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Original Research

Modeling and simulation of a braking energy regeneration system in hydraulic hybrid vehicles in the Colombian topography

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ABSTRACT

In this work presents the modeling and simulation of a braking energy regeneration system in a hydraulic hybrid vehicle. The vehicle used for testing is a 15000 kg heavy duty truck. The proposed regeneration system configured in parallel was subjected to a standard test speed profile such as the NEDC (New European Driving Cycle) and a real speed profile of a delivery vehicle in Colombian territory that was obtained by satellite tracking in order to compare its performance as an alternative in energy saving. A hydraulic circuit has been designed that considers travel characteristics such as track slope and vehicle speed to control hydraulic device displacement and mode of operation (braking and propulsion), as well as the dynamic model of the vehicle. The simulation has been developed in Matlab Simulink. The results of the model simulation are compared according to the torque required by the vehicle, the regenerated torque and the final torque. System performances are also compared with constant and variable displacement hydraulic device based on speed profiles and show a decrease in the torque spikes required in both the fixed and variable displacement of the hydraulic device, these energy savings have been estimated with the calculation of the power consumed based on time and are reflected as a percentage of the total required power. The development of the simulation yielded savings percentages of 15.5% and 22.5% for fixed and variable displacement respectively.

Keywords: Energy, regenerating, braking, consumes, efficiency.

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1. Introduction

The pollution and emission of greenhouse gases are produced mostly by the combustion of fuels in transport vehicles, also, the low energy efficiency of vehicles, the increase in demand for fuel increases the problem of pollution, for this reason, the scientific community has developed several alternatives to increase the energy efficiency of vehicles. One of these alternatives is applied to vehicle braking, when a vehicle uses the brakes, the kinetic energy produced by deceleration is converted into heat due to friction between the brake pads and the wheels, this heat is released into the environment and effectively lost, this loss depends on several aspects, such as the frequency of braking, the magnitude of deceleration and the duration of it. Most hybrid vehicles use this technology, since they use two sources of energy, one primary and the other capable of recovering energy, in this case kinetic.

The recovered kinetic energy remains stored until it is required again by the vehicle, where it is converted back into kinetic energy, which is used to propel the vehicle. The magnitude of the portion available for energy storage varies depending on the type of storage, the efficiency of the transmission, the driving cycle, and the inertia of the vehicle.

On national roads or highways, vehicles can travel long distances without braking, making the most of energy by decreasing heat loss, but in other driving conditions, such as in the city, where driving cycles involve frequent stops, vehicles lose more kinetic energy through braking, an ideal situation for a regenerative system.

The energy regeneration systems provide an increase in the energy used for input energy, this leads to an improvement in energy efficiency, this produces a decrease in the work produced by the engine and the amount of energy needed to accelerate the vehicle

2. Materials and methods

Brake energy regeneration systems are a solution to two main problems arising from the use of internal combustion engines, the first is the environmental impact produced by emissions of pollutant gases into the atmosphere aggravating the greenhouse effect [1]., it should be noted that vehicles do not make efficient use of these fuels, since much of this energy available in the fuel is discarded in other forms of energy that are not usable or easy to use.
Lately, the excessive use of technologies that start from the combustion of these hydrocarbons, has brought many disadvantages, in addition to those previously mentioned, since it does not have the proper structure to meet the growing demand for oil and its derivatives, for this reason, it is necessary to develop new technologies capable of achieving an operation with efficient use of energy from fuels , achieving to a large extent, reduces the emission of greenhouse gases [2].
The second is the direct impact on the economy, the growing demand and diminishing reserves of fossil fuels have produced a continuous rise in the price of fuel, which makes transport costs increases considerably, causing raw materials, food products, and finished products to assume that extra cost, therefore the end-user is the most affected. The land transport sector is one of the major emitters of polluting gases, produced by the combustion of hydrocarbons. These gases exacerbate environmental problems such as the greenhouse effect, acid rain, etc.
In summary, environmental factors and the price of fuel are decisive in the emission of contaminating gases and in economic development, respectively. For this reason, it is justified to study in-depth the technological alternatives applied to the energy efficiency of vehicles in order to reduce fuel consumption and the emission of greenhouse gases. Regenerative braking technology in hybrid vehicles is one option for dealing with the low efficiency of heavy goods vehicles by making competent use of fuel and reducing the production of polluting gases.

A great amount of kinetic energy is converted into heat at the moment of applying the brakes, this energy which is difficult to take advantage of decreases the energy efficiency of the vehicle since the energy consumed in braking is considerable, especially in cities with traffic heavy which ranges from 34.08% to 81.9% [3][4].
Regenerative braking can recover some of the kinetic energy before it is lost as heat, reducing fuel consumption, gas emission and brake wear. This decrease in consumption, emission and wear is strongly linked to the configuration, efficiency of the components and the control strategy applied. Electric hybrid vehicles are able to recover approximately 15% of available kinetic energy while hydraulic hybrids recover 70-80%.
These amounts of lost kinetic energy and the resilience of hybrids indicate that regenerative braking is effective in significantly improving vehicle energy efficiency. The authors even point out that better results are obtained when they are heavy-duty vehicles and have frequent speed changes, such as buses, courier trucks, and garbage collection [5].

The regeneration systems take advantage of the kinetic energy of the vehicle when it is in the braking stage, that energy that is usually lost in the form of heat is recovered and stored for a short time either in hydraulic accumulators or batteries depending on the hybrid. The transfer of kinetic energy from the vehicle to the tank is done by a hydraulic pump/motor connected to the power axe [6]. The description of the hydraulic hybrid system for energy regeneration is explained in detail by Chen [7].

In the braking stage, the hydraulic device works as a pump and drives the oil from the low-pressure reservoir or accumulator to the high-pressure accumulator. This energy is stored in the accumulator until it is required in the propulsion stage. When the vehicle needs propulsion, the hydraulic device functions as a motor and the oil passes from the high-pressure accumulator to the reservoir. The torque generated by the engine contributes to the torque generated by the internal combustion engine, from creasing the required torque peaks, which generally occurs at the start of the vehicle, in the case of the parallel configuration.

2.1. Development of the energy regeneration model

The model of a degree of freedom of the vehicle has been developed for the study of its dynamics, the free body diagram shown in Figure 1, this model will provide the torque and power profiles required in the engine to successfully complete the route. In the dynamic model of the vehicle [8],[4] geometric parameters of the vehicle, the mass and the driving cycle are considered. The vehicle must overcome forces due to gravity, wind load, rolling resistance and inertia.

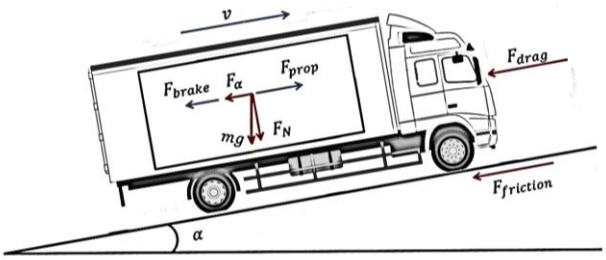

Figure 1. Forces applied during braking

The traction force is described with the following equations taken from Schaltz in $\mathbf{f_t}$[4]:

$$f_t = f_I + f_g * \sin(\alpha) + f_{rr} + f_{drag} \tag{1}$$

$$f_I = M_{truck} * a_{car} \tag{2}$$

$$f_g = M_{truck} * g \tag{3}$$

$$f_{rr} = M_{truck} * g * \cos(\alpha) * c_{rr} \tag{4}$$

$$f_{drag} = \frac{1}{2} * \rho_{air} * C_{drag} * A_{front} * (v_{truck} + v_{wind})^2 \tag{5}$$

where:
$\mathbf{f_I}$ = Inertial force of the vehicle.
$\mathbf{f_g}$ = Gravitational force.
$\mathbf{f_{rr}}$ = Rolling resistance.
$\mathbf{f_{drag}}$ = Wind resistance.
$\mathbf{C_{drag}}$ = Aerodynamic drag coefficient.
$\mathbf{c_{rr}}$ = Rolling resistance coefficient.
$\mathbf{A_{front}}$ = Front area of the vehicle.
$\mathbf{v_{bus}}$ = Vehicle speed.
$\mathbf{v_{wind}}$ = Wind speed.
$\boldsymbol{\rho}_{\mathbf{air}}$ = Air density.
$\mathbf{M_{truck}}$ = Mass of the vehicle.

For the dynamic model (Schaltz 2011) parameters are available such as the front area (6.5 m^2) typical of the vehicle geometry, the weightof the vehicle, the radius of the wheels, the rolling resistance (0.02), the drag coefficient (0.65)[10].
In addition to the speed profile, the model enters the total weight of the vehicle, the front area, the radius of the wheels.

Because the results of the publications did not consider the inclination of the terrain in addition to the need to validate the results under equal conditions, the model did not consider such inclination, although mathematically possible, its results were not validable with results in publications.
The model calculates the traction force, the torque required on the wheels and their angular velocity, the required torque and the angular velocity in the force take-down using equations 6 and 7:

$$\omega_{MCI} = i_{Dif} * \omega_{ruedas} \tag{6}$$

$$T_{MCI} = \frac{T_{ruedas}}{\eta_{Dif} * i_{Dif}} \tag{7}$$

2.1.1. Vehicle

The vehicle chosen is the International Trucks Dura Star Truck, as it meets the tonnage proposed in the overall objective. The traction force in the vehicle will be the sum of inertial force, rolling resistance and windload. With traction force and wheel radius, torque and power are calculated on the rear axle. To calculate torque on the power transmission axis, the differential ratio $i = 7.21$
The mechanical connection between the transmission of the vehicle is made by means of a forcesocket, which reduces the revolutions of the transmission with the ratio according to Song Yunpu [11] $i = 4.875$ and Kumar [12]. For the development of this research, it has been proposed to implement the braking energy regeneration system configured in parallel for a heavy-duty vehicle. This vehicle features a Cummins ISB6.7 engine and a gross vehicle weight of 35,000 lbs (15,875 kg).

Figure 2. International Dura Star 4300 series truck

2.1.2. Hydraulic accumulator

Depending on the speed requirements, the accumulator constantly changes its charging state, either in load mode when the vehicle slows down, or in unloading mode when the vehicle requires propulsion.

The hydraulic accumulator consists of a preloaded gas chamber and an oil chamber, this fluid chamber is connected to the hydraulic system, both chambers are separated by a membrane. If the fluid pressure at the inlet of the accumulator is greater than the preload pressure, the fluid will enter the accumulator and compress the gas, accumulating hydraulic energy [13]. A decrease in fluid pressure at the inlet of the accumulator will decompress the gas and the gas in turn will discharge the stored fluid into the system [14], [15].

During the operation of the regeneration system, this accumulator undergoes constant loads and discharges depending on the acceleration and braking maneuvers of the vehicle.
According to Pourmovahed [16] the energy balance in the accumulator is based on the gas present in it, as shown in equation 8.

$$m_g \frac{dU}{dt} = -p_g \frac{dV}{dt} - m_f c_f \frac{dT}{dt} - hA_w(T - T_w) \tag{8}$$

Where A_w is the effective area of the accumulator for convection heat. For a real gas, the derivative of internal energy can be expressed by equation 9.

$$\frac{dU}{dt} = m\left[C_v(p,T)\frac{dT}{dt} + \left(T\left|\frac{\partial p}{\partial T}\right| - p_g\right)\frac{dV_g}{dt}\right] \tag{9}$$

Where C_v (p,T) is the specific heat at constant volume as a function of pressure and temperature, Vg_{is} the specific volume of saturated steam, and is the partial derivative with constant volume.$\frac{\partial p}{\partial T}$

The temperature of the accumulator is expressed in equation (10) according to [17]:

$$\frac{dT}{dt} = \frac{1}{T_w}\left(T_{amb} - T_g(t)\right) - \frac{T(t)q_a(t)}{C_v m}\left|\frac{\partial p_g}{\partial T_g}\right|_v \tag{10}$$

Gas pressure is a function of gas temperature and specific volume, as expressed in the Beattie-Bridgman equation (11):

$$p_g(t) = \left(\frac{m}{V_g(t)}\right)^2\left[RT_g(t).\left(1 - \frac{mC}{V_g(t)T_g(t)^3}\right)\left(\frac{V_g(t)}{m} + B_o\left(1 - \frac{mb}{V_g(t)}\right)\right) - A_o\left(1 - \frac{ma}{V_g(t)}\right)\right] \tag{11}$$

Where A_0, B_0, and R are constant. The volume of gas can be estimated from the volumetric flow entering the accumulator by means of equation 12:

$$V_g(t) = V_{oa} + \int_0^t Q_a(t)dt \tag{12}$$

where:
U = Internal gas energy per unit mass[J]
P_g = Absolute gas pressure [Pa]
m_f = foam mass [Kg]
c_f = Friction coefficient.
h = Heat transfer coefficient [W/m^2. K]
A_w = Effective wall area of the accumulator[m^2]
T_g = Absolute gas temperature[K]
C_v = Specific heat at constant volume [J/Kg K]
V_g = Gas volume[m^3]
T_w = Accumulator Wall Temperature[k]
Q_a = Current flow[m^3/s]

In this model, Simulink's Simscape accumulator was used for hydraulic accumulators, as it is a simplified representation of the equations set out above and no thermodynamic approach is required for the accumulator in this research work. This representation has some considerations shown below:

- The process in the gas chamber is assumed to be polytropic.
- The load on the membrane, such as inertia or friction, are not considered.
- Hydraulic resistance to the inlet of the accumulator is not considered.
- The compressibility of the fluid is not considered.

In addition, oil in the working pressure ranges of the circuit was considered incompressible.

Hydraulic accumulator configuration parameters include preload pressure and accumulator volume. The preload pressure, according to the vehicle's publications and tonnage has been set to 18 MPa [18][19]and the maximum working pressure at 32 MPa [20]. This maximum working pressure is the humming pressure of the hydraulic system relief valve.
The volume of the hydraulic accumulator must ensure maximum utilization of available kinetic energy, the volume of the high pressure accumulator is 63 L, for a vehicle similar to that selected for the model, according to Kumar at [11] and [21] the NXQJ accumulator of 63L is selected.

2.1.3. Hydraulic pump/motor

Wilson's theory for pump/hydraulic motor in 1947 lays the groundwork for modeling axial, gear or vane piston pumps according to Pourmovahed.
The volumetric flow of the pump [m^3/s] is a nonlinear function of speed and displacement, the ideal flow through the pump/motor is given by equation 13.

$$Q_i = x\omega D \tag{13}$$

Where x it is a fraction of a maximum capacity, for a broken axis unit that refers to the angle of rotation by equation (14):

$$x = \frac{\sin\ \alpha_0}{\sin \alpha_{omax}} \tag{14}$$

The angle of rotation is set to be positive when operating as a pump. Inlet restrictions, leakage and fluid compressibility reduce flow rate. Volumetric efficiency (equation 15) is the reason between current and ideal flow rate.

$$\eta_v = \frac{Q_a}{Q_i} \tag{15}$$

The torque required in ideal operation, i.e. frictionless is given by equation 16.

$$T_i = x\Delta pD \tag{16}$$

Where "D" is the volumetric capacity of the hydraulic device at [m^3/rad]. Mechanical efficiency in the pump is the reason between the ideal and the current torque (see equation 17).

$$\eta_t = \frac{T_i}{T_a} \tag{17}$$

The equivalent equations for motor operation mode are equations 18 and 19, which define volumetric and mechanical efficiency respectively.

$$\eta_v = \frac{Q_i}{Q_a} \tag{18}$$

$$\eta_t = \frac{T_a}{T_i} \tag{19}$$

The current volumetric flow of hydraulic oil coming out of the pump is equal to the compression rate of the gas in the accumulator (see equation 20).

$$Q_a = -m_g \frac{dv}{dt} \tag{20}$$

Regenerated torque is the difference between current engine torque and friction losses and bearing losses.
Differential pressure depends on the mode of operation of the hydraulic machine (see equation 21).

$$\Delta p = p - p_{tank} + \mathrm{sgn}(\alpha_0)\Delta p_L \tag{21}$$

where:

$$\mathrm{sgn}(\alpha_0) = \begin{cases} \dfrac{+1\ \text{pump mode}}{-1\ \text{motor mode}} \end{cases}$$

Hydraulic axial piston machines have several pistons that are placed parallel to the drive shaft. Available for both variable flow and fixed flow rates, these machines operate according to the tilting plate principle, to vary the flow from zero to a maximum.
The hydraulic pump/motor for a regeneration system configured in parallel, for a vehicle of about 15 tons is the Bosch Rexroth A4VG90, used by Liu [21]

The volumetric capacity of this hydraulic pump/motor is 90 cm^3/rev, which for an angular speed of 3050 rpm can transfer a flow rate of 275 l/min.
For the simulink model, the "Hydraulic Pump" and "Hydraulic Motor" blocks (see Figure 3) were used to simulate the pump/motor, as this ensured the timely switching of the regeneration system between propulsion mode and braking mode, by means of the switching valve.
The ports for the variable displacement pump, to the left of Figure 3, are "S" for the connection of the shaft, in this case the axis of the socket connected to the vehicle transmission, "P" and "T" are hydraulic ports for pressure (high pressure accumulator) and tank (reservoir) and port C", which is the control signal for the displacement of the pump by means of the tilting plate [22].

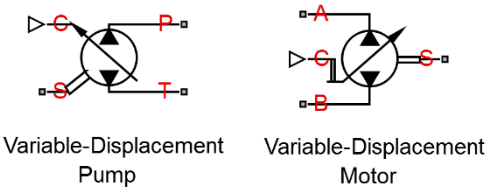

Figure. 3 Simulink variable displacement pump and hydraulic motor.

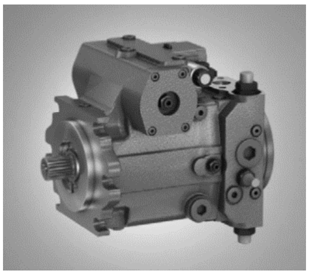

Figure 4. Pump and hydraulic displacement motor A4VG (Bosch Group)

2.1.4. Speed profile

The heavy-duty truck moves under a speed profile, two speed profiles were used for this simulation, one standard profile (New European Driving Cycle) and another actual profile obtained thanks to a company that tracks transport fleets satellitely.

Choosing the NEDC standard speed profile for this simulation was based on having a reference for the model used, that is, simulating that model with a profile and comparing it with known results to determine whether the percentage of decrease is close to the desired one.
The decrease in consumption for this profile is close to 27% according to Taylor [23]. Figure 5 shows the NEDC speed profile.

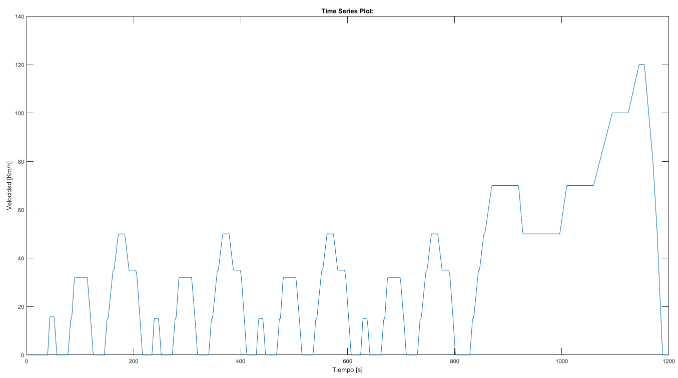

Figure 5. NEDC speed profile

The second speed profile used was a speed profile typical of a transport truck route, this profile was obtained through a data table provided by Navisaf that provides fleet management services.

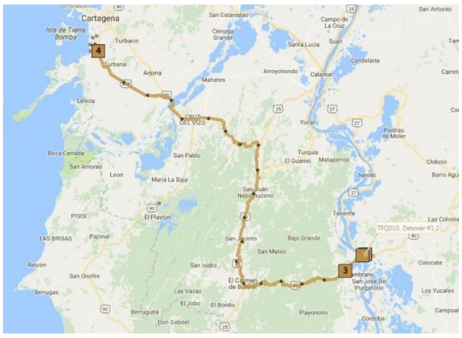

Figure 6. Vehicle route for testing

From this route, a part of it was chosen, where there were significant changes in speed, since these are favorable for the regeneration of braking energy, this fraction is between the municipality of Plato and Villa Aurora (see Figure 6). This speed profile lasts approximately 1800 seconds (see Figure 7).

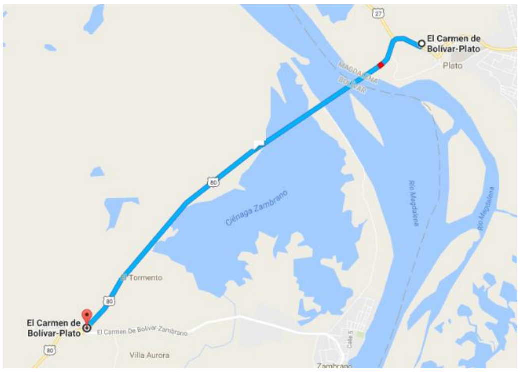

Figure 7. Partial route of the stretch truck of the route between San José del Purgatorio (Magdalena) to Turbaná (Bolívar)

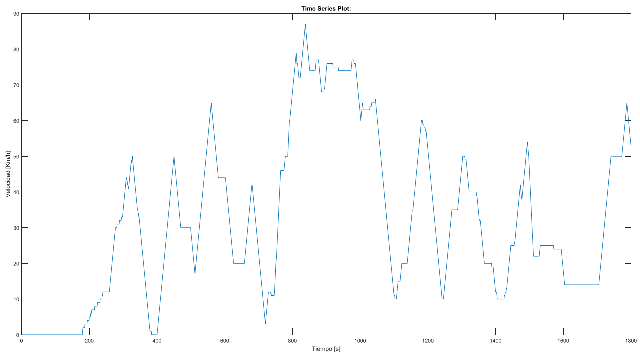

Figure 8. Real speed profile stretch of the route between San José del Purgatorio (Magdalena) to Turbaná (Bolívar)

2.1.5. Regeneration hydraulic circuit design

Within the possible configurations for the braking energy regeneration system (Series, Parallel and Split) the configuration was chosen in parallel, as it does not require a change in the mechanical transmission of the vehicle and is suitable for driving cycles with repeated accelerations and decelerations.

The design of this hydraulic circuit considered switching between the three modes of operation, the retention of pressurized oil until required, and the synchronization between the speed profile and the loading and unloading cycles of the accumulator [24].

The circuit distinguishes three modules, capture, confinement and reintegration module, all three refer to energy. The displacement control of hydraulic devices was carried out using fuzzy logic.

3. Results and discussion

This model was simulated with two speed profiles, as discussed above. In addition to that, fixed and variable displacement hydraulic pump/motor was implemented. For the NEDC speed profile, a fixed and variable displacement pump/motor arrangement was arranged, so the same was done with the actual speed profile obtained by Navisaf.

In the graphs shown, the actual profile refers to the fraction of the road San José del Purgatorio (Magdalena) to Turbaná (Bolívar). The results of the simulations are displayed under different conditions, for the two speed profiles the use of an uncontrolled device at its displacement and with variable displacement was evaluated.

The simulations compared the torque required by each of the profiles based on the required torque, system-regenerated torque, and the final torque required with the regeneration system.

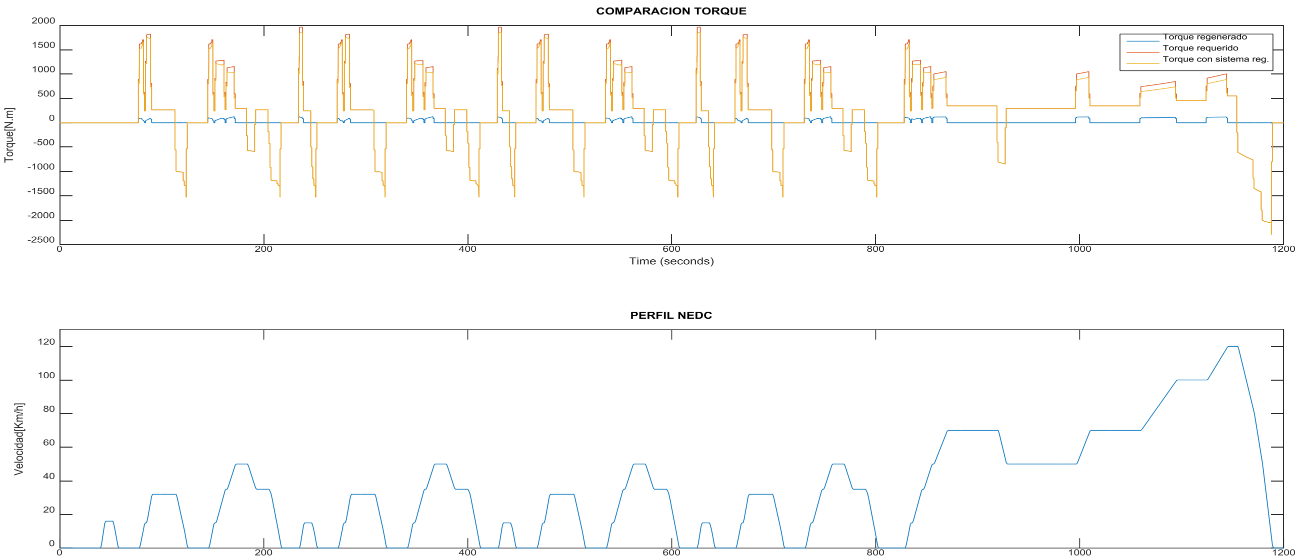

Figure 9. Comparison of torque para the NEDC profile with fixed displacement

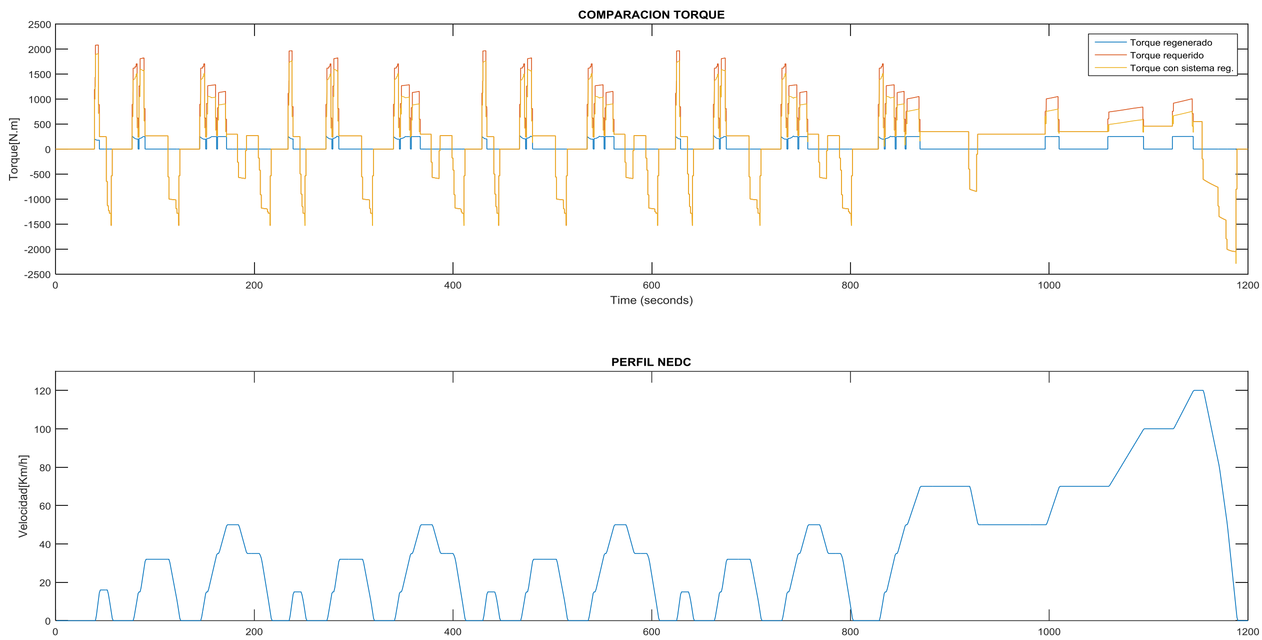

Figure 10. Comparison of torque para the NEDC profile with variable displacement

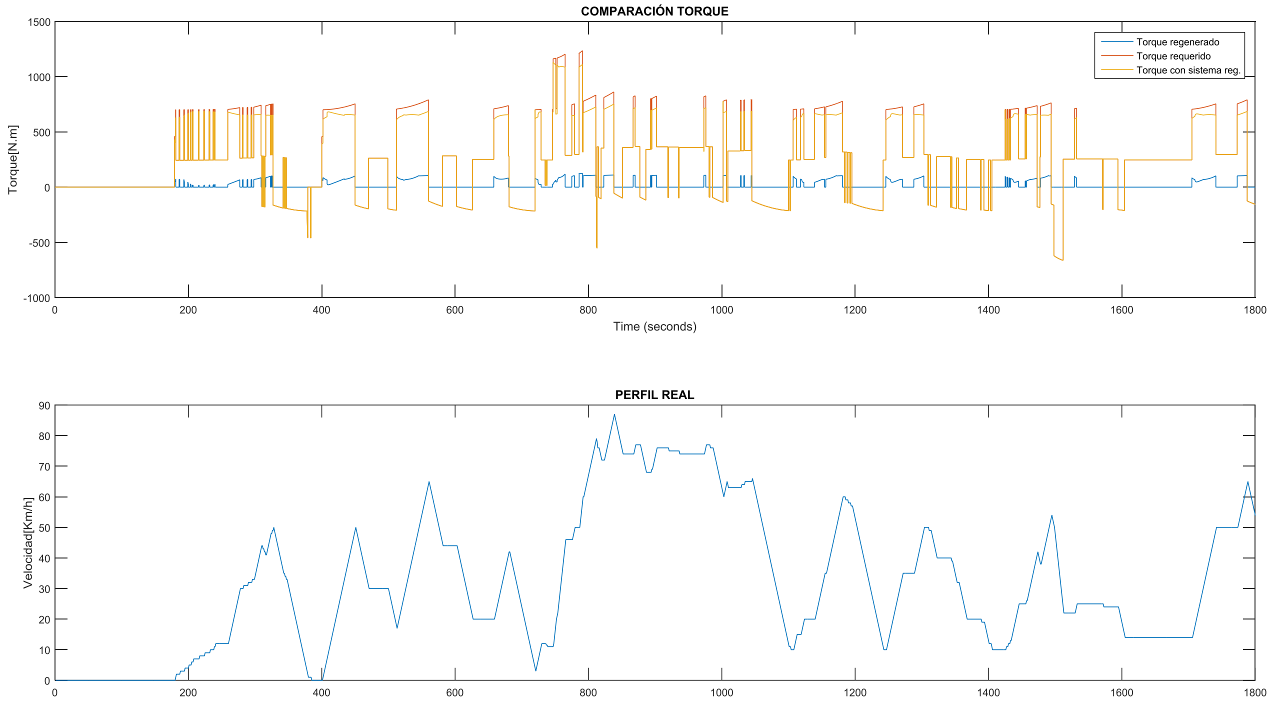

Figure 11**.** Torque comparison for the actual profile with fixed displacement

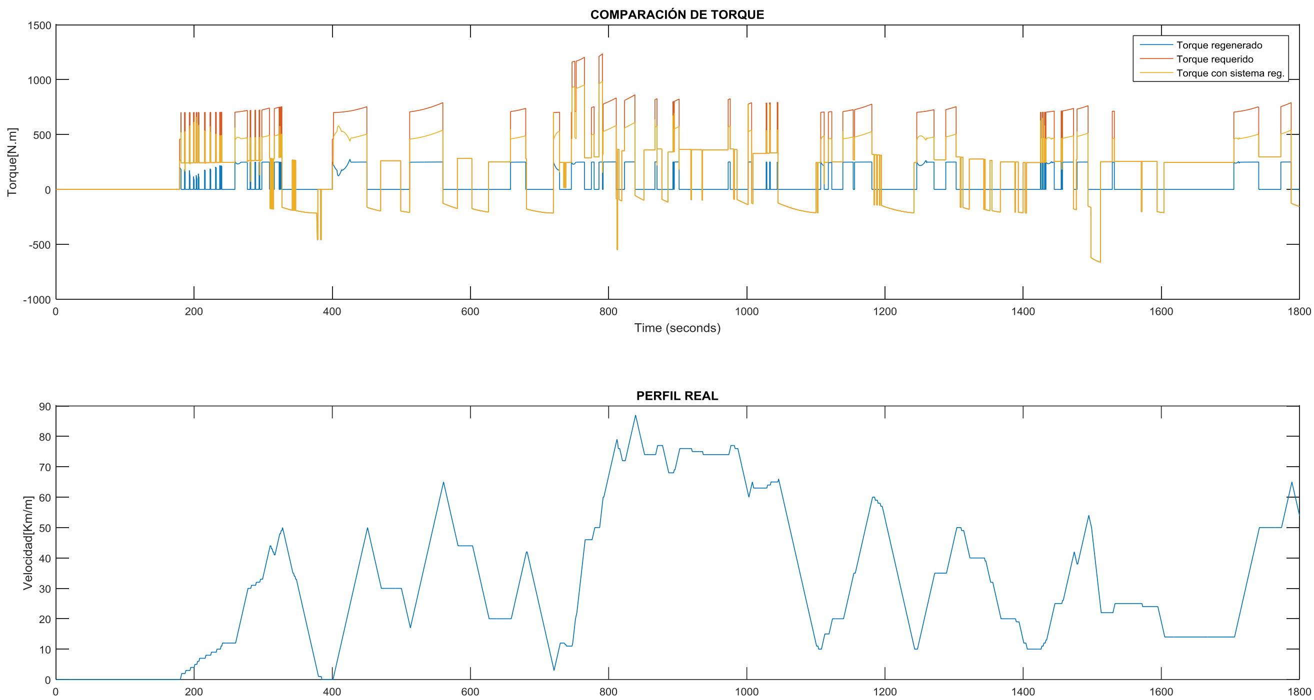

Figure 12**.** Comparison of actual profile torques with variable displacement

By performing the same analysis on one of the profiles and configurations, the comparison of the required power based on time was estimated the percentage of energy savings.

The percentages of energy decrease required depending on the speed profile and the pump/hydraulic motor displacement are shown in Table 1.

Table 1. Percentages of energy savings required

Speed Profile	% Decrease in required energy	
	Fixed displacement	Variable displacement
NEDC	16%	22%
Real profile	15%	23%

4. Conclusions

In accordance with the proposed objectives, a parallel-configured braking energy regeneration system was modeled and simulated for a 15-tonne heavy-duty vehicle, Two speed profiles, a standard profile and a real profile of an intermunicipal courier vehicle were simulated and the torque and power requirements WITH and WITHOUT energy regeneration system were compared obtaining a 15.5% savings percentage on average for a fixed displacement pump/motor system and 22.5% on average for a pump/motor system with variable displacement.

The energy regeneration system is most efficient when the difference in speeds between braking stages is greater, these results are supported by publications indicating that such recoverable energy is defined as a means of mass by the square of speed.

The use of the variable displacement system increases energy regeneration efficiency when kinetic energy when speed deltas are low, increasing by approximately 7%.

The savings percentage for the NEDC speed profile was 19% on average, taking into account the fixed and variable displacement of the pump/hydraulic motor.

The percentage of savings for the actual speed profile, which covered a stretch of the route between San José del Purgatorio (Magdalena) to Turbaná (Bolívar) was 19% on average, taking into account the fixed and variable displacement of the pump/hydraulic motor.

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