{10}$$

where:

ω_p - the angular velocity of the pump shaft [rad/s]

ω_{bx} - the angular velocity of the wheel shaft [rad/s]

- *Model of the hydraulic pump:*

Torque equation [6]:

$$M_p = \frac{d_p.\Delta p_p}{2\pi.\eta_{mp}} \tag{11}$$

Flow rate equation [6]:

$$Q_p = \frac{d_p.\omega_p.\eta_{vp}}{2\pi} \tag{12}$$

where:

d_p - pump displacement (m^3/rev).

Δp_p - Pump pressure (N/m^2)

η_{vp} – pump volumetric efficiency.

η_{mp} – pump mechanical efficiency

- *Model of the pipeline is written in the form of pressure loss* [7]*:*

Δp_{lpvao} - total pressure loss from the oil tank to the pump;

Δp_{lp}- total pressure loss from the pump to the accumulator.

In order to minimize the total pressure losses in the system, the cross dimension of the pipelines should be chosen in such a way that the oil will flow in the laminar regime.

If Δp_{lpi} is total line pressure loss, then the pump pressure may be defined as:

$$\Delta p_p = (\Delta p_{lp} + p_g - p_{vaop}) \tag{13}$$

p_{vaop} – pressure at the pump inlet

$$p_{vaop} = -\Delta p_{lpvao} \tag{14}$$

$$\Delta p_{lp} = \Delta p_{lp1} + \Delta p_{lp2} \tag{15}$$

Δp_{lp1} - pressure losses in pipe [7]

$$\Delta p_{lp1} = 10.(\lambda\frac{l_p}{d} + \xi_v)\frac{\rho.v_{di}^2}{2.g} \tag{16}$$

Δp_{lp2} – Total local pressure losses [7]

$$\Delta p_{lp2} = 10.\sum_{j=1}\xi_j\frac{\rho.v_d^2}{2.g} \tag{17}$$

where:

ρ - the density of the hydraulic oil [kg/m^3]

l_p - length of the charging line [m]

ξ_v - pressure loss coefficient at the entrance of the charging line

ξ_j - local line pressure loss coefficient at a j place

λ - a dimensionless friction factor depending on Reynold number (Re), from the case of laminar flow [8]:

$$\lambda = \frac{64}{Re} \tag{18}$$

$$Re = \frac{v_d.d}{\nu} \tag{19}$$

where:

d – the internal diameter of the pipe [m]

ν - oil kinetic viscosity [m^2/s]

v_d – average oil velocity [m/s]

$$v_d = \frac{4.Q_p}{\pi.d^2} \tag{20}$$

- *Model for the fluid accumulator*

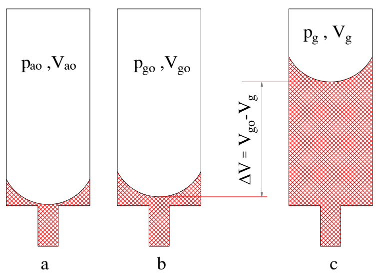

Fig. 4. The hydraulic accumulator and state of charge-SOC: a--The initial SOC, b- The low operating pressure and c – The high operating pressure

Neglecting any heat transfer may be presented in the energy recovering process, the following model for the accumulator could be arrived [9]:

$$p_{ao}.V_{ao}^{k} = p_{go}.V_{go}^{k} = p_g.V_g^{k} \quad (21)$$

where:

k – the adiabatic exponent which can be assumed at 1,4 [9] for two-atom gases such as nitrogen.

V_{ao} – effective gas volume (m^3)

p_{ao} – the pre-charge pressure (N/m^2)

V_{go} - gas volume at the minimum operating pressure (m^3)

p_{go} – the minimum operating pressure (N/m^2) (the initial pressure of accumulator)

p_g – the operating pressure (N/m^2)

V_g – gas volume at the working pressure (m^3)

From equation (21) and Fig. 4, p_g can be found as:

$$p_g = p_{go}.\left(\frac{V_{go}}{V_{go} - \int_0^t Q_p.dt}\right)^{1.4} \quad (22)$$

$$V_{go} = V_{ao}\left(\frac{p_o}{p_{go}}\right)^{\frac{1}{k}} \quad (23)$$

Hence the ratio of recoverable energy α is:

$$\alpha = \frac{E_{a\max}}{\Delta E_v} 100\% \quad (24)$$

where:

E_{amax} – The maximum energy that may be recovered and stored in the hydraulic accumulator during braking process: E_a (J)

$$E_a = \int p_g Q_p dt \quad (25)$$

ΔE_v – The testing vehicle kinetic energy available at the moment of starting to brake (i.e., the vehicle velocity is v_o), can be defined as [10]:

$$\Delta E_v = \left[\frac{m.v_o^2}{2} + \sum_{i=1}^{n} \frac{J_{bxi}.\omega_{bxoi}^2}{2}\right] \quad (26)$$

where: n – the number of the vehicle wheels.

The average braking acceleration of the vehicle: a_{vtb}, can be defined as:

$$a_{vtb} = \frac{v_o}{t_{ph}} \quad (27)$$

where:

v_o – The vehicle braking velocity (m/s)

t_{ph} - Braking time (s)

4. Method of calculation

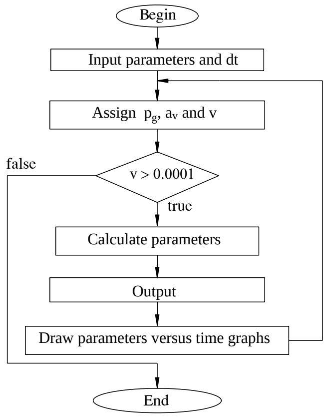

Fig. 5. The flow chart showing calculating steps in the proposed HRBS during the braking process

Based on the above obtained system of equations from 1 to 26 and using the Matlab - Simulink programme to model the braking kinetic energy recovering process. The calculating process is shown by the flow chart, Fig. 5.

5. Some modeling and experimental results

5.1 The system parameter values

The parameter values of the testing vehicle are shown in *Table 1*.

Table 1: Values of the testing vehicle parameters

Names	Designation	Values	Units
Vehicle mass	m	2400	kg
Vehicle wheel's radius	r_{bx}	0,355	m
Inertial torque of the wheel	J_{bx}	5,222	$Kg.m^2$
The cross-sectional area of the vehicle	F	2,4	m^2
The coefficient of air resistance [11]	K	0,58	
The transmission ratio from PTO's shaft to the vehicle wheels (number lever 3)	i_{trp3}	7,8	
The total length of the pipe from pump to H.A.	*l*	1,5	m
Inside diameter of the pipe	d	0,0127	m
Number of wheels	n	6	
Volume of oil that pump made in 1 rotation	d_p	14.10^(-6)	m^3/rev
Gas volume at the pre-charge p_{ao}	V_{ao}	0,025	m^3
The pre-charge pressure of gas	p_{a0}	70	bar

5.2 The results

After 5 times measuring in the test and calculating on the modeling, the results obtained in the braking process when the vehicle was running at a velocity v_o of 30km/h till stop, are shown in Fig.s 6, 7, 8 and 9.

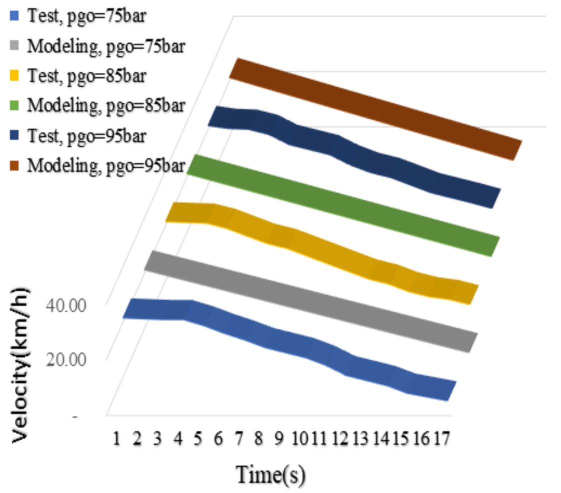

Fig. 6. Velocity versus time in the HRBS measured at various working pressures

As it has been clearly shown in Fig. 6 that, the higher the pressure the shorter the braking times:

Case 1: at a pressure of p_{go}=75 bar, braking time is 16s and the average acceleration a_{vtb} is of 0,52 (m/s^2);

Case 2: at a pressure of p_{go}=85 bar, braking time is 15s and the average acceleration a_{vtb} is of 0,56 (m/s^2);

Case 3: at a pressure of p_{go}= 95 bar, braking time is 14s and the average acceleration a_{vtb} is of 0,60 (m/s^2);

When brakes the test vehicle running at a velocity of 30km/h, the available kinetic energy ΔE_v as per equation 26 is 91937(J), Fig. 9 shows that:

Case 1: at a pressure of p_{go}=75 bar, the maximum accumulated pressure is 88(bar), the recovered volume of oil is 2,51(lít), the recovered energy is 21600(J) and the percentage of energy recovered is 23.49%;

Case 2: at a pressure of p_{go}=85 bar, the maximum accumulated pressure is 99.63(bar), the recovered volume of oil is 2,33(lít), the recovered energy is 22809(J) and the percentage of energy recovered is 24.81%;

Case 3: at a pressure of p_{go}=95 bar, the maximum accumulated pressure is 111.26(bar), the recovered volume of oil is 2,14(lít), the recovered energy is 23408(J) and the percentage of energy recovered is 25.46%;

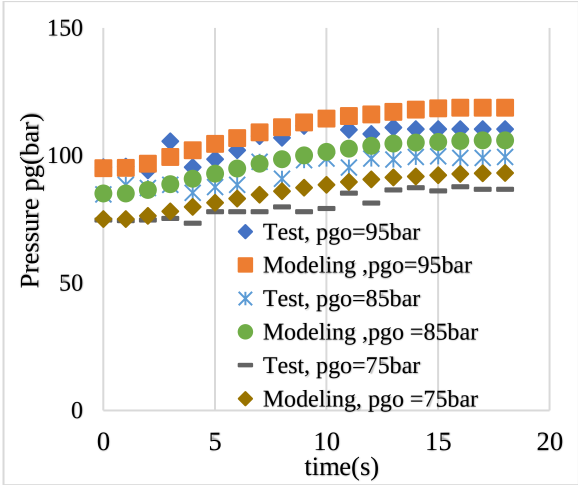

Fig. 7. Pressure versus braking time in the proposed HRBS: modelling and experimental results

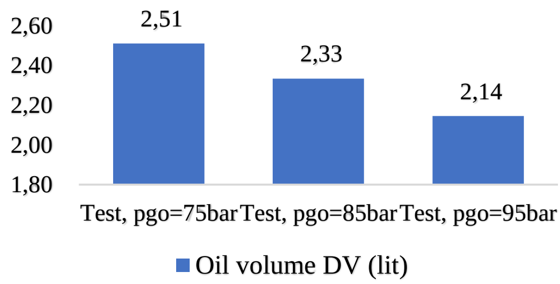

Fig. 8. The oil volume recovered in the proposed HRBS: experimental results under various pressure p_{go}

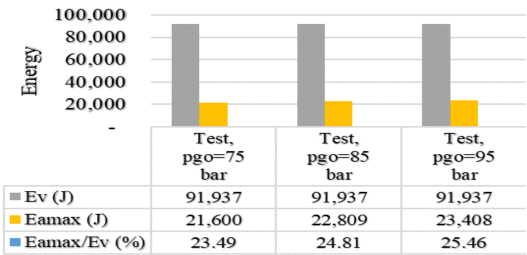

Fig. 9. Percentages of energy recovered in the proposed HRBS: experimental results under various pressure p_{go}

6 Conclusions

Based on the presented above results, the following conclusions can be made:

- The HRBS proposed in this study is a rational and profitably applicable in practice.

- In an operating pressure range around 100bar, the percentage of recovered kinetic energy is about 25%.

- Installation of the HRBS in a traditional vehicle will not adversely affect its safety

- It is advisable to use the Matlab-Simulink programme to simulate and evaluate the desired HRBS's characteristics before doing the actual design of the system.

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