PERFORMANCE OF A CITY BUS EQUIPPED WITH A TOROIDAL TRACTION DRIVE

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Abstract: - In the paper, a comparison of performances of a city bus when equipped with the traditional automatic driveline consisted of a torque converter followed by epicyclic gear trains and when equipped with a *Double Cavity Half Toroidal CVT* is carried out. A simulation model is developed for evaluating fuel consumptions of the bus when a certain velocity cycle is followed. In particular, results of the comparison are evaluated referring to the ECE15 urban cycle. The model is built considering the hypothesis that the controlling system of the CVT is designed for obtaining in each situation the speed ratio that minimizes the specific fuel consumption and, hence, that allows the engine operation at the best efficiency value.

Key-Words: - CVT, Toroidal CVT, fuel consumptions, city bus, torque converter, epicyclic gear trains.

1 Introduction

In the last few years the attention of researchers has been focused on the improvement of vehicle power trains for a better utilisation of the engine features. The best performances, indeed, can be achieved only optimising at the same time comfort and drivability on one side and fuel saving on the other. More over attention have to be paid to the environmental impact of emissions, which values are strictly regulated according to law.

Continuously variable transmissions (CVT) development and applications have been supported as one of the possible solutions for achieving these goals [1-2]. CVT have been studied, indeed, to be implemented and settled in different fields, not only for vehicles, but each time that performance optimisation has to be reached [3-6].

In this paper a CVT based driveline is proposed for equipping a city bus and the performances obtainable with this design are compared with the ones of a traditional automatic power train that, from now on, will be referred to as AT. Bus data and coefficient values utilised are reported in table 1.

The CVT typology considered is the *Double Cavity Half Toroidal* one, which is a *"Traction Drive"* transmission, able to face high torque values and, hence, suitable for being implemented on a city bus.

Since, on one hand, the CVT minimum speed ratio is higher than zero and it does exist, on the other hand, a minimum operative value of the engine rotating speed, a friction clutch or a torque converter is needed for the standing start phases of the bus. Of course if an Infinitely Variable Transmission (IVT) configuration were considered the problem would be easily overcome [7-9]. With an IVT, indeed, obtained by combining a CVT with an epicyclic gear train, it is possible to get the null value of the speed ratio and, hence, no clutch is requested.

By means of a simulation tool, instantaneous and total fuel consumption are evaluated referring to the ECE15 urban cycle, with the aim of carrying out the analysis in different operative conditions.

$m = 13368\ kg$	$R_0 = 0.465\ m$
$I_w = 1.75\ kgm^2$	$\tau_d = 0.195$
$I_d = 0.175\ kgm^2$	$\tau_{CVT}^{min} = 0.29$
$I_{en} = 7\ kgm^2$	$\tau_{CVT}^{max} = 1.5$
$V_c = 9500\ cm^3$	$\eta_d = 0.97$
$C_x = 0.6$	$\eta_{jn} = 0.99$
$S = 7.15\ m^2$	$\eta_{lb} = 0.99$

Table 1: Bus data and coefficient values

2 Toroidal traction drive CVT

In fig.1 a double cavity half Toroidal CVT is represented. Basically power is transmitted from the input disk to the output disk by means of the interposed rollers. No direct contact takes place between the said surfaces because of the presence of an oil film. A mechanical or hydraulic device has the task of producing the axial force needed for developing the extremely high pressures requested in the contact area. In these severe operative conditions the oil viscosity increases a lot originating, for the relative motion existing between disks and rollers, very high shear stresses, and, hence, the transmission of power. The lubrication conditions can be described by using the hard EHL theory [10].

Regarding the speed ratio, it can be easily and continuously modified by changing the angle of inclination of power rollers. For this reason, although the mechanical efficiency of a Toroidal CVT is lower than the one of a gear box, in several application it's been already demonstrated that since the continuous transmission allows the engine to

operate always at the best efficiency value, the fuel saving is, altogether, remarkable [14].
In the case of a vehicle equipped with a continuously variable power train, the engine working conditions move along an *economy line*. Each point of this line has the property of representing the couple of engine torque and rotating speed values that give a desired power with the best efficiency and, hence with the minimum fuel consumption. Consequently the values of the rotating speed of the engine result to be lower then the ones that are typical of discrete drivelines and the range of variation of this quantity is quite small. In other words if a CVT is implemented for transmitting power, the engine undergoes less to transient phases during which the rotating acceleration is considerable with high production of exhaust emissions.

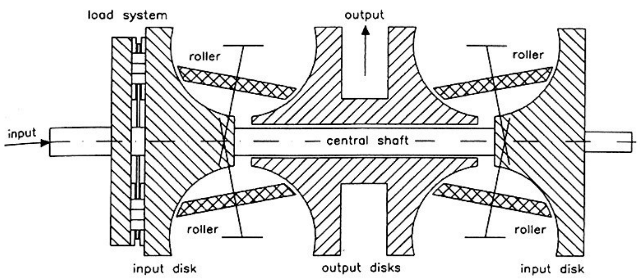

Fig.1: Double Cavity Half Toroidal CVT

From the comfort enhancement point of view it is possible to say that since the speed ratio changing is smooth and it is produced under load conditions, the longitudinal acceleration of the vehicle results to be regular with no interruptions produced during shifting manoeuvres. Passengers, indeed, do not undergo to jerks produced during phases characterised by strong variations of rotating speed and acceleration, and this increases their feeling with regards to this type of transmission design.

3 Performance analysis in the CVT case

In this section a double cavity half Toroidal CVT is considered equipping the city bus.
The needed contact forces are thought generated by a *loading cam* that's a mechanical device able to provide an axial force proportional to the input torque.
The efficiency of the CVT is assumed to be a function of the input torque, according to experimental results [11-13], and not dependent on the engine rotating speed, since the range in which this parameter moves is quite small in the case of a city bus. For small values of the input torque the efficiency is very low due to the fact that the losses in these operative conditions are reasonably big if compared with the input power. When the input torque is high enough the efficiency reaches a value, between 0.90 and 0.92, that remains quiet constant for increasing values of the torque.
The first step for generating the simulation model has been the interpolation of the specific fuel consumption map with the computation of the economy line. In each working condition, indeed, and, hence, in each instant of time it's needed to be evaluated the value of the speed ratio corresponding to the minimum fuel consumption.
After that, once the velocity cycle during which the consumptions have to be evaluated has been chosen, that means with an imposed kinematics of the bus, it is possible to calculate, from the expression of the power requested for vehicle motion, the needed engine power and, hence, the fuel consumption.
In the case of the motion of a vehicle, equipped with CVT, moving on a flat not sloping road with clutch engaged, the needed engine power has the following expression:

$$P_{en}=\frac{\dfrac{\rho C_x S v^3}{2}+\left(f_{id}Z_{id}+f_{dr}Z_{dr}\right)v+\left(m+\dfrac{I_w}{R_0^2}+\dfrac{I_d}{\tau_d^2 R_0^2}\right)av}{\eta\eta_d\eta_{jn}\eta_{lb}}+ \\ +I_{en}\left(\frac{\omega_{en}\dot{\omega}_{en}}{av}\right)av \qquad (1)$$

3.1 Standing start phases

In this paper it is considered the possibility of having the intervention of a friction clutch during bus standing start phases, with CVT speed ratio equal to the minimum allowed. In the model for the evaluation of fuel consumption during this phases, the following hypothesis is considered: the engine rotating speed, equal to 1200 rpm at the beginning, decreases linearly till the value of 900 rpm is reached, when the phase is thought to be ended. The power dissipated by the clutch can be written as follows:

$$P_{clutch}=T_{clutch}\left(\omega_{en}-\frac{v}{R_0\tau_d\tau_c^{\min}}\right) \qquad (2)$$

where T_{clutch} is the torque transmitted by the clutch.

3.2 Acceleration phases

Regarding the equivalent mass of the engine $meq_{en}=I_{en}\left(\frac{\omega_{en}\dot{\omega}_{en}}{av}\right)$, it's important to underline that, when the case of a CVT is analysed, the following relation holds:

$$\frac{\dot{\tau}_c}{\tau_c}=\frac{a}{v}-\frac{\dot{\omega}_{en}}{\omega_{en}} \qquad (3)$$

which shows that the ratio between linear acceleration and velocity of the vehicle is not equal to the ratio between angular acceleration and velocity of the engine shaft as it is in the case of a discrete power train.
Since the needed power is a function also of the time derivative of the speed ratio, the instantaneous fuel consumption is itself related to the variations of this quantity and, hence, the minimization of fuel consumption of a city bus equipped with a CVT, along a certain well defined cycle, corresponds to the evaluation of the minimum achieved by the following definite integral:

$$Min \int_{cycle} c(\tau_c, \dot{\tau}_c, t)\, dt \qquad (4)$$

where the function $c(\tau_c, \dot{\tau}_c, t)$ to be integrated represents the instantaneous fuel consumption.
The procedure for achieving the analytical solution of this variational problem is complicated by the fact that the expression of the relation between the instantaneous fuel consumption and the CVT speed ratio is quiet complex.
It is possible, however, to consider the equivalent mass of the engine constant in each acceleration phase of the cycle and equal to an appropriate average value that can be calculated by following a simple iterative way [14].
It's useful to notice that since during acceleration phases the ratio $\frac{\dot{\tau}_c}{\tau_c}$ is higher then zero and generally is $\tau_c \geq \tau_g$, from eq. 3 it's possible to conclude that $\frac{\dot{\omega}_{en}}{\omega_{en}}\Big|_{CVT} \leq \frac{\dot{\omega}_{en}}{\omega_{en}}\Big|_{discrete}$ and, hence, that $meq_{en}\big|_{CVT} \leq meq_{en}\big|_{discrete}$ in these phases.
From supplied power point of view, this is an advantage since it means that less power is needed if a CVT is implemented on the vehicle and less power means smaller fuel consumption. This effect is, of course, strongly important when the vehicle in analysis is a bus.

3.3 Deceleration phases

During the deceleration phases the control strategy of the vehicle cuts off the fuel flow sent to the engine for having null fuel consumptions. The ratio between vehicle speed and CVT speed ratio is proportional to the engine rotating speed and reduces with time since bus speed is decreasing. For avoiding that engine speed undergoes the minimum allowed regime for engine operation, the control strategy reduces opportunely the CVT speed ratio till the minimum value of this parameter is reached. After that if bus velocity is still reducing, since, for example, bus is being stopped, clutch intervention disconnects the engine from the load and the minimum fuel flow needed for maintaining it in motion at minimum speed is sent again to the cylinders.

4 City bus traditional AT

In this section a traditional AT [15] is considered equipping the city bus. It consists, as anticipated of a torque converter followed by epicyclic gear trains.
The torque converter operates only during the standing start phases of the bus, providing a remarkable torque multiplication. The higher is the relative speed of the turbine with respect to the pump, the higher is the torque multiplication.
A lock-up clutch is implemented with the aim of eliminating losses related to the torque converter, when a direct transmission is allowed. The clutch, indeed, provides a direct mechanical link between the engine and the epicyclic gear box when the bus velocity and the torque loading are respectively high and low enough.
The torque converter performances are evaluated referring to diagrams provided by the supplier.
Regarding the epicyclic gear box, it consists of three epicyclic gear train that can be combined in different ways making available five speed ratios and the reverse gear.

4.1 Torque converter

In fig.2 the scheme of the AT thought equipping the city bus is represented. As said, the torque converter operates only in the standing start phases and in particular situation in which, with the engine rotating at the minimum speed, the high torque multiplication is utilised for making available the needed traction force (speed along a steep slope, strong acceleration moving from low values of the bus velocity, etc.). The said lock-up clutch makes the pump and the turbine connected, linking the engine directly to the planetary gear box, when the torque converter is useless, increasing the whole transmission efficiency.

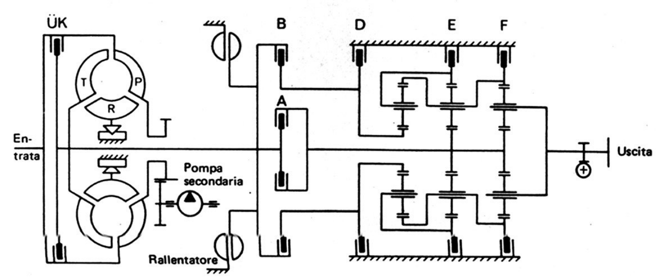

Fig.2: Scheme of the considered AT (*e.g.* [*15*])

At the beginning of the standing start, the turbine/pump speed ratio ν is null and the corresponding turbine/pump torque ratio τ is maximum. This situation is referred to as *stall condition* of the torque converter. The hydraulic efficiency η_y of the torque converter is null in this condition and the whole engine power is transmitted, as heat, to the oil. With increasing the turbine speed and, hence, for higher values of ν, τ decreases while η_y, of course, goes up. For evaluating behaviours and performances of the considered engine-torque converter pairing, it's needed, as usual, to have available the *primary diagrams* obtained from the supplier. Recall, indeed, that the following relation holds:

$$\frac{T_p}{n_p^2} = f(\nu) \qquad (5)$$

4.2 Epicyclic gear box.

As said, in the AT considered equipping the bus, three epicyclic gear trains are present, following the torque converter, and they can be combined in different ways making available five different speeds plus the reverse gear. For having this result brakes and clutches are utilised: brakes for hampering at all the rotation of certain gears, friction clutches for making two elements of a certain train rigidly connected (fig.2). A control strategy is implemented for governing the whole power train: shifting manoeuvres are imposed and related to several parameters among which very important is, of course, the vehicle speed.

SPEED	*I*	*II*	*III*	*IV*	*V*
EPICYCLIC GEAR TRAINS INVOLVED	*3*	*2, 3*	*1, 2, 3*	*-*	*1, 2, 3*
SPEED RATIO	*0.291*	*0.497*	*0.704*	*1*	*1.20*
EFFICIENCY	*0.989*	*0.991*	*0.992*	*1*	*0.994*

Table 2: Planetary gear box characteristics

In table 2, speed ratios obtainable from the planetary gear box are reported with the way in which the three epicyclic trains are combined in each speed and with the mechanical efficiency corresponding to each gear. The efficiency in each gear can be evaluated on the base of the following considerations. Since the power of the friction forces acting on parts in relative motion, that's dissipated in heat, is the same in each reference frame it is possible to consider that particular frame joined to the *planet carrier*, that's a relative frame in which the epicyclic gear train results to be a *compound gear train*. If the efficiency of the gear train in this reference frame is known and equal to η_0, assuming that it is not dependent on the torque values, it is possible to evaluate the efficiency η in the generic operative condition of the train as a function $\eta = \eta\,(\eta_0,\ \tau_W,\ \tau_{s1\text{-}pc},\ \tau_{s2\text{-}pc})$, where τ_W represents the ratio between the angular velocities of the two *sun gears* in the frame joined to the planet carrier, while $\tau_{s1\text{-}pc}$ and $\tau_{s2\text{-}pc}$ are the speed ratios between the two sun gears and the planet carrier in that particular working condition. The reasoning can be, of course, extended to systems consisting of two or more epicyclic gear trains. Values τ_W for the three epicyclic gear trains are in table 3.

EPICYCLIC GEAR TRAIN	*1*	*2*	*3*
τ_W	*-0.411*	*-0.410*	*1.41*

Table 3: Values τ_W of the three epicyclic gear trains

4.3 Auxiliary pump

Brakes and clutches used for making available the different gears are hydraulically activated and a rotative pump, moved by the engine shaft, generates the needed overpressure. For evaluating power P_{ap} needed for pump operation it is possible to move as follows. First of all the element interested by the higher torque load needs to be singled out. Then it is possible to calculate the closing force and, hence, the needed pressure.

5 Performance analysis in the AT case

As previously seen, by using the eq. 1, it's possible to calculate the needed engine power for bus motion on a flat not sloping road. Also in the case of the traditional AT, in the phases during which the torque converter is active, speed ratio changes in a continuous way and, again, the result condensed in eq. 3 has to be taken into account to evaluate correctly the engine equivalent mass.

5.1 Standing start phases

This phases can be split into two parts. During the first part the torque converter is operative. The end of the first part corresponds to the time when the lock-up clutch starts working for making pump and turbine rigidly connected. The second part consists of the time needed to the friction surfaces of the clutch for stopping sliding. The second part, hence, ends when pump and turbine are effectively rigidly connected. During the whole phase, the epicyclic gear trains are configured in the way corresponding to the *I* gear and since the kinematics of the bus is imposed by the velocity cycle to be covered, in each instant of time the rotating speed of the turbine is known.

Regarding the first part of the phase it is possible to write eq. 1 in the following way:

$$P_t = \frac{\dfrac{\rho C_x S v^3}{2} + \left(f_{id} Z_{id} + f_{dr} Z_{dr}\right) v + \left(m + \dfrac{I_w}{R_0^2} + \dfrac{I_d}{\tau_d^2 R_0^2}\right) a v}{\eta \eta_d \eta_{jn} \eta_{lb}} \qquad (6)$$

in which the term related to the kinetic energy variation of the engine does not appear because the power at left hand side is the power generated by the turbine torque. Eqs. 5 and 6 constitute a system in the two unknowns v and T_p, that can be, then, calculated in each instant of the period of time that the first part lasts. If $v(t)$ is known also $\omega_p(t)$ is known and, hence, $\omega_{en}(t)$ and $\dot{\omega}_{en}(t)$. But since it is also:

$$P_{en} = P_p + I_{en}\omega_{en}\dot{\omega}_{en} + P_{ap} \qquad (7)$$

the needed engine power and, hence, the corresponding fuel consumption can be evaluated. The control strategy of the transmission determines when the first part has to finish.

Regarding the second part of the phase it is possible to consider it lasting 0.5 s. Since the instant of time when this second part starts is known for what said about the first one, it is possible to evaluate engine speed and deceleration by posing the hypothesis that engine shaft reduces its velocity linearly with time. Since, as usual, kinematics of the bus is imposed by the velocity cycle followed and, hence, the rotating speed of the turbine is known, it is possible to use again eqs. 5 and 6 modified for taking into account the power generated by the clutch torque T_{clutch} as follows:

$$P_t + T_{clutch}\omega_t = \frac{\dfrac{\rho C_x S v^3}{2} + \left(f_{id} Z_{id} + f_{dr} Z_{dr}\right) v + \left(m + \dfrac{I_w}{R_0^2} + \dfrac{I_d}{\tau_d^2 R_0^2}\right) a v}{\eta \eta_d \eta_{jn} \eta_{lb}} \qquad (8)$$

This time the system is in the unknowns T_{clutch} and T_p, that can be, then, calculated in each instant of the period of time that the second part lasts.

The needed engine power can be evaluated as follows:

$$P_{en} = P_p + I_{en}\omega_{en}\dot{\omega}_{en} + P_{ap} + T_{clutch}\omega_{en} \qquad (9)$$

and, hence, the corresponding fuel consumption.

5.2 Acceleration phases

Once pump and turbine are rigidly connected and a certain gear is engaged, τ be the speed ratio, the engine equivalent mass is known and the following relation holds:

$$P_{en} = \frac{\frac{\rho C_x S v^3}{2} + \left(f_{id}Z_{id} + f_{dr}Z_{dr}\right)v + \left(m + \frac{I_w}{R_0^2} + \frac{I_d}{\tau_d^2 R_0^2} + \frac{I_{en}}{\tau^2\tau_d^2 R_0^2}\right)av}{\eta\eta_d\eta_{jn}\eta_{lb}} + P_{ap} \qquad (10)$$

Since, again, bus kinematics is imposed by the velocity cycle chosen, eq. 10 is sufficient for evaluating the engine power needed. Shifting manoeuvres are imposed by the control strategy of the power train and are related to several parameters among which, again, the bus velocity plays a very important role.

5.3 Deceleration phases

During deceleration phases an hydraulic brake, interposed between the torque converter and the planetary gear box, becomes active supporting friction brakes. In this phase the transmission is not set in *neutral configuration*. If the deceleration phase ends stopping completely the bus, the hydraulic brake is disconnected and the transmission goes automatically in neutral configuration, since the control electronic unit makes alive the program *Neutral when Bus Stops*. From consumption evaluation point of view, during the deceleration phase consumption is null because of the cut-off intervention. When the bus stops and the transmission is put in neutral configuration the only needed power is the one related to the auxiliary pump operation.

5.4 Shifting phases

During shifting manoeuvres the control unit reduces the fuel flow and, hence, the relative consumption. It is possible, then, consider null the engine power supplied.

6 Fuel consumption of the bus following the ECE15 urban cycle

In this section the analysis of performances of the city bus when equipped with the traditional AT and when equipped with the Toroidal CVT is carried out by comparing the fuel consumption in the two cases.
As anticipated, the comparison is made considering the bus running along the ECE15 urban cycle, that lasts altogether 780 s with an average vehicle speed equal to 18.7 km/h. In particular the attention is focused on the elemental urban sub-cycle having duration equal to 195 s and during which the maximum reached speed is equal to 50 km/h. A short urban run is, hence, reproduced, useful to study bus fuel consumption during several acceleration phases, each one preceded by a standing start phase. More over it is possible to see what does happen during low and constant speed phases.
Recalling what said regarding the shifting manoeuvres in the case of the AT, it's important to underline that the two velocity cycle, corresponding to the two different typologies of drivelines equipping the bus, are slightly different. In the CVT case, indeed, the possibility of performing the variation of the speed ratio value in a continuous manner allows a constant value of the vehicle acceleration during a ramp. In the case of the AT, instead, when a shifting manoeuvre is performed the engine power becomes null, as the acceleration does, giving rise to a short phase during which the velocity is quiet constant.
Results of the comparison in terms of instantaneous fuel consumption are reported in fig.3.
Performing the ECE15 urban cycle the bus equipped with the traditional AT covers altogether 994.7 m and burns 265 g of fuel: the average consumption is 3.75 m/g.
The bus equipped with the CVT covers, instead, 994.0 m and burns 244 g of fuel with an average consumption of 4.07 m/g.
The average per cent fuel saving along the cycle obtainable adopting a CVT transmission is about 8.5%.

6.1 First acceleration ramp phase

When the first acceleration ramp of the cycle is concluded the bus velocity is equal to 15 km/h and it is wholly covered in I gear, referring to the AT case. This phase represents the typical standing start of the vehicle. Results show how at the beginning of the phase, consumption is lower in the case of the AT. This is related to the fact that the torque ratio τ of the converter in stall condition is the maximum possible. This means, indeed, that engine torque and power requested are smaller than the ones needed in the CVT case where the standing start phase is performed with the intervention of the friction clutch. Once the friction surfaces sliding phase is concluded, the torque transmitted by the clutch increases and CVT becomes more economic than the other power train. This does happen since on the other side the torque ratio τ of the torque converter decreases with the speed ratio ν , with the effect of requesting more power to the engine with higher consumption.
After 2.4 s the standing start phase of the bus equipped with CVT is finished and during the remaining part of the acceleration ramp CVT allows smaller consumption then the AT, besides the phase during which the locking procedure of the torque converter is performed.
It's important to observe that during the whole phase the engine power requested is always not exceeding 80 kW and this is the engine operative range in which specific consumption variations with the rotating speed are remarkable. In this condition CVT capability of making the engine to operate at peak efficiency gives rise to a considerable reduction of fuel consumption. On the other side the equivalent mass of the engine is bigger in the case of the traditional AT than in the other, and this translates in

a higher needed power and, hence, in a higher fuel flow requested. More over the torque converter is operative in almost the entire phase, affecting the whole resulting efficiency with its own one, that is never exceeding 0.8.

6.2 Second acceleration ramp phase

When the second acceleration ramp of the cycle is concluded the bus velocity is equal to 32 km/h and the epicyclic gear trains are configured in the way corresponding to the *III* gear, referring to the AT case. For the initial phase during which the bus moves off, it is possible to repeat considerations reported in the previous section.

Regarding the remaining part of the ramp, it's important to notice that the per cent fuel saving obtainable with the CVT as regards to the other power train typology becomes smaller when the increasing bus velocity produces the shifting in *II* and *III* gear. The ratio between the two instantaneous fuel consumptions is, indeed, about 1.5 and more when the epicyclic gear trains are configured in the way corresponding to the *I* gear, while it is equal to 1.2 and less with the planetary gear box configured for the *II* and the *III* gear.

This result can be explained recalling that first of all the torque converter is not active for gears following the *I*, not affecting, hence, the global efficiency of the transmission. Second, the difference in terms of equivalent mass of the engine becomes smaller for gears following the *I*, and third, the more the vehicle velocity increases, the less the corresponding engine operative conditions are sensible to fuel consumption variations with the engine rotating speed.

At last it has to be observed that, again, the diagram of instantaneous fuel consumption of the phase regarding the case of the AT presents strong reduction of this quantity when the torque converter is locked and when the two shifting manoeuvres take place.

6.3 Third acceleration ramp phase

When the third acceleration ramp of the cycle is concluded the bus velocity is equal to 50 km/h and the epicyclic gear trains are configured in the way corresponding to the *IV* gear, referring to the AT case.

Results seen in the previous two acceleration phases can be repeated. In particular the ratio between the two instantaneous fuel consumptions, the CVT one and the AT one, is equal to 1.5 and more when the epicyclic gear trains are configured in the way corresponding to the *I* gear, is around 1.2 in the *II* gear case, is close to 1.1 with the planetary gear box configured for the *III* gear, becoming *less than 1* in the *IV* gear case.

This results can be explained thinking that on one hand the difference in terms of equivalent mass of the engine becomes negligible in this condition and, on the other, the worse mechanical efficiency of the Toroidal CVT affects the whole result. At last, since the needed power is around 140 kW, the operative engine conditions are so that the effect of reduction of fuel consumption related to optimal choice of the rotating speed regime is negligible.

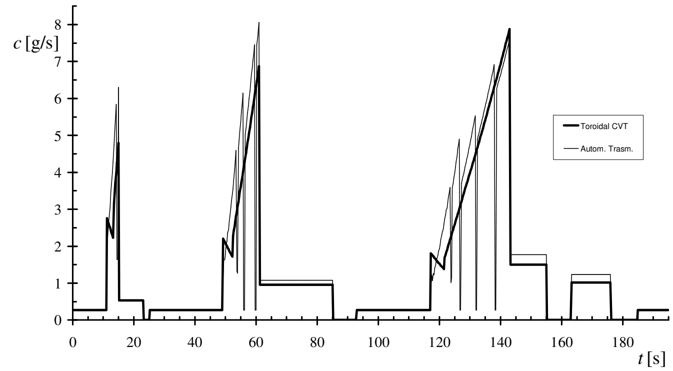

Fig.3: Instantaneous fuel consumption versus time for both the considered transmissions

6.4 Constant velocity phases

The first phase during which bus velocity is get constant is with speed equal to 15 km/h. The instantaneous fuel consumption is equal to 0.53 g/s for the bus equipped with the CVT and to 0.54 g/s for the one equipped with the AT. The difference is negligible and the reason is related to the low efficiency performed by the CVT when the engine torque is small as in this case.

The second constant velocity phase is with speed equal to 32 km/h. The instantaneous fuel consumption is equal to 0.96 g/s for the bus equipped with the CVT and to 1.1 g/s for the one equipped with the AT. The *III* gear is engaged. The difference is this time quiet remarkable since the per cent difference is about 11%. Since the CVT efficiency is now at its best values, the opportunity offered by CVT of generating the engine speed corresponding to the best engine efficiency for each needed power value is now winning.

The third phase at constant speed is with velocity equal to 50 km/h. The instantaneous fuel consumption is equal to 1.5 g/s for the bus equipped with the CVT and to 1.8 g/s for the one equipped with the AT. The per cent fuel saving guaranteed by the CVT is about 16%.

The fourth constant velocity phase is with speed equal to 35 km/h. The instantaneous fuel consumption is equal to 1.0 g/s for the bus equipped with the CVT and to 1.2 g/s for the one equipped with the AT.

6.5 Comments on results and considerations regarding comfort

If from fuel saving point of view the adoption of a Toroidal CVT seems to be encouraging, it is important to carry on considerations about the impact of a similar choice could have on passenger comfort and, hence, also in this field a comparison has to be done.

In fig.4 the trend of bus acceleration versus time is reported during the second and the third acceleration ramp phases for both the transmissions. As it is possible to see during a ramp, bus acceleration remains constant if the vehicle is equipped with the CVT. This is due to the fact that for a CVT the speed ratio can be modified with load always

acting, in a continuous manner. In the case of the traditional AT, instead, shifting manoeuvres have to be performed, each one lasting a very brief period of time during which an interruption of engine power flow is imposed by the control strategy of the power train. This means that the bus acceleration goes down for a while.
For this considerations it's possible to say that a CVT equipped with city bus is more comfortable than the traditional one. Instantaneous variations of bus acceleration, typical of the epicyclic gear trains shifting manoeuvres are negatively felt by all the passengers, interesting more the ones that stand up. More over it has to be underlined that besides the phases in which it is null, acceleration is always higher than the one obtainable when the bus has a CVT. During the second ECE15 acceleration ramp, for instance, it does exist a per cent difference that move from 2.8% up to 23%. This translates, of course, in higher values of inertia forces acting on passengers and in a strong reduction of their comfort, especially for people standing up.

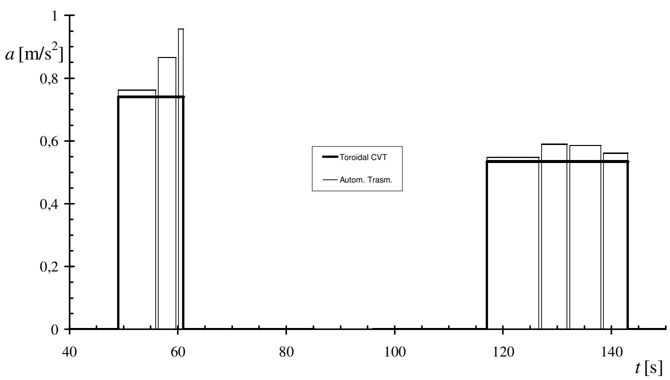

Fig.4: Comparison of bus acceleration along the second and the third ramp of the ECE15 urban cycle

In conclusion a CVT equipping a city bus, besides reaching the goal of performing smaller fuel consumptions gets better comfort and feeling of passengers during acceleration phases. This is a remarkable point for a city bus that is strongly employed on urban paths characterised by frequent stops and a lot of standing start and acceleration phases.

6.6 Considerations about the engine rotating speed

The engine of a bus adopting a CVT operates at a rotating speed always lower than the one having the traditional AT. The comparison is reported in fig.5.
The Toroidal CVT utilised in this paper, indeed, is capable of changing under load conditions, in a continuous way, the speed ratio getting the engine operation moving on the previously defined economy line. More over the Toroidal Drive is characterised by a speed ratio range between 0.29 and 1.50 and this is important when the bus velocity is higher since this speed values can be got with the engine operating at mid-low regimes. Since, then, the vehicle considered is a city bus this is strongly important. The adoption of a Toroidal CVT, besides what said regarding fuel saving and comfort, could reduce a lot transient phases characterised by high values of engine shaft rotating acceleration during which polluted emissions values and noise level can be remarkable.

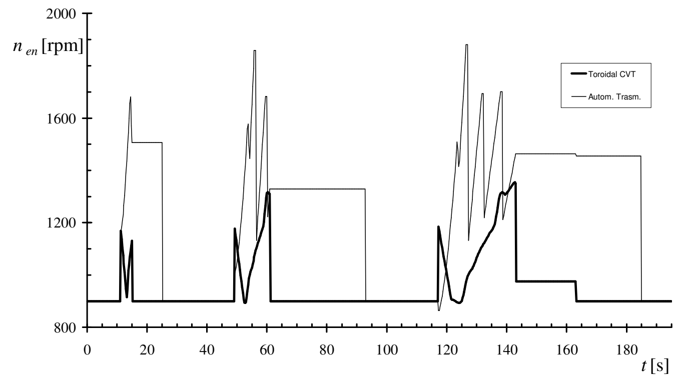

Fig.5: Comparison of engine rotating speeds performed along the ECE15 urban cycle

7 Conclusions

In the paper a simulation model is developed with the aim of comparing the performance of a city bus for transport of passengers strongly employed along urban paths, when equipped with the traditional automatic transmission (AT) composed by torque converter and epicyclic gear trains and when equipped with a Double Cavity Half Toroidal CVT. The idea followed in the paper is that the CVT, if effectively controlled with an appropriate strategy, could give rise to a sensible fuel saving as regards to the traditional transmission and, more over, a better utilisation of the engine and a better comfort for passengers conveyed. From environment saving point of view the analysis performed seems to indicate that CVT adoption could produce a certain reduction of polluted emissions and a lower level of noise. All this features can be got because of the possibility offered by the CVT of changing the speed ratio in a continuous way under load conditions. It's possible, then, once the engine needed power is known to get the speed ratio that allows to obtain that value of the power at the rotating speed corresponding to the minimum fuel consumption. In other words CVT allows the engine to operate along an economy line on the engine map, consisting of points having the property of being operative conditions of minimum fuel consumption.
The comparison has been carried out considering the bus moving along the ECE15 urban cycle.
The results show that the average per cent fuel saving along the cycle obtainable adopting a CVT transmission is about the 8.5%.
Fuel saving with regards to the AT is higher when the bus velocity is not so high and, hence, when the vehicle moves off after a stop and an acceleration phase follows. This kind of behaviour is typical of a city bus, which paths are consisting of frequent stops and standing start phases with an acceleration ramp following.
But the possibility offered by CVT of continuously varying

the speed ratio under load conditions gives rise during this frequent acceleration ramp to a constant rate of changing of the vehicle speed with the result of eliminating shifting manoeuvres characterised by discontinuity of acceleration signal related to power flow interruption and different values of this quantity in each gear. In other words with the CVT it's possible to eliminate step variations of inertia forces acting on people on board, producing better comfort conditions.
More over since the engine rotating speed is lower and less frequently variable for the bus equipped with the CVT, emissions produced in transient phases characterised by strong rate of changing of the engine shaft angular velocity could be reduced as the corresponding level of noise.

Nomenclature:

a	bus acceleration	[m/s^2]
c	instantaneous fuel consumption	[g/s]
C_x	aerodynamic drag coefficient	
f_{dr}, f_{id}	drive and idle wheel rolling drag coefficient	
I_d, I_{en}, I_w	equivalent inertia moment of the whole drive train, of the engine, of the wheels	[kgm^2]
m	bus mass	[kg]
meq_{en}	equivalent mass of the engine	[kg]
n_{en}	engine rounds per minute	[rpm]
n_p	torque converter pump rounds per minute	[rpm]
P_{ap}	auxiliary pump power	[W]
P_{clutch}	clutch power dissipation	[W]
P_{en}	engine power	[W]
P_p, P_t	torque converter pump and turbine power	[W]
R_0	tyre rolling radius	[m]
S	bus frontal section	[m^2]
T_{clutch}	friction clutch torque	[Nm]
T_p, T_t	torque converter pump and turbine torque	[Nm]
X	drive wheel traction force	[N]
v	bus velocity	[m/s]
V_c	engine displacement	[m^3]
Z_{dr}, Z_{id}	drive and idle wheel normal load	[N]
η	generic operative gear box efficiency	
$\eta_d, \eta_{jn}, \eta_{lb}$	efficiency of the final compound gear train, of joints and rising from lubricant shaking	
ρ	air specific mass	[kg/m^3]
τ	generic operative condition speed ratio	
τ_c, τ_d, τ_g	CVT speed ratio, final compound gear train speed ratio, discrete gear box speed ratio	
ω_{en}, ω_t	engine and torque converter turbine rotating speed	[s^{-1}]
$\dot{\omega}_{en}$	engine rotating acceleration	[s^{-2}]

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