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Efficiency sensitivity analysis of a hydrostatic transmission for an off-road multiple axle vehicle

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Abstract

There is a large variety of multiple driven axle vehicles. Some of the most common are the 3-axle rigid vehicles and the 4-axle articulated vehicles, which can in some cases have different steering mechanisms, adaptive suspension, etc. This last kind of vehicles usually have very complex transmission configurations. Moreover, the required torques in each of the wheels can be very different, especially when the vehicle is working in rough terrains. The aim of this work is to study and model the driveline of this kind of vehicles, when using a hydrostatic transmission, from the performance and efficiency point of view, by analysing the influence of the operating conditions in the transmission efficiency. A global model is used to quantify the power flow in each of the transmission elements and the overall performance of the entire vehicle driveline, given the operating conditions thereof.

A sensitivity analysis has also been done showing the influence of vehicle speed, rolling resistance, terrain slope and hydraulic motors displacement in the overall transmission efficiency.

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The interest of this work is also to make a contribution to the literature in the field of global modelling of drivelines under variable operating conditions and its application to ATVs. One important aspect is the influence of different actuation requirements that occur in different wheels at the same time.

The results show that the overall performance of the transmission is highly dependent on operating conditions, on the selected transmission configuration and on the used components.

Keywords

Hydrostatic, transmission, vehicle, driveline, modelling, efficiency, performance, multi-axle, off-road, ATV, sensitivity, operating conditions.

Introduction

Nowadays there is a large variety of all terrain wheeled vehicles (ATVs) with many different configurations in relation to the number of drive axles, steering mechanisms, and requirements (Goering, 2003).

Some of these special vehicles have mechanisms that allow adapting the position of the wheels in relation to the chassis (adaptive suspension). Also different types of transmissions are used (mechanical, hydrostatic, electrical, mixed, etc.). However, the adaptive suspension may in some cases substantially hinder the power transmission from the engine to the wheels, so the use of a hydrostatic transmission is advisable in these cases.

In addition, vehicles used to overcome obstacles and with a considerable load capacity pose very different requirements in each of its axles (Potau et al., 2010).

The methodology for the design of this kind of transmissions is a complex and far from trivial process.

A rule-based expert system, called HSTX, has been developed to aid in the selection and sizing of the main components (pumps and motors) of a hydrostatic transmission for single-path configurations (Li et al., 1990).

The use of hydrostatic transmissions in agricultural and forestry vehicles is widespread. Nevertheless, there are relatively few studies covering the modelling of specific components of the transmission and even fewer studies focused on the interaction between components to analyse the efficiency of the whole driveline according to the operating conditions. There are some examples in the literature of modelling focused on the hydrostatic part of mechanical-hydrostatic mixed transmissions concerned with the mathematical modelling of the dynamic behaviour of the swash-plate mechanism of the variable displacement pump (Kugi et al., 2000) as well as the particularities of their integration in agricultural tractors using planetary gear trains (Linares et al., 2010).

Other studies have been found focused on minimizing the operating cost per unit of acreage in agricultural vehicles equipped with hydrostatic transmission, where a model that predicts the performance and the efficiency of a hydrostatic transmission for both maximum and partial flow is obtained (Pacey et al., 1983).

Other authors have relied their studies on Bondgraph simulation techniques in order to model the dynamic performance behaviour of open loop hydrostatic transmission systems (Dasgupta, 2000).

Some studies are related to hybrid technology, in which the factors that influence energy consumption in urban driving conditions are analysed for vehicles with electric-hydrostatic mixed transmission and regenerative braking strategies (Hui et al., 2008).

Validation studies of the performance models of different transmission components using test rigs can also be found in the literature (Czyński, 2008).

Some static and dynamic simulation tests using constant and variable efficiency values for the main elements have been done to evaluate the transmission efficiency using MATLAB-SIMULINK software (Jędrzykiewicz et al., 1997).

Finally, the integration of all these aspects in the vehicle driveline and the evaluation of the overall performance in terms of vehicle operating conditions has been carried out by other researchers for the case of mechanical transmission (Yi et al., 2007).

Also noteworthy are the studies found in the literature on the energy consumption of hydraulic systems in construction machinery (Zimmerman et al., 2008), although no references have been found for the case of a hydrostatic transmission driveline aimed at a vehicle with a high ability to surmount obstacles.

On the other hand, there are studies focused on the conceptual description of the design procedure and the performance analysis of a hydrostatic transmission for multiple axle vehicles intended to overcome obstacles (Comellas et al., 2010).

This paper presents an efficiency sensitivity analysis of the hydrostatic transmission in this type of vehicles, quantifying the power flow in each of the transmission elements and the overall performance of the whole vehicle driveline, depending on its operating conditions.

Studied vehicle

This study is based on a vehicle consisting of 2 modules linked by a double articulation joint, with 8 wheels grouped in 4 bogie assemblies, 2 per module. The double articulation joint and the bogie mechanisms allow the 8 wheels to adapt their position in order to ensure the contact of all the wheels with the ground.

When operating in a slope or in rough terrain, every wheel can have different actuation requirements. So the transmission system has to provide the corresponding torque and power to each wheel.

The vehicle must be able to operate in terrains with different surface characteristics and different slopes up to 35º.

In Figure 1 a scheme of the vehicle is shown and the general specifications are summarized.

The vehicle concept on which the study is based is shown in Figure 2, where it can be seen a simulation of the vehicle overcoming an obstacle.

This all-terrain vehicle is aimed to be driven through irregular terrains. One possible application is the use in forestry areas, where there are slopes and obstacles not accessible with a conventional 2-axle vehicle.

Mathematical model of transmission components

This section analyses the power losses occurring in the different components of the transmission that have an effect in its overall efficiency. These losses can be classified as:

- Volumetric and hydromechanical losses in hydraulic pumps and motors
- Pressure drops in hoses and singular elements of the hydraulic circuit
- Mechanical losses in both gear reducers and chain mechanisms

Power losses of every transmission component depend on their working conditions. Once the operating condition of the vehicle is known, that is the speed and torque required at every wheel, and power losses can be estimated, it is possible to calculate the power requirements of the engine that provides the necessary mechanical power.

Regarding to the volumetric losses, in any hydraulic component, pump or motor, there is a leakage flow dependant on its construction, its internal tolerances and the specific working conditions: speed, pressure, viscosity of the fluid, temperature, etc.

The clearances between moving parts in hydrostatic machines are relatively small, typically of the order of tens of micrometers. Correspondingly, Reynolds numbers for the leakage flows are low, and flow is laminar (Burrows and Vaughan, 1988). As a result, if it is considered as a typical pressure drop in pipes, the friction coefficient is inversely proportional to the fluid velocity, so the pressure drop is proportional to the flow.

It can be considered then that the leakage flow is proportional to the pressure difference inside the component. To this end, a constant of proportionality for the pump and another for the motor is defined (k_{vp} and k_{vh}). These constants establish the dependence between the leakage flow (q_l) and the pressure difference as shown in Eq. (1), where ΔP_i is the pressure difference between the high-pressure line and the drainage channel, and the subscript (i) is (h) for hydraulic motors or (p) for hydraulic pumps.

$$q_{li} = k_{vi} \cdot \Delta P_i \tag{1}$$

Hydromechanical losses in hydraulic pumps and motors include hydraulic losses, which are pressure losses due to the viscosity of the fluid, and mechanical losses, due to the friction caused by the relative motion between elements, such as bearings, pistons, etc.

Mechanical losses cause a pressure loss that can be considered constant, also known as no load pressure loss (ΔP_{lci_0}). This is the pressure difference between inlet and outlet that is required to keep the pump or the motor rotating. Hydraulic losses can be considered proportional to the circulating flow squared. They have been calculated as a typical pressure drop in hoses where there are some unknown parameters that have been grouped in a hydromechanical constant (k_{hmi}) as it can be seen in Eq. (2).

As a result, the calculated pressure loss (ΔP_{lci}) is obtained from Eq. (2), where (k_{hmi}) is the hydromechanical constant for the component. So the real pressure increase at the pumps (ΔP_{rp}) is the difference between the theoretical pressure increase (ΔP_{tp}) and the calculated pressure loss (Eq. (3)). The real pressure decrease at the motors (ΔP_{rh}) is the theoretical pressure decrease (ΔP_{th}) plus the calculated pressure loss (Eq. (4)).

$$\Delta P_{lci} = \Delta P_{lci\_0} + k_{hmi} \cdot q_{ini}^2 \tag{2}$$

$$\textit{Pump: } \Delta P_{rp} = \Delta P_{tp} - \Delta P_{lcp} \tag{3}$$

$$\textit{Motor: } \Delta P_{rh} = \Delta P_{th} + \Delta P_{lch} \tag{4}$$

The values of the constant parameters (volumetric k_{vi} and hydromechanical k_{hmi}) used in the mathematical model definition of each of the components have been fit to adjust the model efficiency curve to the manufacturer data.

The pressure drops in the piping of the hydraulic circuit have been modelled as the sum of the pressure drops in hoses and the pressure losses due to singular elements (valves, elbows, etc.). They have been obtained using the common model of Darcy-Weisbach, as it is described in Eq (7), where (f) is the hose friction coefficient, (L) is the hose length or the equivalent length of the singularity, (D) is the hose inner diameter, (v_{oil}) the fluid velocity, and (g) is the gravity acceleration.

The hose friction coefficient has been calculated using the Colebrook-While expression, with the simplification of Swamee-Jain, under turbulent flow for all the situations analysed. Although the Reynolds number range obtained in this study includes situations for both laminar and turbulent flow in the hydraulic circuit, only turbulent flow has been considered. The reason for doing this assumption is because when fluid first enters a pipe (after a singularity) its flow is not fully developed. Instead the fluid has to travel a certain distance undisturbed before it becomes fully developed. This is also true when a fluid goes around a curve in the pipe system. The curve in the pipe will disrupt the velocity profile of the fluid, and it will need to travel a certain distance in a straight pipe to have a fully developed flow again.

In the hydraulic circuit there are a lot of valves, elbows, curved hoses, etc., and they make it difficult to achieve a fully developed laminar flow downstream one of these singularities. That is why only turbulent flow has been considered along the hydraulic circuit for all the situations.

$$\Delta P_{pipe} = f \cdot \frac{L}{D} \cdot \frac{{v_{oil}}^2}{2 \cdot g} \tag{7}$$

Mechanical losses in gearboxes and chain mechanisms increase with the torque transmitted, so they have been modelled as derived from a friction torque in the output shaft proportional to the torque transmitted (see Eq (8)), regardless of the angular velocity. The resulting efficiency can be evaluated according to Eq (9).

$$M_f = k_f \cdot M_{out} + M_{f\_0} \tag{8}$$

$$\eta_{mec} = 1 - \frac{M_f}{M_{in}} \cdot \tau \tag{9}$$

Analysed transmission configuration

The analysed transmission configuration comprises two hydraulic pumps, four hydraulic motors (one per bogie) and eight chain mechanisms. The hydraulic motors are placed on the bogie at the same distance between both corresponding wheels.
Two operation modes have been considered, regarding the connection between pumps and hydraulic motors. They can be selected in every situation depending on the operating requirements of the vehicle.
The operation modes are described as S.O.M. and P.O.M., and Figure 3 shows the corresponding layouts of the hydraulic and mechanical connections.

- **S.O.M.** (Series Operation Mode): The output of both pumps are joined and the flow is directed to the group of 2 motors of the front module of the vehicle, and then to the group of 2 motors of the rear module. The 2 groups of motors are connected in series. This operation mode allows high speed of the vehicle, since the total flow goes through the 2 groups of motors.
- **P.O.M.** (Parallel Operation Mode): Each pump is connected to a group of motors, so they define 2 independent circuits that operate in parallel. This mode allows high torque of the motors, since the whole pressure difference is used in only one group of motors.

Hydrostatic transmission components

In this example two different groups of commercial components have been selected for the study of the transmission performance. In order to have comparative results, the only difference between them is the displacement of the hydraulic motors. A brief description of the main features of each component is provided.

- Hydraulic pumps: The selected pumps (Bosch Rexroth A4VG/40) are reversible for closed circuit, with axial pistons, variable displacement and inclined plate. Their maximum displacement is 45.3 cm^3/rev.

In Figure 4 it can be seen, as an example, the efficiency curves as a function of the input speed of the pump shaft. They have been obtained from the model that has been previously described.

- Hydraulic motors: The selected motors (Poclain Hydraulics MS 02) have radial pistons with fixed displacement, being 213 cm^3/rev and 255 cm^3/rev for the group of Components 1 and 2, respectively.

Determination of the overall transmission efficiency

The parameters used in the mathematical model definition for each of the elements have been deduced from the manufacturer data.

The global transmission efficiency has been evaluated as a ratio between the sum of the output powers transmitted to each of the vehicle axles and the sum of mechanical power to drive the two main hydraulic pumps and the feedback pump, which has to be equal to the power provided by the combustion engine Eq. (9).

In Figure 5 it can be schematically observed the power data parameters for the global transmission efficiency analysis.

$$\eta_{tot} = \frac{Power_{Out}}{Power_{C.E.}} = \frac{Power_{F1} + Power_{F2} + Power_{R1} + Power_{R2}}{Power_{p1} + Power_{p2} + Power_{feedback}} \quad (9)$$

In all the cases and results shown in this work, the limits in the operating conditions of the components have been taken into account. So, it has never been overtaken, under any circumstance, the maximum pressure of 450 bar in any point of the hydraulic installation. The hydraulic pumps have been limited to 45.3 cm^3/rev. of displacement per revolution. The hydraulic motors never operate with a power demand higher than 18 kW.

The refrigeration and renovation oil flow of the closed loop, extracted from the low pressure pipe, has been calculated from the relieve valve actuation depending on the pressure at which it is working.

The power consumed by the feedback pump has been calculated taking into account that it provides a constant flow, as it is an 11 cm^3 gear pump, at a 10 bar pressure increment. The amount of flow that is not injected to the closed loop is rerouted to the oil accumulation tank.

Actuation requirements of the different axles

The actuation requirements in the axles of the vehicle are defined as the angular speed and torque needed to run in a specific operating condition.

Usually, each of the vehicle axles has different requirements. Adding the complex configuration of the vehicle transmission is what makes difficult to transfer the power from the engine to the wheels.

These two parameters (angular speed and torque) adopt different values depending on the operating conditions (vehicle speed, terrain slope, and terrain rolling resistance).

In this study, the actuating torque in each axle has been considered as proportional to the corresponding normal load. Then, all the axles should begin the slippage at the same time.

So, as the vehicle is running on a planar sloped surface, the needed angular speed in each of the wheel axles is the same for all of them, however, the torque requirements are very different in each axle.

Figure 6 shows different operating scenarios where it can be seen the actuating torque in each axle.

Efficiency sensitivity analysis

The influence of the vehicle operation requirements (vehicle speed, terrain slope and rolling resistance) and the transmission parameters (operation mode and hydraulic motor displacement) over the total efficiency is analysed.

The previously defined mathematical global model has been used to evaluate the efficiency values obtained in the different operating conditions.

In a first comparative study it has been analyzed the vehicle speed influence over the global transmission efficiency for different particular terrain slopes values, varying from 0º to a maximum of 35º, with a constant rolling coefficient of 0.05, with the 213 cm^3 displacement motor and for both operation modes (series and parallel) (Eq (10)).

$$\eta_{tot} = f(v, \beta_t, k_r = 0.05, motor\,disp. = 213\,\mathrm{cm^3/rev.}, S.O.M., P.O.M.) \qquad (10)$$

A second sensitivity analysis shows the total transmission efficiency as a function of the terrain slope, given a vehicle speed (3 km/h) and several rolling resistance values (from

0 to 0.25) with the 213 cm^3/rev. displacement motor and for both operation modes (series and parallel) (Eq (11))

$$\eta_{tot} = f(\beta_t, k_r, v = 3\,\text{km/h}, motor\,disp. = 213\,\text{cm}^3/\text{rev}., S.O.M., P.O.M.) \tag{11}$$

After that, a sensitivity analysis to compare the hydraulic motors displacement influence has been done using two different hydraulic motors models, one of them with a 213 cm^3/rev. displacement and the other one with a 255 cm^3/rev. displacement.

Results and discussion

First, the influence of the operating conditions is analysed. Figure 7 shows a simplified scheme of the two operation modes (series and parallel) for the same actuation conditions. The flows and the pressure difference at the elements can be comparatively observed. Table 1 shows, for the hydraulic pumps and motors, if the flow and the pressure in the circuit are lower, higher or equal to the opposite operation mode.
It can be deduced that, comparatively, for a parallel operation mode the higher circulating flow through the pumps implies a reduction in the global efficiency compared to a series operation mode. On the other hand, the detrimental aspect in a series operation mode is the higher pressure reached both in pumps and in the inlet of the first group of hydraulic motors.

In Figure 8 the transmission efficiency as a function of the vehicle speed for different terrain slopes can be observed. It can be seen that at low speeds the efficiency is always higher in a parallel compared to a series operation mode regardless of the terrain slope. This is due to the fact that in the parallel mode the hydraulic circuits of each of the vehicle's modules works independently (as far as flow and pressure concerns), which allows the maximum circuit pressure to be lower. In both operation modes (series and parallel) the circulating flow through the motors when the vehicle moves at a certain speed is almost the same, as the speed is also the same. However, in a series operation mode the pressure has to be higher to reach the required torques in each of the axles and it decreases in cascade from its maximum value at the inlet of the first group of motors to the suction pressure before the pump inlet.

As previously mentioned, the flow is the detrimental effect in a parallel operation mode. As the vehicle runs at a low speed there is a low flow rate and its influence in losses is lower.

When the vehicle speed increases and the required torque conditions are maintained, the losses due to the high flow have a greater importance. While the flow is the same in the motors, the flow in the pumps doubles in the P.O.M.

In a parallel operation mode it is not possible to reach the whole vehicle speed range because the maximum pump displacement limits the output flow.

In P.O.M., the maximum attainable speed decreases as the terrain slope gets higher. The reason is that for higher slopes higher pressure is needed, and then it leads to greater flow losses (leakage flow, both in pumps and motors) and a reduction of the effective flow.

In the series operation mode the limiting aspect is the maximum pressure reached in the circuit. That is why the cases with a terrain slope higher than 20º are not technically feasible with this operation mode.

For high speed and high slope simultaneously is not possible to reach the maximum specified speed of 30 km/h. The maximum power for the hydraulic motors (18 kW) is exceeded.

It is important to notice that the global transmission efficiency is very variable depending on the operating conditions, and its maximum value for a given slope can range between a 25% when the vehicle runs in a flat terrain to 70% in maximum slopes. It can be appreciated that the global transmission efficiency is null when the speed is 0 km/h, as there is no output power, and that for low speeds the global efficiency is very low as it is necessary to overcome the no load losses and the output power is low. The efficiency increases rapidly when the vehicle speed rises as the output power also increases. That happens from 0 to around 5 km/h, thereafter the influence of the speed increase is not as significant. It can be observed that in some of the studied scenarios there is a speed from which the efficiency line takes a slightly negative slope value. This is the speed at which the losses caused by the speed increase begin to be higher than the output power increase. With the considered model, the efficiency of the mechanical elements of the transmission depends on the transmitted torque but is independent on the working speed.

The crossing points of the efficiency lines for a series operation mode and a parallel operation mode (Figure 8) are the points where the excess of losses due to the pressure (volumetric losses) in motors and pumps when the transmission is operating in series are equivalent to the excess of losses due to the flow rate (hydromechanical losses) in pumps operating in parallel.

As an example of this effect, the power losses for the scenario conditions at the efficiency crossing point for a terrain slope of 5º (Figure 8) have been analysed in more detail. At this point, the scenario operating conditions are:

Vehicle speed = 13.5 km/h

Terrain slope = 5º

Rolling resistance coefficient = 0.05

In these conditions the minimum friction coefficient is 0.1382 and the total efficiency is 0.4585 for both series and parallel operation modes.

The volumetric and hydromechanical losses for both, series and parallel operation modes, have been calculated in this scenario. The pressure inside the elements (hydraulic pumps or hydraulic motors) causes a leakage flow which is considered as a volumetric loss, its power loss ($Pow_{loss,vol,i}$) can be calculated multiplying the leakage flow (q_{li}) by the difference between the high pressure ($P_{high,i}$) and the reference pressure (P_{ref}) (Eq (13)). On the other hand, the flow rate circulating throughout the elements and the friction between their moving parts causes a pressure loss, which is known as a hydromechanical loss and its power loss ($Pow_{loss,hyd,i}$) is calculated multiplying the inlet flow (q_{ini}) by the pressure losses at the element (ΔP_{lci}) (Eq(14)). Again the subscript (i) is (h) for hydraulic motors or (p) for hydraulic pumps.

$$Pow_{loss,vol,i} = q_{li} * \left(P_{high,i} - P_{ref}\right) \tag{13}$$

$$Pow_{loss,hyd,i} = q_{ini} * \Delta P_{lci} \tag{14}$$

In the considered scenario conditions, the mechanical power losses in the chain mechanisms are the same for both, series and parallel mode, and it has been assumed that the power losses in hoses and singular elements of the hydraulic circuit are not very different depending on the operation mode (series or parallel). So the excess of volumetric or hydromechanical losses using a series or parallel operation mode can be seen in Figure 9.

In order to have more information at this point (Figure 8) some essential values of circulating flow and pressure difference in the main parts of the hydraulic transmission have been specified for the S.O.M. and P.O.M. (Table 2). As it can be seen, the pressure difference in hydraulic motors for both, the series and parallel operation modes, are the same as the required torque in the wheels axles is also the same. However, the pressure difference in the hydraulic motors is lower than the pressure difference provided by the pumps due to the hydraulic losses in hoses. The pressure difference provided by both pumps in P.O.M. is slightly higher than in S.O.M. because the circulating flow is also higher and then there are more hydromechanical losses (pressure loss).

On the other hand, the flow provided by both pumps in S.O.M. is higher than in P.O.M. because the pressure difference is also higher and then there are more volumetric losses, or what is the same, there is a bigger leakage flow.

In Figure 10, the transmission efficiency, operating both in series and parallel, as a function of the terrain slope for a vehicle speed of 3 km/h for several rolling resistance values is represented.

It is shown that the maximum efficiency values reached are around 35% using a series operation mode while in parallel an efficiency of up to 55% can be reached.

For a 0º slope the transmitted power already has a significant value when the rolling resistance is not null, being this the needed power to overcome the rolling resistance.

It should be obvious that for a 0º terrain angle a higher efficiency is obtained when the rolling resistance is bigger, for both the parallel and the series operation mode, as the output power is increased and the no load losses have a diminished effect. On the other hand, at 0º slope, there is a difference in the efficiency value depending on the operation mode that is used. For low rolling resistances, this difference is almost null. However, there is a significant difference for high k_r values. Since a low speed situation is studied, the reason is that the influence of the hydromechanical losses caused by the flow in parallel is lower than the influence of the volumetric losses caused by the pressure difference due to the moving vehicle resistance influence. For slopes steeper than 12º and 20º using a series and parallel operation mode respectively, the rolling resistance influence over the global transmission efficiency is negligible. It can be noted the importance of a proper tire pressure when running in non steep slopes, whereas at steep slopes the tire pressure can be reduced in order to improve the vehicle contact surface without worsening the global efficiency. In the same way as the analysis of the global efficiency as a function of the speed, it is observed that there is a terrain angle from which the efficiency line takes a lightly negative slope value. This is the angle from which the losses due to the pressure increase because of the increase in the terrain slope begin to be higher than the transmitted power increase.

Lastly, the results for the influence of the hydraulic motors displacement are analysed. As it has been previously commented, two different hydraulic motors with different displacements (213 cm^3 and 255 cm^3) have been considered. It is noteworthy that the 255 cm^3 displacement motors need less oil pressure than the 213 cm^3 displacement motors to achieve the same output shaft torque, but at the same time, more flow is needed by them to achieve the same shaft speed (Table 3).

In Figure 11 and Figure 12 the transmission efficiency as a function of speed, terrain slope and for a rolling coefficient of 0.05, for both hydraulic motors (213 cm^3/rev. and 255 cm^3/rev.) and for both operation modes (series and parallel) is represented. Figure 11, shows the results for the series operation mode while Figure 12 shows the results for the parallel operation mode.

Observing both pictures (Figure 11 and Figure 12) it can be seen that at low speeds and steep slopes better efficiencies are obtained with the 255 cm^3 motor in both operation modes. On the other hand, for high speeds and not extremely demanding slopes better efficiencies are obtained with the 213 cm^3 motor. When high torques are demanded, with the 255 cm^3 motors a less pressure is needed and as a result the volumetric losses are lower although the hydromechanical ones are bigger, but as the flow is small its effect is also small. In contrast, at high running speeds, the 213 cm^3 motors provides higher revolutions with less flow and as a result less hydromechanical losses, and as the torque requirements are not heavy demanding, the volumetric losses have less influence in the efficiency.

It can be deduced then that independently from the operation mode, the hydraulic motor that will provide better efficiencies during the vehicle service life will depend on the vehicle main usage.

Conclusions

With an off-road multiple driven axle vehicle with a hydrostatic transmission configuration given, a mathematical model to analyse the theoretical performance and efficiency has been defined.

All the components have been integrated into the mathematical model of the transmission which evaluates the performance conditions of each of them, as well as the efficiency of the whole driveline.

The influence of different parameters (vehicle speed, terrain slope, rolling resistance and hydraulic motor displacement) on the overall transmission efficiency has been analysed. Sensitivity analyses have been done using the developed global mathematical model of the driveline.

It has been shown that the efficiency results obtained for the whole transmission are considerably lower than the combination of the maximum efficiency values of each of the components that make up the driveline. It is demonstrated that combining different components decreases the overall performance because they all never work simultaneously at their maximum efficiency working point. It is verified that the overall driveline efficiency depends on the operating conditions of each of the elements and the entire vehicle as a whole, as well as the relationship between elements and their operation mode (series or parallel).

At low slopes very low efficiencies are obtained because the transmission components are far from their optimal operating point.

It has been observed that when going at high speeds with high flow rate flowing throw the hydraulic elements, the hydromechanical losses have more influence when using a parallel operation mode. On the other hand, when the vehicle is facing a steep terrain slope and, consequently, high pressures are obtained in the hydraulic system, volumetric losses have more effect on the series operation mode.

It is noted that the hydraulic motor displacement influence is determinant to the overall transmission performance, showing that for high terrain slope and low speed a bigger displacement hydraulic motor is best suited while for low terrain slope and high speed a smaller displacement hydraulic motor is recommended. Therefore, the choice of the hydraulic motor that provides higher efficiencies during the vehicle life depends on its priority of use within the different possible operation scenarios.

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References

Burrows, C. R. and Vaughan, N. D. (1988). *Design, Modelling and Control of Pumps*. First Bath International Fluid Power Workshop. University of Bath, England.

Comellas, M., Potau, X., Nogués, M. and Roca, J. (2010) Análisis del funcionamiento de la transmisión hidrostática de un vehículo todo terreno de múltiples ejes. *XVIII CNIM – CVIII Congreso Nacional de Ingeniería Mecánica*, Ciudad Real, España.

Czyński, M. (2008) Energy efficiency of hydrostatic transmission. Comparing results of laboratory and simulation test. *Scientific Problems of Machines Operation and Maintenance* **2,** 154.

Dasgupta, K. (2000) Analysis of a hydrostatic transmission system using low speed high torque motor. *Mechanism and Machine Theory* **35**, 1481-1499.

Goering, C. (2003). Off-road vehicle engineering principles. *American Society of Agricultural Engineers.*

Hui, S., Ji-Hai, J. and Xin, W. (2008) Factors influencing the system efficiency of hydrostatic transmission hybrid vehicles. *IEEE Vehicle Power and Propulsion Conference (VPPC),* Harbin, China.

Jędrzykiewicz, Z., Pluta, J. and Stojek, J. (1997) Research on the properties of a hydrostatic transmission for different efficiency models of its elements. Acta *Montanistica Slovaca* **4**, 373-380.

Kugi, A., Schlachera, K., Aitzetmüllerb, H. and Hirmannb, G. (2000) Modeling and simulation of a hydrostatic transmission with variable-displacement pump. *Mathematics and Computers in Simulation* **53** 409-414.

Li, B., Heatwole, C.D. and Perumpral, J.V. (1990) Knowledge-base assistant for component selection and preliminary design of hydrostatic drives. *American Society of Agricultural Engineers*.

Linares, P., Méndez, V. and Catalán, H. (2010) Design parameters for continuously variable power-split transmissions using planetaries with 3 active shafts. *Journal of Terramechanics* **47** 323–335.

Pacey, D.A., Turnquist, R.O. and Clark, S. J. (1983) A coefficient model for a hydrostatic transmission. *American Society of Agricultural Engineers* **26,** 669-672.

Potau, X., Comellas, M., Nogués, M. and Roca, J., (2010) Comparison of different bogie configurations for a vehicle operating in rough terrain. *Journal of Terramechanics*, **45**, 75-84.

Yi, J., Yu, B., Hu, D. and Li, C. (2007). Research on modularized design and performance assessment based on multi-driver off-road vehicle driving-line. *International Journal of Automotive Technology* **8**, **3**, 375-382.

Zimmerman, J.D., Williamson, C.A., Pelosi, M. and Ivantysynova, M. (2008) Energy consumption of an LS excavator hydraulic system. *ASME International Mechanical Engineering Congress and Exposition. Proceedings* **4**, 117-126.

Nomenclature

q_{inh} Inlet flow for the hydraulic motor

q_{outh} Outlet flow for the hydraulic motor

q_{inp} Inlet flow for the hydraulic pump

q_{outp} Outlet flow for the hydraulic pump

q_{lh} Leakage flow for the hydraulic motor

q_{lp} Leakage flow for the hydraulic pump

k_{vh} Volumetrical constant for the hydraulic motor

k_{vp} Volumetrical constant for the hydraulic pump

ΔP_h Pressure difference between the high-pressure line and the drainage channel of the hydraulic motor

ΔP_{hF} Pressure difference between the high-pressure line and the drainage channel of the hydraulic motors of the front module

ΔP_{hR} Pressure difference between the high-pressure line and the drainage channel of the hydraulic motors of the rear module

ΔP_p Pressure difference between the high-pressure line and the drainage channel of the hydraulic pump

ΔP_{lch_0} No load pressure loss for the hydraulic motor

ΔP_{lcp_0} No load pressure loss for the hydraulic pump

ΔP_{lch} Calculated pressure loss for the hydraulic motor

ΔP_{lcp} Calculated pressure loss for the hydraulic pump

k_{hmh} Hydromechanical constant for the hydraulic motor

k_{hmp} Hydromechanical constant for the hydraulic pump

ΔP_{rh} Real pressure decrease at the hydraulic motor

ΔP_{rp} Real pressure increase at the hydraulic pump

ΔP_{th} Theoretical pressure decrease at the hydraulic motor

ΔP_{tp} Theoretical pressure increase at the hydraulic pump

ΔP_{pipe} Pressure loss in the hydraulic circuit pipes

f Hose friction coefficient

L Hose length or equivalent length of the singularity

D Hose inner diameter

v_{oil} Fluid velocity inside the hose

g Gravity acceleration

M_f Friction torque at output shaft

k_f Friction proportionality constant

M_{in} Input torque

M_{out} Output torque

M_{f_0} No load output torque

τ Gear ratio

η_{mec} Mechanical efficiency

η_{tot} Total efficiency

$Power_{F1}$ Output power at F1 axle

$Power_{F2}$ Output power at F2 axle

$Power_{R1}$ Output power at R1 axle

$Power_{R2}$ Output power at R2 axle

$Power_{Out}$ Output power

$Power_{p1}$ Input power at hydraulic pump 1

$Power_{p2}$ Input power at hydraulic pump 2

$Power_{feedback}$ Input power at feedback hydraulic pump

$Power_{C.E.}$ Combustion engine power

v Vehicle speed

β_t Terrain slope

k_r Rolling resistance coefficient

$S.O.M.$ Series operation mode

$P.O.M.$ Parallel operation mode

$Pow_{loss,vol,h}$ Volumetrical power losses at the hydraulic motor

$Pow_{loss,vol,p}$ Volumetrical power losses at the hydraulic pump

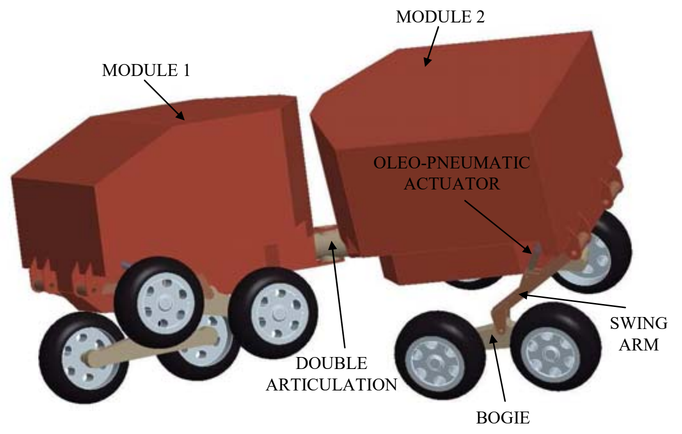
$Pow_{loss,hyd,h}$ Hydromechanical power losses at the hydraulic motor

$Pow_{loss,hyd,p}$ Hydromechanical power losses at the hydraulic pump

$P_{high,h}$ Pressure at the high pressure line of the hydraulic motor

$P_{high,p}$ Pressure at the high pressure line of the hydraulic pump

P_{ref} Reference pressure



Variable	Value
Length	3,600 mm
Width	1,500 mm
Height	1,800 mm
Wheel diameter	600 mm
Vehicle weight	1,200 kg
Load Capacity	800 kg
Maximum weight	2,000 kg
Maximum speed	30 km/h

Figure 1. Scheme and general specifications of the vehicle

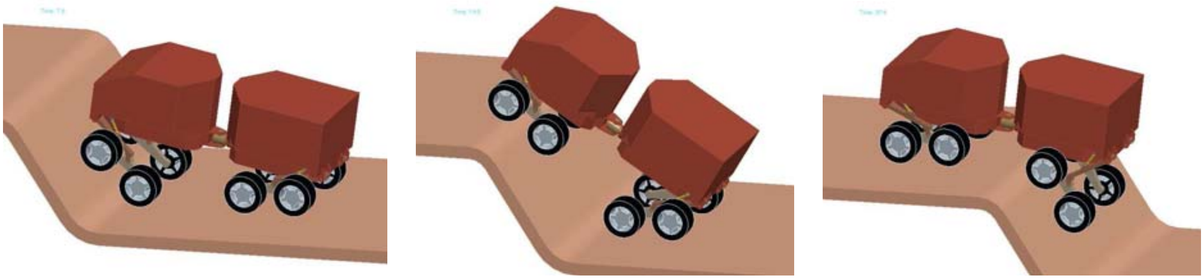


Figure 2. Simulation of the vehicle overcoming an obstacle

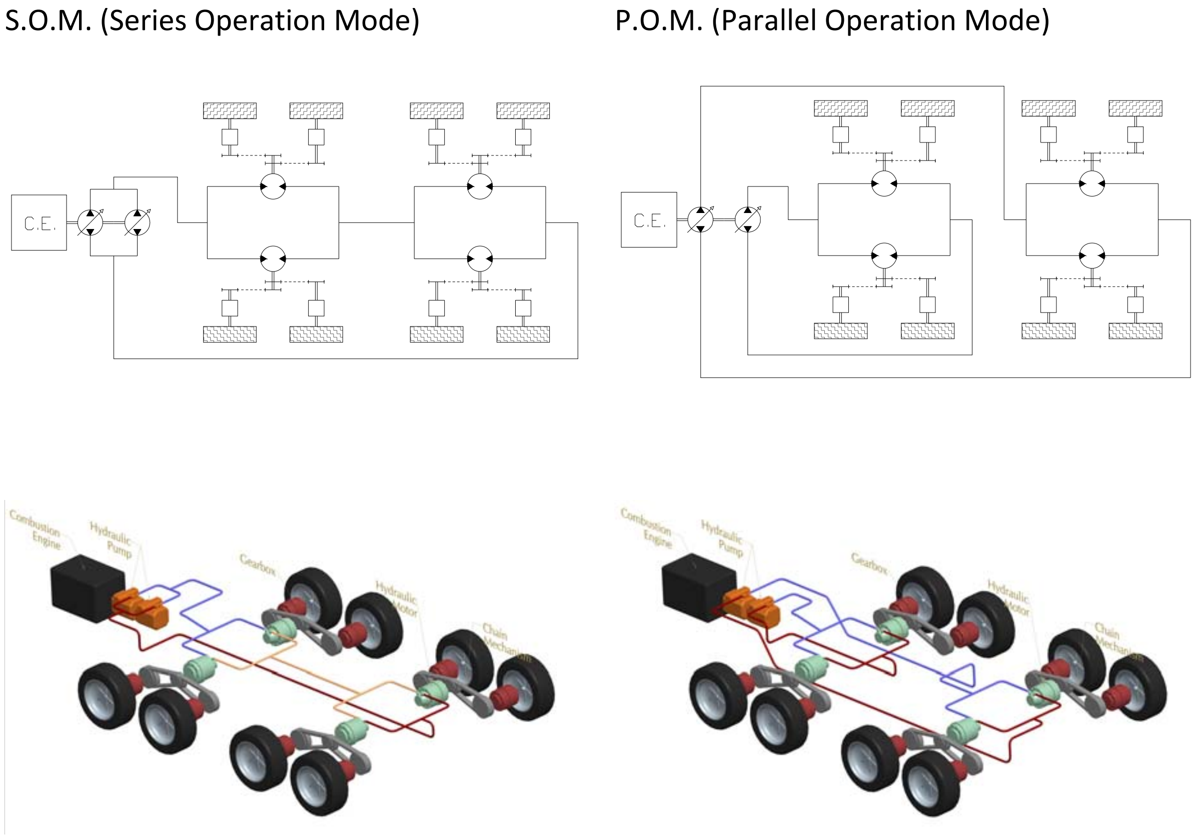


Figure 3. Layout of the configuration with the series operation mode (left) and the parallel operation mode (right)

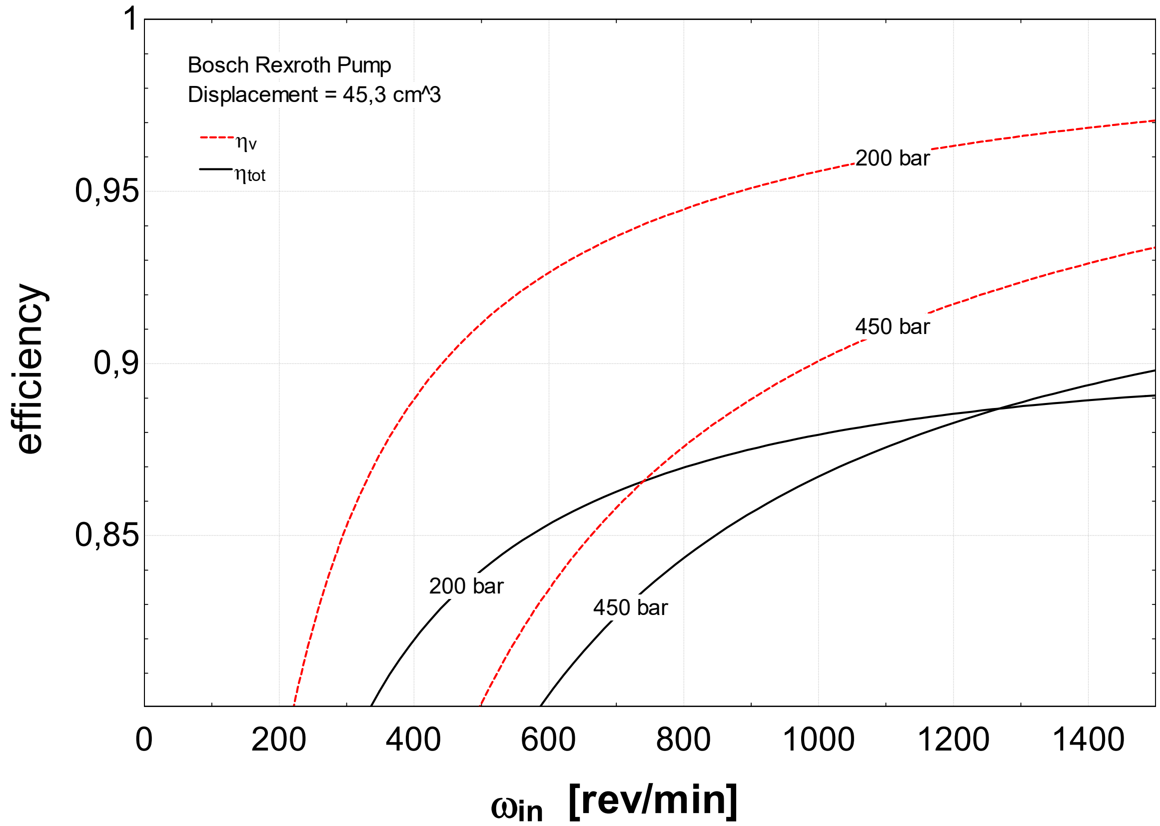


Figure 4. Efficiency curves as a function of the input speed of the pump shaft

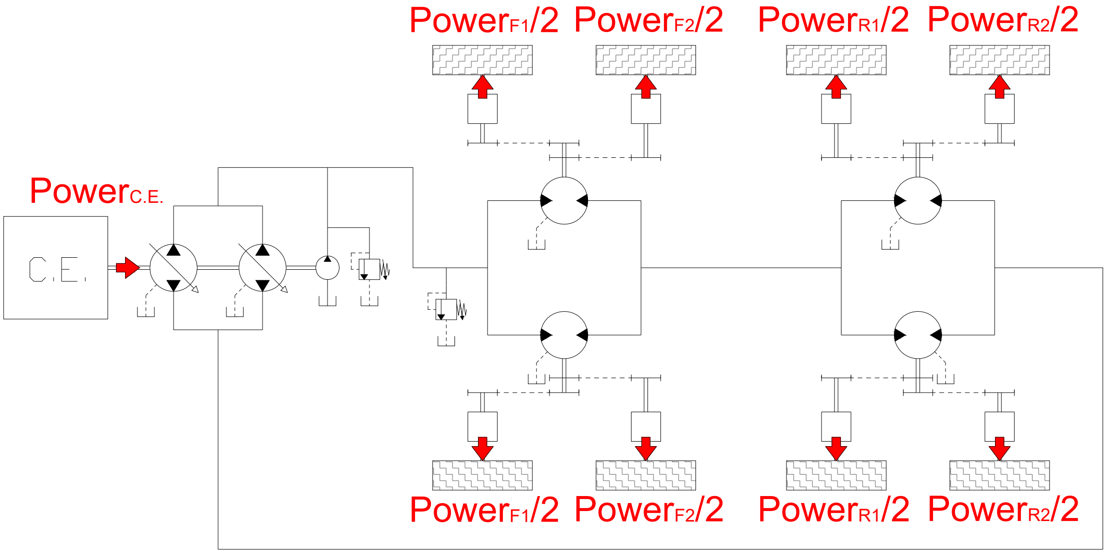
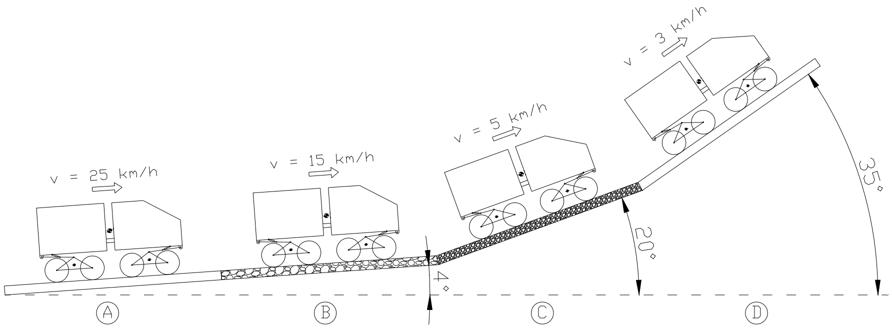


Figure 5. Power flows in the overall transmission



Operating conditions	Scenario A	Scenario B	Scenario C	Scenario D
Slope β (º)	4 (7%)	4 (7%)	20 (36%)	35 (70%)
Speed (km/h)	25	15	5	3
Rolling coefficient k_r	0.012	0.25	0.15	0.05
Required friction coefficient μ_{min}	0.09	0.33	0.54	0.78

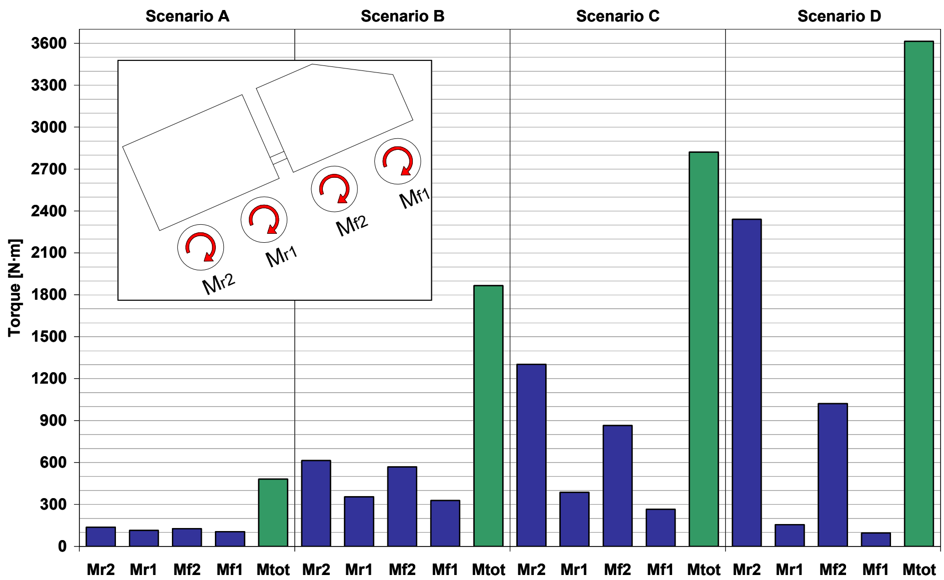


Figure 6. Actuating torque in each axle depending on the operating scenario

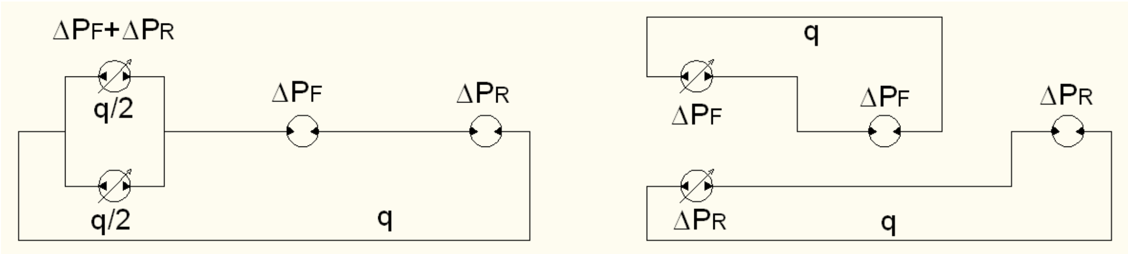


Figure 7. Simplified schemes of the operation modes, series (left) and parallel (right), for the same actuation conditions.

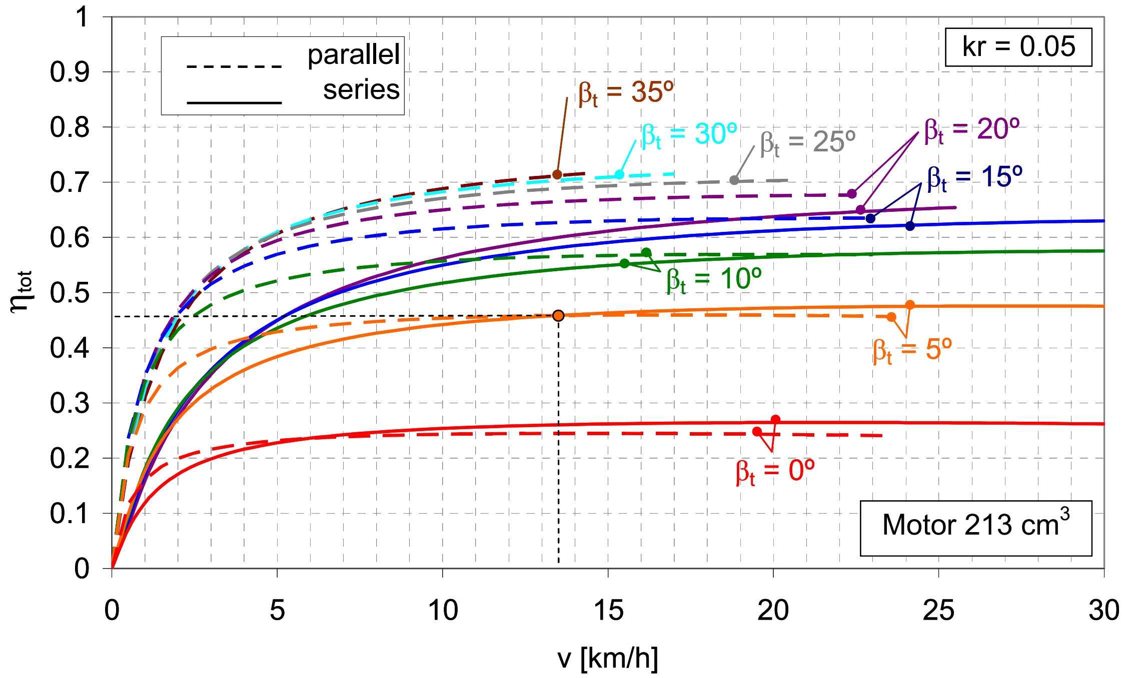


Figure 8. Efficiency as a function of vehicle speed and terrain slope

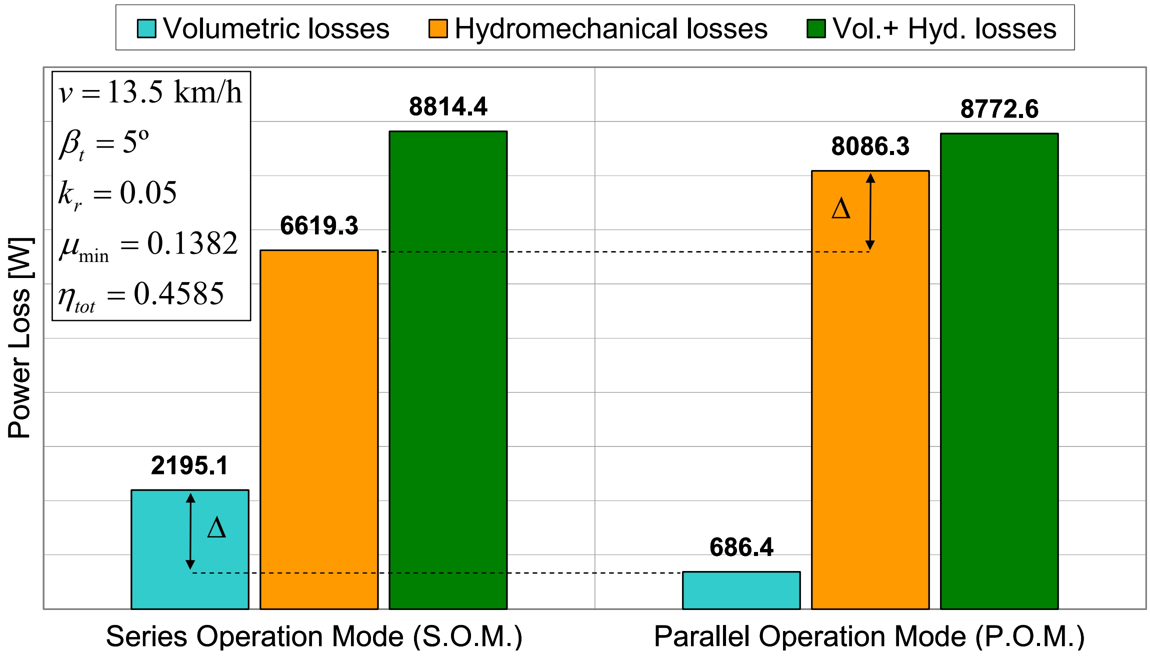


Figure 9. Volumetric and Hydromechanical losses in pumps and motors for the same scenario and with the same global transmission efficiency obtained

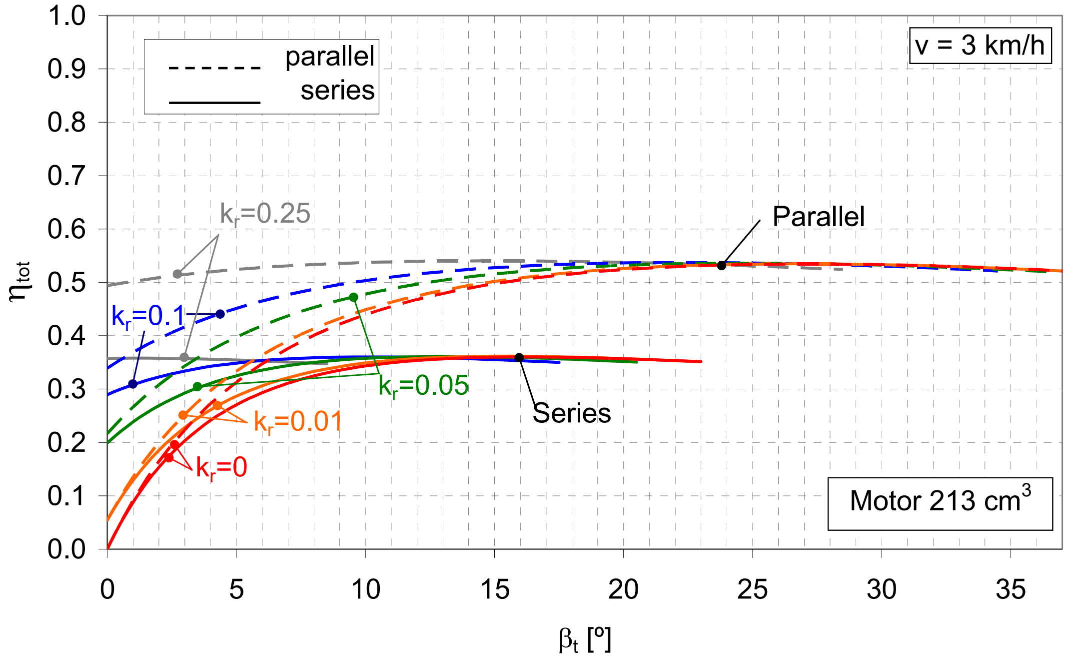


Figure 10. Efficiency as a function of the terrain slope and rolling resistance

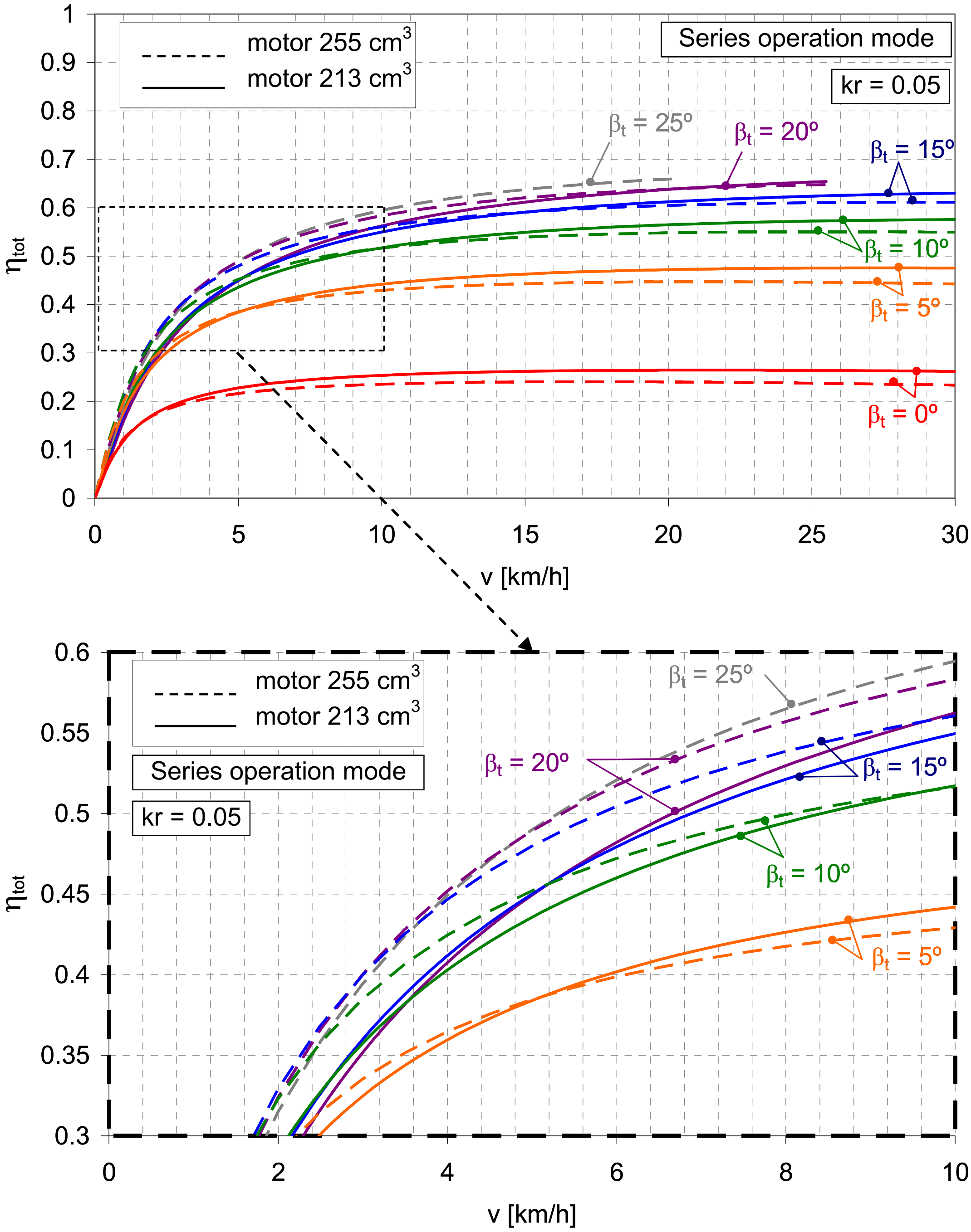


Figure 11. Efficiency as a function of speed and terrain slope in a series operation mode for the 213 and 255 cm^3 displacement hydraulic motors

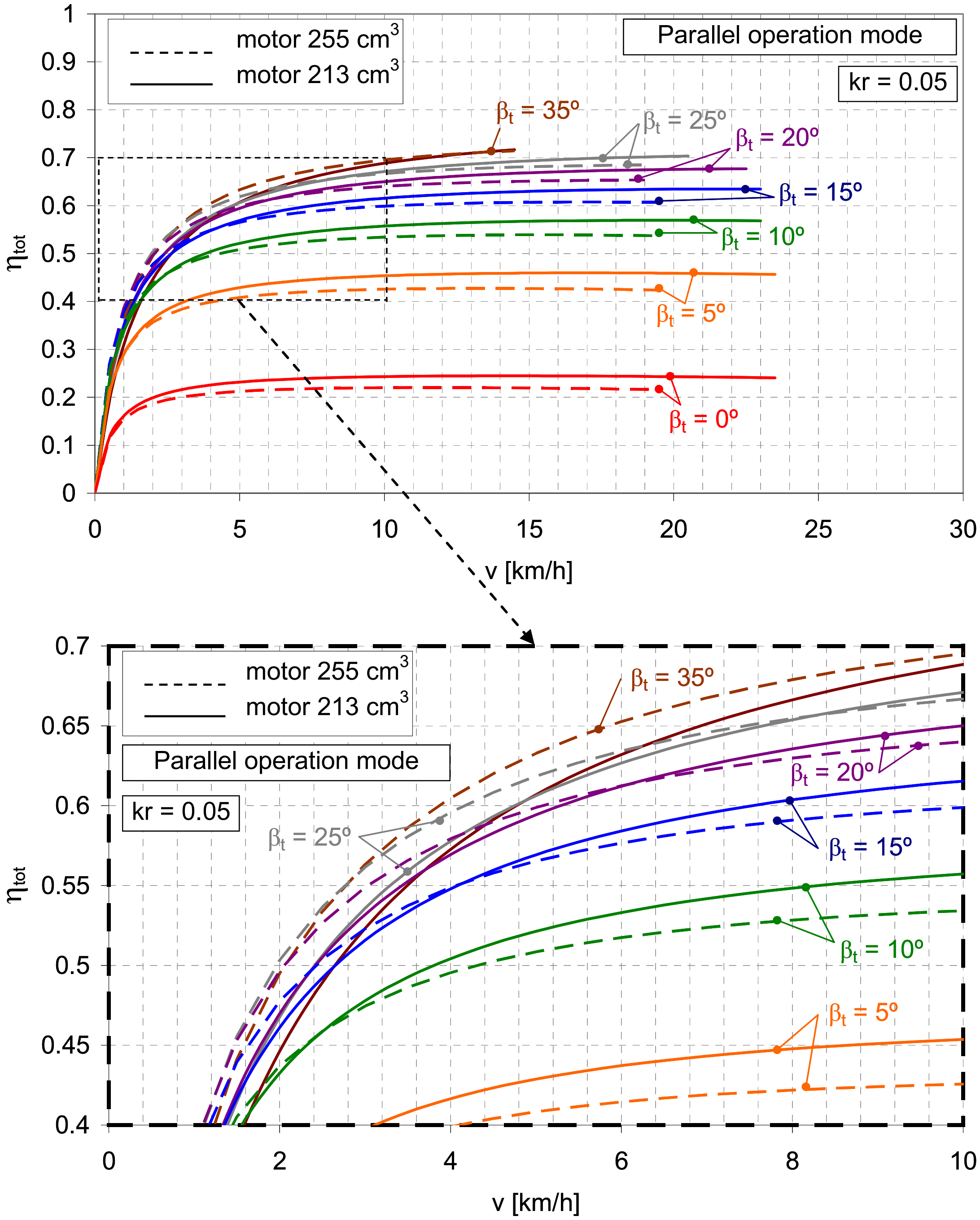


Figure 12. Efficiency as a function of speed and terrain slope in a parallel operation mode for the 213 and 255 cm^3 displacement hydraulic motors

		Operation mode	
		Series	Parallel
Hydraulic motors	q	=	=
	$P_{max.}$	↑	↓
Hydraulic pumps	q	↓	↑
	ΔP	↑	↓

Table 1. Flows and pressures comparison depending on the operation mode (series or parallel)

	S.O.M.	P.O.M.
Hydraulic pumps		
ΔP_{p1} [bar]	211.1	102.6
ΔP_{p2} [bar]	211.1	108.8
q_{outp1} [l/min]	26.1	51.5
q_{outp2} [l/min]	26.1	51.5
Hydraulic motors		
ΔP_{hF} [bar]	101.6	101.6
ΔP_{hR} [bar]	107.4	107.4

Table 2. Flow and pressure difference values at the studied point of Figure 8.

	Hydraulic Motor Displacement		
	213 cm^3	255 cm^3	
Pressure (*P*)	↑	↓	Shaft Torque
Flow (*q*)	↓	↑	Shaft Speed

Table 3. Comparative table of pressures and flows in hydraulic motors with different displacements