

Fuel consumption reduction in urban buses by using power split transmissions



Alarico Macor, Antonio Rossetti *

Department of Engineering and Management, Università di Padova, Stradella S. Nicola, 3-36100 Vicenza, Italy

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ABSTRACT

The reduction of fuel consumption of urban vehicles is one of the challenges of the society today. Many promising technologies such as hybrid electric vehicles and fuel cells are the focus of a lot of research and investments, but high costs prevent their development on a large scale, especially in heavy duty vehicles. Simpler solutions, such as continuously variable transmissions, are then interesting alternatives: they preserve some of the advantages of hybrid solutions significantly reducing the overall cost of the system. Since these technologies allow the decoupling of the engine's velocity from that of the wheels, optimised management of the engine becomes feasible, leading to lower fuel consumption than that of traditional solutions.

This work compares the hydromechanical transmission with a traditional power shift in an application where efficiency and comfort are required: a city bus.

The performance of a vehicle with a dual stage hydromechanical transmission has been investigated by means of a model implemented in the AMESim environment. A second model of a vehicle with a power shift transmission has been taken as reference solution. During the simulation of two standard test cycles, the first vehicle showed the lower fuel consumption, due to its better engine management, and to the operation of the torque converter during starts, which lowers the efficiency of the reference vehicle.

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1. Introduction

The hybrid vehicle technology is an effective answer to the request for the consumption and emissions reduction in vehicles [1,2]. In recent years, scientific research has proposed several electric hybrid powertrains [3–5], while the automotive industry has proposed various models of light duty vehicles with hybrid engines. The extension of this technology to heavy duty vehicles, although feasible, is nevertheless prevented by the increase of costs of such a solution.

An alternative and cheaper solution to the hybrid engine could be the use of a continuous transmission instead of a stepped one: it allows, as in some hybrid solutions, the optimal management of the engine, thanks to the separation between the engine speed and wheels' speed [6,7]. This aspect may be critical in vehicles whose engine often operates at low loads and variable speeds, as is the case for city buses.

It should be noted that the continuous transmission does not involve all the advantages of hybrid vehicles, such as the engine downsizing and the regenerative brake; however, it shows lower complexity and lower costs, fully developed technology with high

reliability [7]. Furthermore, it could be easily placed inside the vehicle, preventing radical vehicle redesign.

The feasible type of continuous transmission for buses is the hydromechanical one, known also as power split transmission (PSP). The capability of handling high powers with compact units makes the PSP the preferred solution between other CVT elements (chain belts, hydraulic converters, electric converters, etc.) for heavy duty vehicles [8] and is a nowadays common solution for high power agricultural tractors [9–11]. Nevertheless, its application to heavy duty trucks has been studied by many authors [12–14].

In the PSP, an epicyclic gear connects a variable speed path, made up of a hydrostatic transmission, with a fixed speed ratio path. For this reason, they are also called 3 shaft layouts. The variation in the hydrostatic group displacements produces a continuous speed variation at the wheels. In a particular functional condition, called lock-up point, the power only flows through a mechanical path and the transmission efficiency is close to the maximum [7]. In order to reach high efficiencies, the fraction of the engine power flowing through the hydraulic path must be small, but this fact limits the speed excursion of the transmission and requires further reduction gears before the wheels.

To increase the speed range and efficiency of the 3 shaft layouts, 4 shafts layouts with two epicyclic gears have been proposed [7,13–15]; they define two lockup points and therefore

* Corresponding author. Tel.: +39 049 827 7474; fax: +39 049 827 6785.

E-mail address: antonio.rossetti@unipd.it (A. Rossetti).

Nomenclature

A_f	frontal area of the vehicle (m^2)
B	brake
C	clutches
D	torque converter diameter (m)
F_a	aerodynamic resistance of the vehicle (N)
F_r	rolling resistance (N)
H_c	low heating value of the fuel (kJ/kg)
J	rotational inertia (kg m)
L_k	lock up clutch
M	vehicle mass (kg)
P	power (kW)
ΔP	pressure drop of the hydraulic machines (bar)
T	torque (Nm)
TC	torque converter
V	hydraulic unit displacement (m^3) – (cc)
c_d	drag coefficient (/)
d	hydraulic orifice diameter (m)
e	error between actual and mission velocity (m/s) – (km/h)
k_i	integral gain of the controller (/)
k_p	proportional gain of the controller (/)
k_r	rolling resistance coefficient (/)
$\dot{m}$	mass flow rate (kg/s)
n	rotational speed (rad/s) – (rpm)
r	control signal; reset of the integral controller
s	control signal, accelerator if positive, brake if negative
t	time (s)
t_{lim}	oscillation period of the system when the stability limit of the proportional gain is exceeded (s)
v	vehicle velocity (m/s) – (km/h)

Greek symbols

α	actual to maximum displacement ratio (/)
η	efficiency (/)
λ	torque converter power factor (/)
ν	torque converter speed ratio (/)
ρ	density of the torque converter fluid (kg/m^3)
τ	transmission ratio (/)
ω	angular velocity (rad/s)

Subscripts

CH_4	methane
gear	referring to the gear
I, II	referring to the first and second hydraulic units
ICE	referring to the internal combustion engine
Tr	referring to the transmission
Tot	total referring to the engine-transmission group
i, p	referring to the turbine/pump of the hydraulic converter
miss	referring to the mission
actual	referring to actual condition
out	referring to the rear axis
ref	reference point for the PSp transmission control
lk	referring to leakage flow (hydraulic CVT)
flow	referring to the main flow (hydraulic CVT)
ff	feed forward
PI	proportional and integral part of the closed loop control
Wheels	wheels

Superscripts

tc	referring to the torque converter
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two functional areas with high efficiency. Further significant improvements in the transmission efficiency were also achieved substituting the classical design procedure with optimised design [16,17].

In the buses, the PSp is the competitor of the traditional power shift transmission (PSh): both of them are automatic transmissions that guarantee the continuous delivery of power to the wheels over the whole velocity range of the vehicle. Furthermore, both layouts could be classified as intermediate–high complexity/cost systems: this fact places the two solutions in the same market sector. Despite the intrinsic inferiority in terms of transmission efficiency, the PSp solution presents interesting advantages when the focus is moved from the transmission itself to the entire vehicle. In fact, the driveline can be seen as two machines in series, the engine and the transmission, so its efficiency is expressed by: $\eta_{Tot} = \eta_{ICE} \cdot \eta_{Tr}$. Now, considering the differential of this efficiency $\delta\eta_{Tot} = \eta_{ICE} \cdot \delta\eta_{Tr} + \eta_{Tr} \cdot \delta\eta_{ICE}$ it is evident that an improvement in the engine efficiency can more than offset the negative effect of a decrease in transmission efficiency.

The aim of this paper is the evaluation of a 4 shaft layout in an application different from the usual agricultural machines, i.e. a city bus, in order to verify possible advantages in terms of energy consumption. The evaluation will be made by comparing the performance of a city bus equipped with a 4 shaft PSp with the performance of the same bus equipped with a traditional PSh transmission. The dynamic model of both transmissions was implemented in the 1-D commercial software LMS Imagin.Lab AMESim [18]. The efficiency and energetic performance of the transmissions were compared on different routes, including constant power acceleration and two standard emission cycles.

2. Test case

The 12 m IVECO Cityclass bus was taken as the reference for the present work; its dimensions and main characteristics are shown in Fig. 1 and Table 1.

Although the reference for the calculations is a precise model, the test cases can be considered representative of the entire market segment. In fact, models belonging to the same class all have similar characteristics.

The internal combustion engine of the bus is an IVECO Cursor 8 spark ignition EURO V engine, fuelled with natural gas. Its main characteristics are reported in Table 2.

3. Transmissions' layout

The PSh and PSp transmission layouts compared in the present work are presented in Figs. 2 and 3 respectively. Dashed lines were used to highlight functional groups, while dash-dot lines were used to sketch the main sensor and command signals.

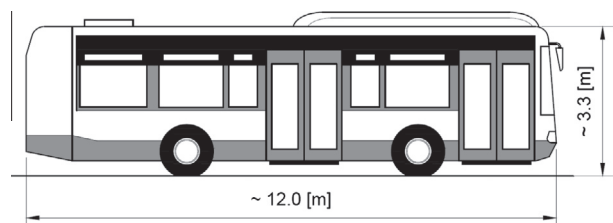


Fig. 1. Test case.

Table 1

Main characteristics of the reference vehicle.

Vehicle model	IVECO-IRISBUS: Cityclass
Driving layout	Rear-Wheel Drive
Vehicle mass	12 000 kg
Maximum speed	Limited to 68 km/h
Engine	IVECO Cursor 8, 6 – cylinder.

Table 2

IVECO Cursor 8 characteristics.

Type	6 cylinder in-line turbocharged
Bore × stroke	115 × 125 mm
Compression ratio	11.0:1
Rated power	200 kW @ 2100 rpm
Maximum torque	1100 Nm @ 1250–1650 rpm
Best efficiency point	1080 Nm @ 1100 rpm

The PSh transmission can be considered a typical solution for bus applications. The layout consists of two planetary gears and one Ravigneaux gear, capable of five gear ratios, which are selected by means of four brakes (B_1 , B_2 , B_3 and B_4) and three clutches (C_1 , C_2 , a , and C_3). The truth table of the five allowed gear ratios is reported in Table 3.

A torque converter (TC) is placed between the PSh unit and the I.C.E. shaft, which improves the comfort and drivability when the vehicle starts to move. The lock up clutch (L_k) allows the torque converter to be bypassed when the impeller and the pump are nearly synchronous, eliminating slippage and improving efficiency.

The second transmission is a PSp transmission of the input coupled dual stage type. Its layout derives from the basic input coupled layout, substituting the epicyclic gear with an advanced one, with two outlets. In this way the transmission has two lockup points and, therefore, it takes advantage of two high efficiency zones; moreover the hydraulic units work in a more efficient way since they operate at low speeds and at high displacements.

Both transmissions were dimensioned according to the considered test case. In order to obtain a significant comparison, none of the transmissions were optimised. The PSh planetary train cinematic was sized to match the gear change velocities with the data available from the test case. The torque converter characteristic was modelled on the basis of data available in open literature. The PSp was sized according to classical relations, similar to the one presented in [13]; the hydraulic units' sizes were then rounded to commercial standards. In Table 4 the main characteristics of the tested transmissions are reported.

4. Transmission modelling and simulation software

The mechanic and hydraulic elements of the vehicle, as well as the whole control system, were modelled and simulated using LMS Imagin.Lab AMESim commercial software [18].

In this section, the models of the main elements of the two transmissions will be synthetically described; in the last part of this section the control strategies of the two vehicles will be discussed. The complete layouts of the numerical models used for the analyses are reported in the Appendix A.

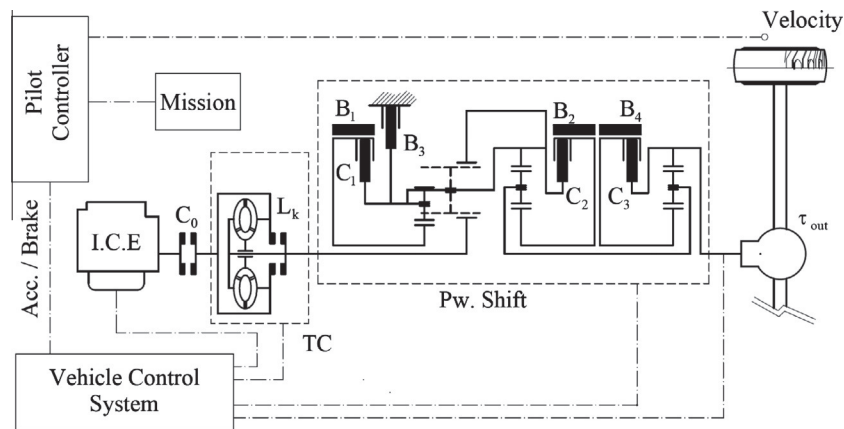
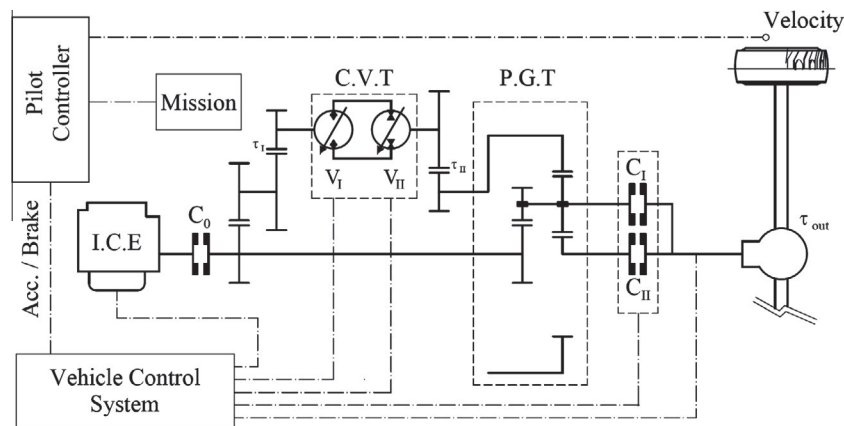
**Fig. 2.** Power shift transmission layout.**Fig. 3.** Power split transmission layout.

Table 3

Truth table of the brakes and clutches of the power shift transmission.

Gear	I	II	III	IV	V	R.G.
C ₁	×	×	✓	✓	✓	×
C ₂	✓	×	×	✓	✓	✓
C ₃	✓	✓	✓	✓	×	✓
B ₁	×	✓	×	×	×	×
B ₂	✓	✓	✓	×	×	×
B ₃	×	×	×	×	×	✓
B ₄	×	×	×	×	✓	×

Table 4

Main data of the tested transmissions.

Power shift transmission		Power split transmission	
<i>Torque converter</i>		<i>Unit I</i>	
Pump diameter	0.34 m	Gear ratio τ_I	0.58
Oil density	800 kg/m ³	Gear efficiency	0.98
Efficiency	Fig. 6	Maximum displacement	125 cc
		<i>Unit II</i>	
		Gear ratio τ_{II}	0.58
		Gear efficiency	0.98
		Maximum displacement	125 cc
		Hydraulic units efficiency	Fig. 9
		<i>Complex planetary gear</i>	
		Ring tooth number	90
		First sun tooth number	45
		Second sun tooth number	30
<i>Rear Axle Gears</i>		<i>Rear Axle Gears</i>	
Overall ratio	5.90	Overall ratio	10.13
Differential gear ratio	4.00	Differential gear ratio	4.00
Ordinary gear pair	1.475	Planetary reducer	2.53
Axle overall efficiency	0.955	Axle overall efficiency	0.960

The engine was modelled using an ideal torque generator, whose driving torque was a function of the rotational speed and of the accelerator position. The fuel consumption was calculated on the basis of the actual load and rotational speed. The speed–torque characteristic of the engine and the fuel consumption data were derived from experimental measures reported in [19]. The normalised engine map and the experimental points are reported in Fig. 4. In order to take into account the engine dynamics, the engine inertia and a first order delay on the accelerator signal have been added to the torque generator model, as shown in Fig. 5.

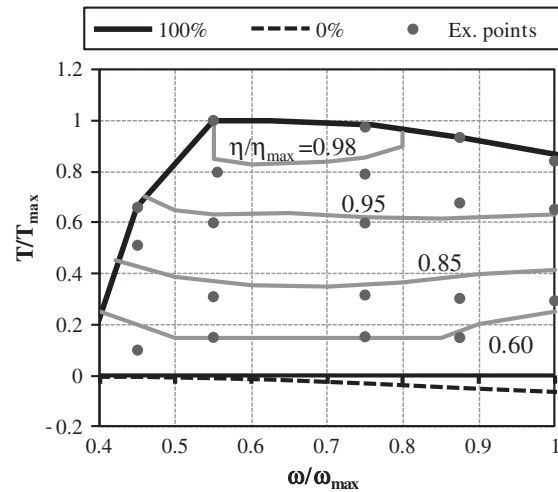
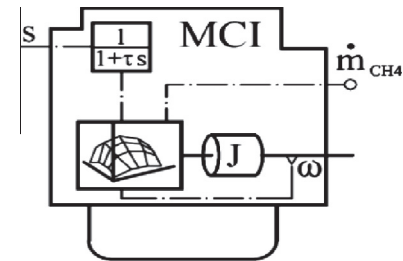
Each gear pair considered into the two models was assumed to have a constant efficiency η_{gear} , equal to 0.980. The assessment of the gear pair needed for a specific transmission ratio was obtained assuming the maximum ratio for the single pair $\tau_{\text{max}} = 2$.

The planetary gear train losses were modelled through the efficiency of each elementary gear pair. The ring–planetary couple and the planetary–sun couple efficiencies were then assumed to be constant and equal to 0.980 each. The same concept was also applied to the Ravigneaux element of the PSh transmission.

The rear axle gear was assumed to be a composition of the differential gear and of one planetary or ordinary gear reducer to obtain the design reduction ratio. The efficiency of the rear gear was computed accordingly.

In order to simplify the model, the friction losses of clutches and brakes were neglected. This assumption benefits the PSh transmission, in which many clutches and brakes are involved in each gear ratio shift. Nevertheless, the losses caused by these elements were considered to be small when compared to the engine and gear losses.

The vehicle loads, i.e. the aerodynamic friction and the rolling friction, were modelled according to $F_d = 0.5c_d v^2 A_f$ and $F_r = k_r M g$; where for the drag coefficient, frontal area and rolling resistance

**Fig. 4.** Normalised efficiency map of the engine.**Fig. 5.** Dynamic model of the engine.

coefficient the following values were respectively assumed: $c_d = 1.18$, $A_f = 7 \text{ m}^2$, $k_r = 0.008$.

The vehicle mass M was considered as a translating point of mass and no complex dynamic was applied during the simulations. No slope resistance was considered because the test cycles are intended to be performed on a flat road.

The torque converter behaviour was described using classical non dimensional parameters: the power factor λ and the torque ratio: $T_p^{\text{tc}} = \lambda \rho \omega_p^2 D^5$; $\mu = T_i^{\text{tc}} / T_p^{\text{tc}}$; where the subscripts i and p refer to the impeller and the pump side of the converter respectively. Both of them are functions of the speed ratio $v = \omega_i^{\text{tc}} / \omega_p^{\text{tc}}$, as shown in Fig. 6a and b. The torque converter efficiency, which can be obtained as the product of speed ratio and torque ratio, is also shown in Fig. 6.

The lock-up clutch control system was set to insert the clutch at $v = 0.85$.

A classical closed circuit layout was employed for the hydraulic path of the PSp (Fig. 7); its set up parameters are reported in Table 5.

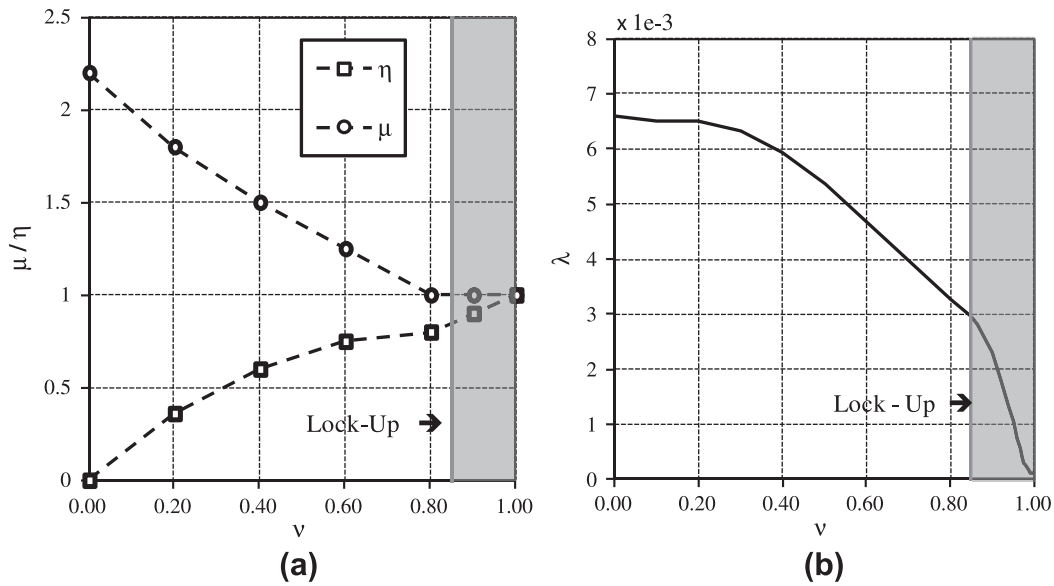


Fig. 6. Torque converter characteristic curves as a function of the speed ratio; (a) torque ratio and efficiency; (b) power coefficient.

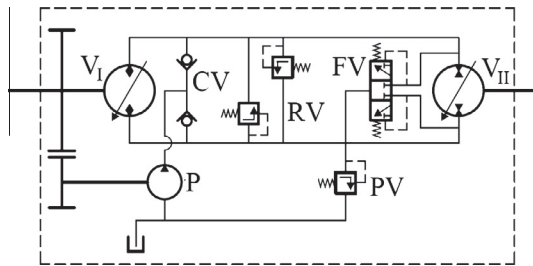


Fig. 7. Hydraulic closed circuit.

Table 5
Hydraulic component data.

Displacement of unit I	V_I	125 cc
Displacement of unit II	V_{II}	125 cc
PV cracking pressure		15 bar
RV cracking pressure		450 bar
Displacement of unit P		20 cc

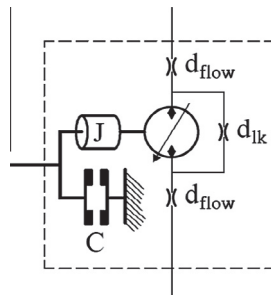


Fig. 8. Real pump functional scheme.

The behaviour of the hydraulic units was modelled by applying to an ideal unit the main loss sources (Fig. 8): friction by means of a brake (C), and leakage by means of the orifices d .

In order to consider the influence of the operating conditions on the unit losses, both the friction torque of the brake C and the diameter of the orifices (d_{flow} and d_{lk}) were expressed as

polynomial functions of the rotational speed, the load and the actual to maximum displacement ratio. The coefficients of the loss models were adjusted on the basis of experimental data supplied by a manufacturer. The overall performance of the hydraulic CVT transmission is reported in Fig. 9 for different operating conditions.

5. Control elements

5.1. Pilot controller

The pilot was modelled as a controller receiving as input signals the vehicle velocity and the desired velocity defined by the route. The output s was then bound to the range $[-1 \dots +1]$. The positive values were passed to the control system of the I.C.E. and correspond to the accelerator position, while the negative values were passed to the braking system.

The output signal s was computed by summing the signal obtained from a feed forward model (s_{ff}), based on the velocity prescribed by the bus route, and a closed-loop proportional and integral controller (PI) s_{pi} , which compares the actual velocity with the desired velocity.

The feed forward model has the primary purpose of predicting inertial loads arising as the vehicle changes velocity. The output is then computed as the ratio between the power associated with inertial forces and the maximum power: $s_{ff} = Mav/P_{max}$. This contribution to the overall output was essential to improve the response of the system to sudden acceleration after stops.

The PI controller used is reported in Fig. 10, where $e = v_{miss} - v_{actual}$. The proportional constant k_p was initially set to 0.2, assuming that an error of 5 km/h will lead to the full accelerator signal in order to correct the error. This value was then refined testing the vehicle on a series of trapezoidal velocity profiles with increasing slope up to the saturation of the engine power. The value of 0.35 was then set as optimum values in order to obtain a prompt response to velocity changes avoiding in the same time excessively sharp accelerator changes. To avoid windup during strong accelerations, the integral part was bounded between -0.5 and 0.5 using a saturation element and a negative feedback of proportional gain k equal to 100. The integral part was initially set according to the Ziegler–Nichols method: $k_i \cong k_p/t_{lim}$. Considering to the significant mass of the system, the oscillation period t_{lim}

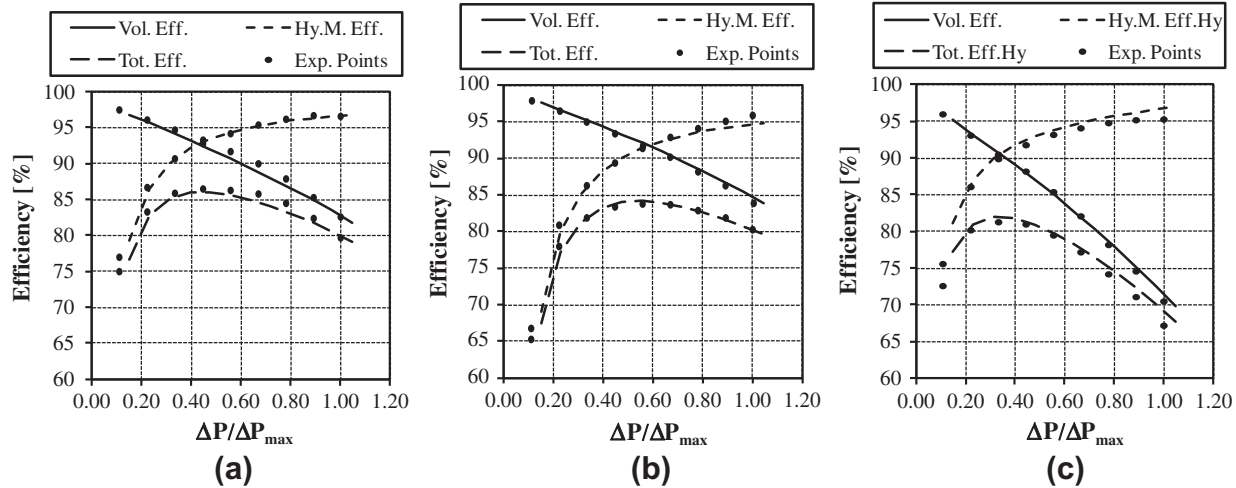


Fig. 9. Efficiency of the hydraulic CVT for different operating conditions; comparison between model (lines) and experimental data (dots); (a) $n_1 = 2000$ rpm, $\alpha_1 = 1$, $\alpha_{II} = 1$; (b) $n_1 = 3000$ rpm, $\alpha_1 = 1$, $\alpha_{II} = 1$; (c) $n_1 = 2000$ rpm, $\alpha_1 = 0.5$, $\alpha_{II} = 1$.

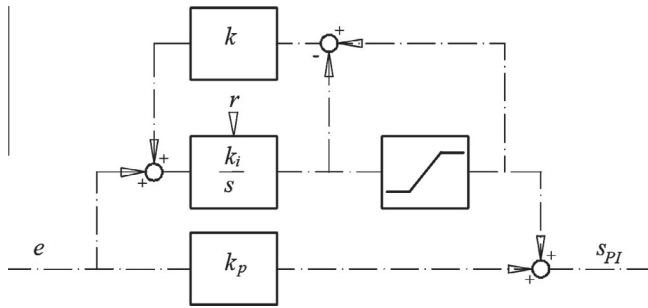


Fig. 10. Pilot PI controller.

was assumed in the order of 10 [s]. After iterative tuning, the final value of 0.03 was set as definitive value. In order to avoid control instability due to strong decelerations, the integral element was provided with a reset port controlled by the signal r . This reset signal was activated when $v_{miss} = v_{actual} = 0$.

5.2. Transmission – I.C.E. control strategy

This control system has two main objectives: the first is to relate the pilot output signal s to the engine; while the second is to control the transmission, then changing the gear ratio when proper conditions are met.

As the PSp transmission allows a continuous change of the gear ratio, while the PSh is based on step changes, different control strategies were developed for the two transmissions. The control strategy plays a key role in the transmission performance; nevertheless, in this study no optimisation was accomplished on this element, assuming the simplest control strategy for both transmissions.

5.2.1. Power shift transmission

The control of the PSh transmission focuses on the set points of the gear change. In order to obtain the best efficiency the actual gear shift should be chosen between the available gears at a certain velocity to be the one with the lowest fuel consumption. However this approach encourages low engine speeds, as the engine's most efficient point is reached at a relative low velocity (1100 rpm), reducing the maximum power which could be delivered by a fixed gear. Furthermore, it reduces the engine promptness and drivability in urban environments. On the contrary, in order to obtain the highest power on the wheels, the gear change should be delayed as

the engine reaches the speed of maximum power, in this case 2100 rpm. The set points were then defined as a compromise between the two approaches, assuming the engine speed before gear change to be around 1880 rpm. Table 6 reports the velocities at which the gear change is carried out.

The accelerator signal s was interpreted as a request for power. As the I.C.E. velocity is defined by the vehicle velocity and the transmission ratio, the only available degree of freedom is to operate on the engine load. The maximum fuel admission (corresponding to 100% load line of Fig. 4) was associated with $s = 1$ while $s = 0$ was associated with the engine-brake curve (0% load line of Fig. 4).

5.2.2. Power split transmission

The presence of the CVT element allows the control of the engine velocity apart from the actual vehicle velocity. The operating point of the I.C.E. could be then selected on the engine map along the iso-power line corresponding to the actual power demand. The single track control [20], for example, forces the I.C.E. operating point to move on the line connecting the highest efficiency points, as a function of the delivered power. Nevertheless, this control system requires the real time measure of the engine power, i.e. of the torque on the engine shaft, which is not generally available on a vehicle.

A simpler control system was therefore implemented in the present model, which does not require the torque measurement. The engine speed and load were expressed as functions of the accelerator position s as follows:

$$\begin{cases} n_{ICE} = n_{ref}; & \text{load} = 0 \dots 100\%; & \text{when } s = 0 \dots 0.6 \\ n_{ICE} = n_{ref} \dots n_{max}[\text{rpm}]; & \text{load} = 100\%; & \text{when } s = 0.6 \dots 1 \end{cases}$$

As the accelerator signal is within the range $[0 \dots 0.6]$, the power increase is obtained by increasing the engine load at constant velocity. The value of 1100 rpm was chosen as reference velocity n_{ref} as it is the velocity at which the engine highest efficiency is attained. When $s = 0.6$ the maximum load line is reached, therefore

Table 6
Velocity of gear change for the Power shift transmission.

Gear change	Velocity (km/h)	I.C.E. rotational speed (rpm)
I st to gear II nd	15.8	1894
II nd to III rd gear	27.0	1896
III rd to IV th gear	38.0	1886
IV th to V th gear	54.0	1887

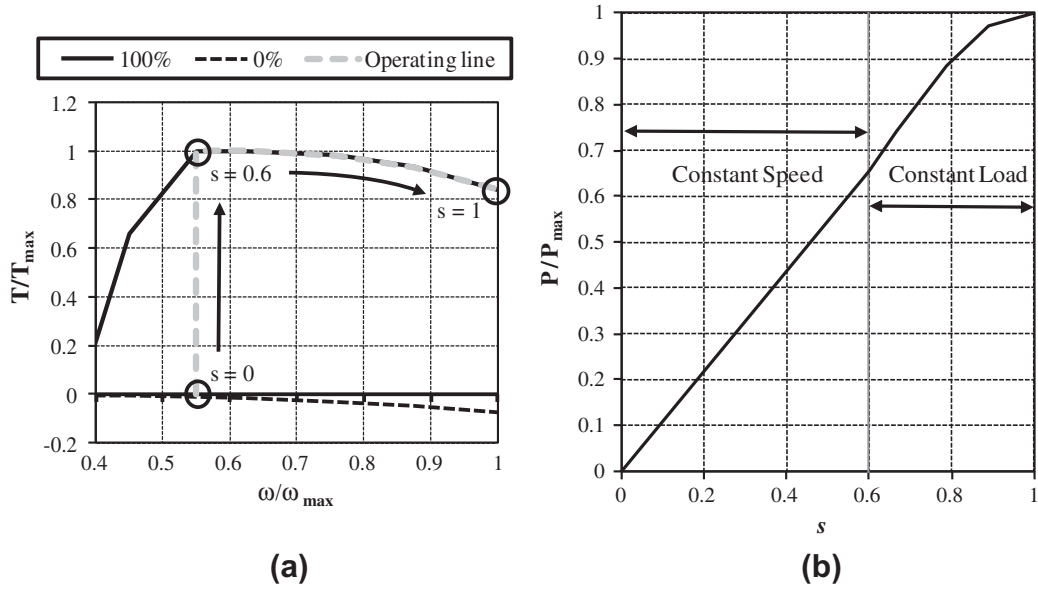


Fig. 11. Engine-transmission control strategy for the power shift transmission; (a) engine operating points as a function of the accelerator position s ; (b) the engine power as a function of the accelerator position.

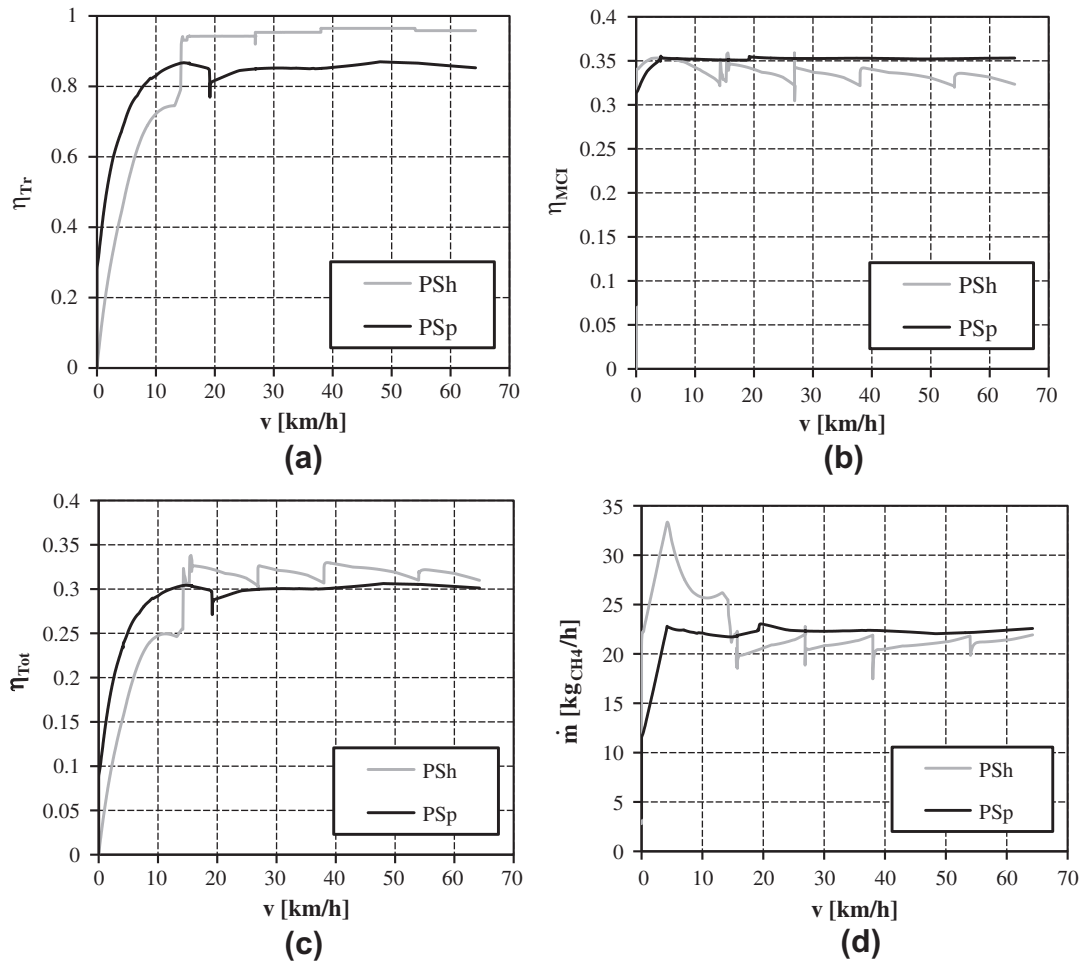


Fig. 12. Efficiency comparison for the constant power test; (a) transmission efficiency; (b) I.C.E. efficiency; (c) overall efficiency; (d) fuel consumption.

Table 7
Test I results.

	Power shift	Power split	
η_{Tr}	0.878	0.822	(–)
η_{ICE}	0.336	0.351	(–)
η_{Tot}	0.295	0.289	(–)
m_{CH_4}	1.09	1.10	(kg)

Table 8
Test cycle main characteristics.

	Manhattan Bus	New York Bus	
Duration	1089	600	(s)
Stops time/duration	36	66	(%)
Total driving distance	3.33	0.99	(km)
Maximum speed	40.9	49.6	(km/h)
Average speed	11.0	5.94	(km/h)
Average speed without stops	17.17	17.22	(km/h)
Maximum (positive) acceleration	2.24	2.83	(m s ^{−2})
Average (positive) acceleration	0.59	1.18	(m s ^{−2})
Number of stops per km	6	11	(–)
Peak power for the tested vehicle	265	360	(kW)
Total time with required $P > 200$ (kW)	9.8	9.4	(s)
Average time with required $P > 200$ (kW)	0.36	1.3	(s)

the power cannot be further increased at constant velocity. For s between 0.6 and 1 the I.C.E. is then accelerated up to maximum velocity, maintaining the full load condition. The switch value of $s = 0.6$ was defined to obtain a continuous trend of the power as function of the accelerator position. This is shown in Fig. 11b, while the path of the prescribed operating points at equilibrium is reported in Fig. 11a. A controller acting on the CVT units was used to maintain the I.C.E. velocity at the prescribed value. The structure of this controller is similar to that one of the pilot controller, implementing a feed forward model and a PI closed loop controller.

6. Results

The presented transmissions were compared over two different tests. First, the engine and the transmission are considered separately from the vehicle and tested at a constant power equal to the 50% of the I.C.E. maximum power. In the second test, the transmissions were tested considering the aerodynamic and dynamic

load on standard emission cycles for urban buses. Two different bus cycles were used: the Manhattan Bus Cycle and the New York Bus Cycle [21].

6.1. Constant power test

Since the efficiency of the hydraulic units depends strongly on the transmitted power, many comparisons and optimisations of the literature were conducted at constant power [10,12–15]. The two transmissions were then compared on a slow acceleration ramp from rest to the maximum velocity of 68 km/h in 180 s. A variable breaking torque was applied to the wheel axle to obtain a constant power at the axle, equal to the half the maximum engine power. Moreover, the breaking torque was bound at a low velocity, in order to not exceed the maximum design pulling force of 40 kN.

In order to highlight the role of the different elements of the transmission, the mechanical efficiency of the transmission, the engine efficiency and the total efficiency were considered; they are defined by the following ratios: $\eta_{Tr} = P_{Wheels}/P_{ICE}$; $\eta_{ICE} = P_{ICE}/\dot{m}_{CH_4}H_c$; $\eta_{Tot} = \eta_{ICE}\eta_{Tr}$ where the P 's are powers and H_c is the low heating value of the fuel.

These efficiencies and the fuel consumption are reported for the constant power test case in Fig. 12.

The PSh solution, as expected, shows the highest transmission efficiency when the torque converter is locked up, reaching values greater than 90% (Fig. 12a). The torque converter significantly decreases the transmission performance for low velocity: the efficiency of the converter increases slowly as the vehicle accelerates, reaching 73% just before lock-up.

The efficiency of the PSp transmission reaches a maximum of 86% twice, one for each gear, in the proximity of the full mechanical point. In this case the efficiency of the transmission increases rapidly with vehicle velocity, exceeding 80% for velocities greater than 8 km/h. The transmission efficiency shows a discontinuity at 20 km/h, which is the velocity of the synchronous shift between the first and second gears.

Fig. 12b presents a comparison between the engines' efficiency. The PSh curve is characterised by the discontinuities due to the gear change: the engine presents the best efficiency at lower engine velocity, then when the gear is engaged; as the engine accelerates the efficiency decreases. When the following gear is engaged, the engine slows and the efficiency suddenly increases. On the other hand, the engine of the PSp, under constant power request, is independent of the external velocity of the vehicle, guaranteeing a relatively high efficiency.

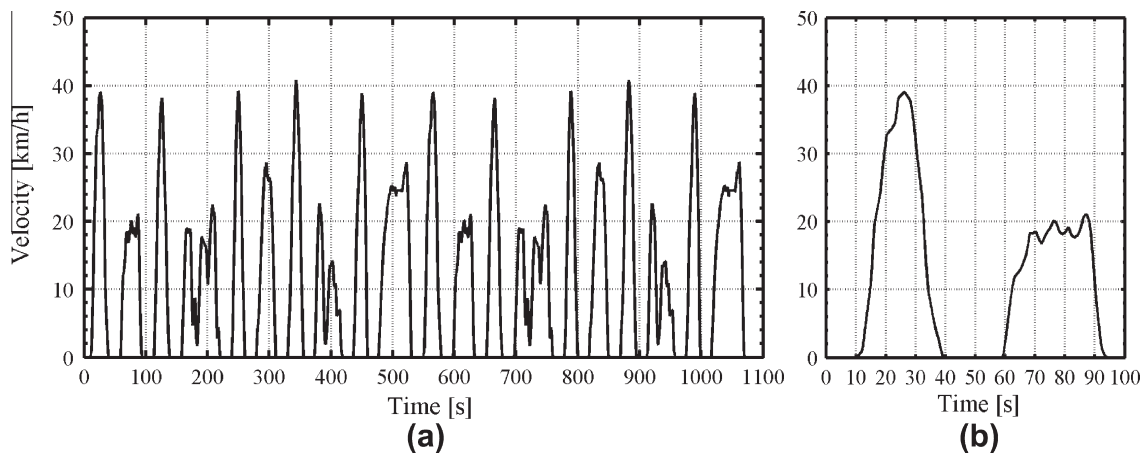


Fig. 13. Manhattan Bus Cycle; (a) Full cycle; (b) highlight of the first 100 s.

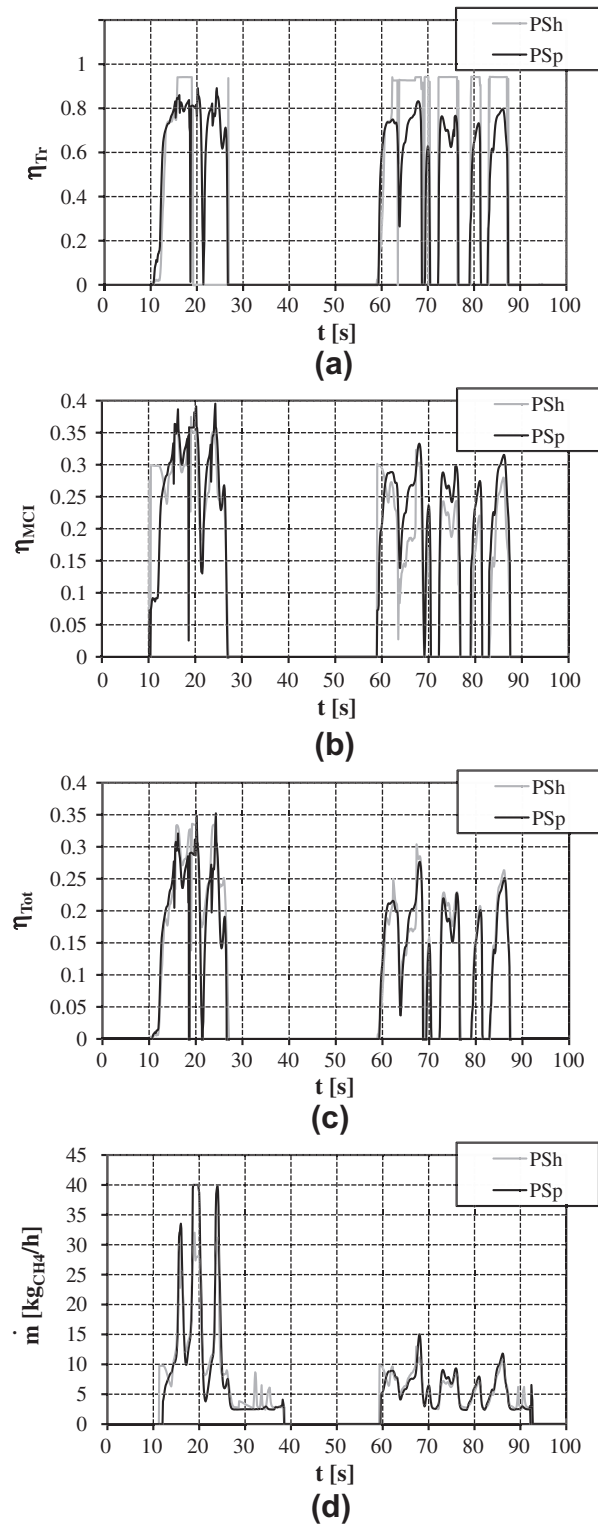


Fig. 14. Efficiency comparison during the first 100 s of the Manhattan cycle; (a) transmission efficiency; (b) I.C.E. efficiency; (c) overall efficiency; (d) fuel consumption.

The overall efficiency then results from the two curves previously presented (Fig. 12c): the PSh curve is penalised by the torque converter at low velocity, and for each gear it is evident that the decrease is associated with the I.C.E. efficiency. The same consideration can be applied to the fuel consumption graph in Fig. 12d: as the requested power is the same for the two transmissions.

Table 9

Test II results.

	Manhattan Bus		New York Bus		
	Power shift	Power split	Power shift	Power split	
$\bar{\eta}_{Tr}^b$	0.806	0.664	0.863	0.732	(–)
$\bar{\eta}_{ICE}^b$	0.236	0.260	0.277	0.312	(–)
$\bar{\eta}_{Tot}^b$	0.177	0.185	0.227	0.236	(–)
$\bar{e}^a$	0.279	0.153	0.333	0.380	(km/h)
$m_{CH_4}^a$	1.54	1.52	0.504	0.496	(kg)

^a Average limited to in-movement time.

^b Average limited to in-movement time and positive delivered power.

The averaged value for the three efficiencies and the overall fuel mass consumed during the test are reported in Table 7.

The higher transmission efficiency of the PSh and the higher efficiency of the engine in the PSp lead to very similar results in terms of total efficiency and fuel consumption. It is important to note that the small engine efficiency increase of the PSp engine (+1.6%) can balance the 5.7% gain of the PSh transmission efficiency, thanks to the relative incidence of these efficiencies on the total efficiency.

6.2. Bus route tests

The constant power test results encouraged a more realistic comparison between the two transmission layouts. The constant power test cannot be considered representative of the standard operating conditions of the engine and the transmission of a city bus. For this reason, the two transmissions were compared on the basis of two missions closer to real-life driving conditions. In order to avoid an arbitrary choice, the two test cycles were taken from the standard emission cycles for bus vehicles. The New York Bus and Manhattan Bus cycles were considered representative for this test [21], as both are derived from dynamometric measurements on real city environment. Both of them are characterised by frequent stops, fast accelerations and low peak speeds. The main characteristic of the two cycles are reported in Table 8. The Manhattan Bus cycle is shown in Fig. 13a, while Fig. 13b reports the highlights of the first 100 s of this cycle.

In order to evaluate the feasibility of these two cycles, an approximate analysis of the required power level was carried out in advance, taking into account only the inertia forces: $P = a_{miss} M v$.

Some results of this computation are reported in the lower part of Table 8.

Both cycles require power peaks higher than the ICE power of 200 kW. Nevertheless the Manhattan cycle appears not significantly above the maximum power, and the exceeding power occurs only for short time peaks (average of 0.36 [s]), for an overall time of less than ten seconds over a mission of more than 1000 s.

The New York cycle appears to be significantly challenging, as the power peaks are higher and last for longer time compared with the mission duration. Nevertheless, this was not considered a major problem because of the comparative nature of the present work.

In fact, when the required power exceeds the maximum power of the engine, the vehicle will no longer be able to precisely follow the mission; but this will happen in both transmissions and to the same extent, because the vehicles are equipped with the same engine, the same pilot and the same set up. Therefore, the comparison appears significant and its results will be able to point out the effects of the transmission in such challenging conditions.

For these reason, the mission have not been modified from its original format and not even the engine power or the vehicle mass, as considered representative of the European city vehicle class.

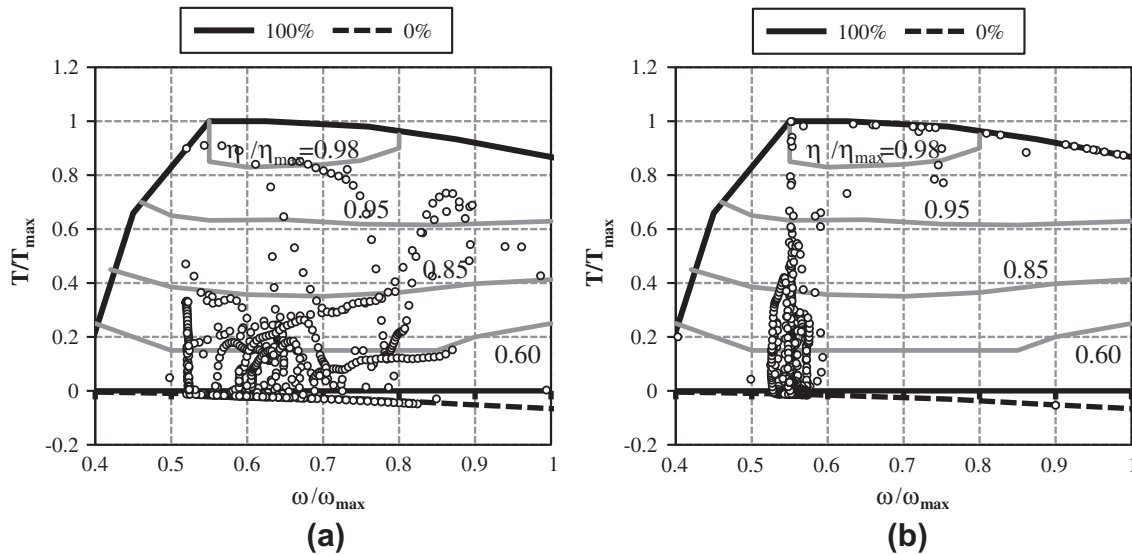


Fig. 15. Operation points (sampled with a frequency of 10 Hz) on the normalised efficiency map for the first 100 s of the Manhattan Bus Cycle; (a) PSh; (b) PSp.

As the two routes are too long to permit the complete discussion of the results over the whole cycle, detailed comparison between the two transmissions will be performed only considering the first 100 s of the Manhattan Bus test (Fig. 13b). Quantitative results, which will be presented later on this section, will instead refer to the entire route. Efficiency and fuel consumption comparison between the two layouts were reported in Fig. 14 for the first two accelerations of the Manhattan Bus test cycle.

The mechanical efficiency of the PSh, as in the constant power test, is significantly higher than the efficiency of the PSp for a speed higher than about 15 km/h. At the same time, the torque converter significantly lowers the performance for low velocities. The engine performance trends are similar, with the PSp transmission having higher efficiencies with respect to the PSh, especially during the second part of the considered mission interval (60–90 [s]). In this latter layout is also possible to note the negative effects of the sudden insertion of the torque converter lock-up, which slows the engine temporarily reducing the efficiency.

The total efficiency and the fuel consumption graphs provide evidence of the PSp transmission's superiority at low velocities, at the start of the acceleration and during the last part of the deceleration, while the PSh obtains the best results for velocities higher than 16 km/h, and when the torque converter is locked-up.

The results of the analyses on the two cycles are summarised in Table 9. The averaging was performed only considering in-movement time. As both transmissions were equipped with a cut-off clutch, assuming the engine is idle when the vehicle stops, the vehicle has the same energetic behaviour when stops occur. Furthermore, the efficiency averages were again limited to exclude the breaking time, in which the power flow in the drive line is reversed, leading to non-physical negative values.

For both cycles, the same conclusions of the constant power test can be drawn, i.e. in the PSh solution the transmission has a higher efficiency, although, at the same time, its engine is less efficient because of the low velocity and the continuous stops leading to the frequent use of the torque converter. This fact results in a slightly higher total efficiency for the PSp solution in both test cycles, proving that it can be considered a competitor of the most widely used PSh transmission.

Table 9 also reports the average deviation between the test speed and vehicle speed $e = v_{\text{miss}} - v_{\text{actual}}$. The presence deviation between the target mission and the actual mission is mainly related to the saturation of the available power which prevents the pilot from following exactly the mission. As expected, the New York Bus Cycle, which has the highest power requests (see Table 8), presents higher errors. Despite this, the average velocity errors $\bar{e}$ are small enough to not influence the average performance of the vehicle.

The difference between the average efficiency of the I.C.E. could be appreciated comparing the operation points of the engine during the test cases. In Fig. 15 the operating points of the first 100 s of the Manhattan Bus Cycle are reported on the normalised engine map.

The active control logic allows the PSp to operate with a lower engine velocity and a higher efficiency than the PSh.

Despite no direct modelling of pollution being made in this work, the results of Fig. 15 and Table 9 allow some considerations to be drawn in this regard. In fact, the reduction in fuel consumption, the improvement in the engine average efficiency and the capability to control, to some degree, the operating line in the engine map, are all elements leading to the better behaviour of the PSp layout in terms of emissions. This feature could represent a significant benefit in urban environments.

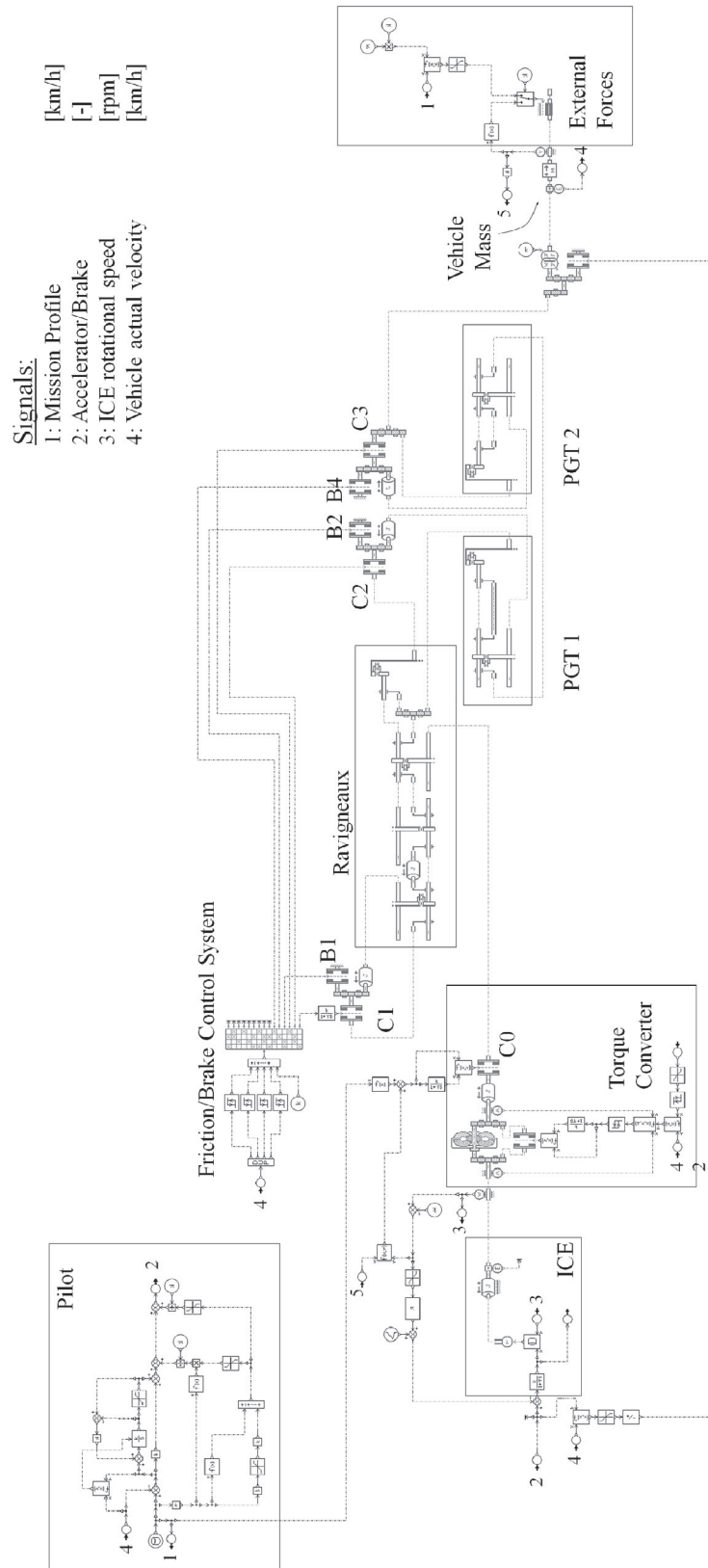
7. Conclusions

The PSp transmission has recently gained an important place in the agricultural machinery sector, where driving comfort is an important requirement. It is well known that the efficiency of the PSp transmission is lower than the conventional transmission, but being a continuous transmission it provides better comfort and allows optimal management of the engine. The engine operating point is in fact uncoupled from the vehicle velocity and it can be controlled to improve the engine efficiency.

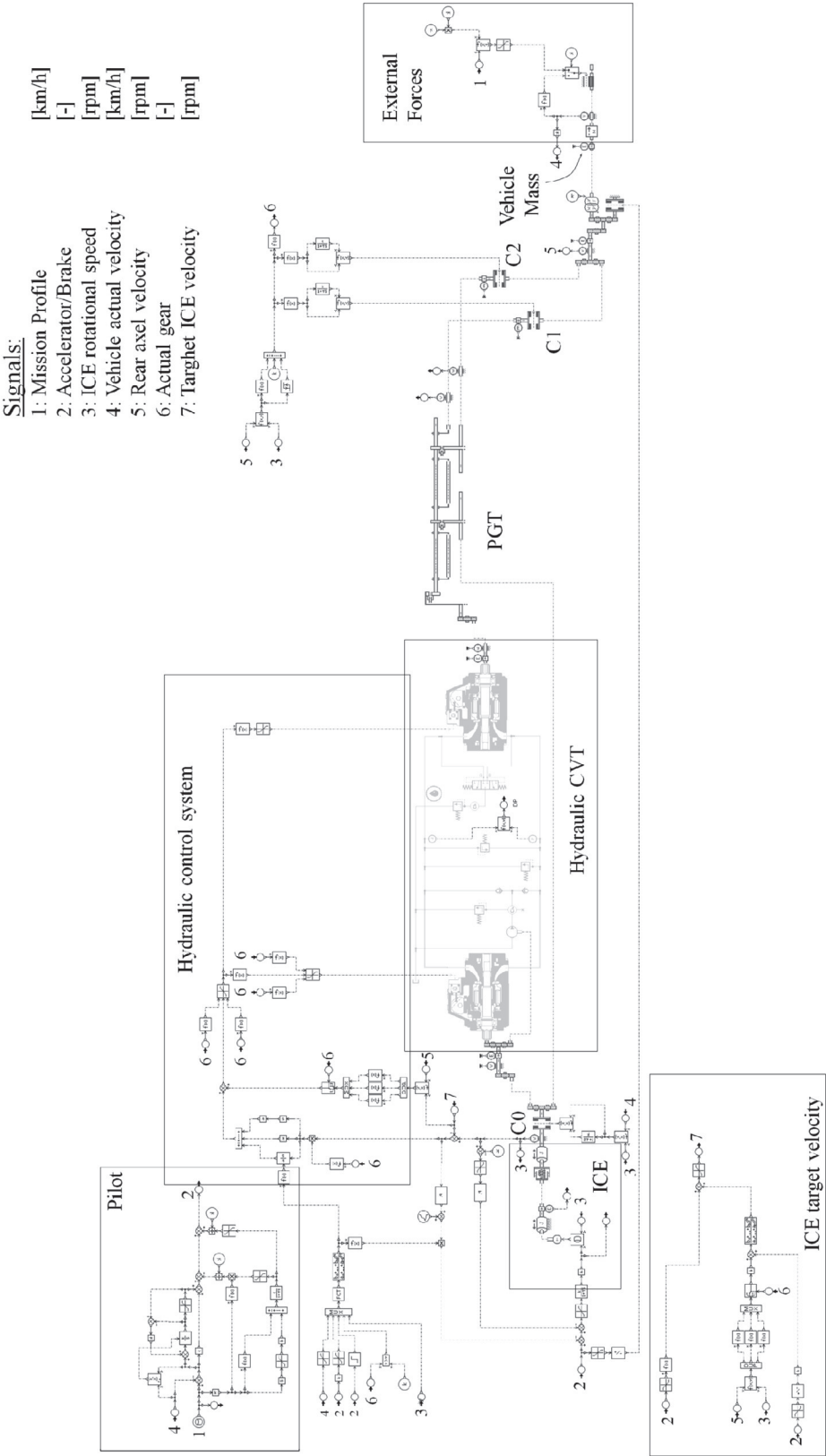
In this work, the potentiality of PSp has been investigated in an application where comfort and performance should be attained despite the discontinuous driving condition: a city bus.

Therefore, the energy performance of a vehicle equipped with a dual stage input coupled PSp transmission was monitored by means of models implemented in the AMESim environment. A

Power Shift Model AMESim v.10.1



Power Split Model
AMESim v.10.1



second model of a standard five gear PSh transmission equipped with a torque converter was also developed. This model was used to compare the PSp performance with a current standard solution for bus transmission. Both solutions were designed to fulfil the power and acceleration characteristics of the test case.

The control strategy of the two transmissions was kept as simple as possible, in accordance with the vehicle characteristics. Both control systems were based on the accelerator position, the engine rotational speed and the vehicle velocity.

At first, a simulation at constant power level equal to half the maximum power was carried out in order to point out the different behaviours of the transmissions and the engines. It has been observed that in all cases the PSh transmission shows greater efficiency than that of the PSp transmission, except where the converter works, i.e. at low speeds. In contrast, its engine does not operate in optimal conditions, especially after each gear change when the engine suddenly slows down.

Significantly better performance was attained by the engine coupled with the PSp solution. In fact, the control system keeps the load and the speed quite close to the minimum specific fuel consumption. This results in almost the same fuel consumption for the two transmissions.

Afterwards, two standard test cycles were simulated. Also in these cases the PSp showed higher total efficiency. As in the constant power test, the improvements obtained in the engine efficiency overcame the lower mechanical efficiency of the transmission.

One might conclude that the PSp transmission could be an interesting alternative to the PSh transmission, not only in agricultural and heavy duty machinery but also in heavy urban transport. In particular, the PSp could reduce fuel consumption where the discontinuity of the driving cycle significantly reduces the average performance of the engine. Furthermore, the capability of controlling the operating conditions on the engine maps allows a minimum consumption control strategy to be developed and to avoid regions characterised by high pollution production.

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Appendix A

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