Optimal designs of a hydromechanical hybrid transmission by minimizing fuel consumption and damage to human health

Nicola Andretta[1], Antonio Rossetti[2], Alarico Macor[1]

1-Department of Engineering and Management, University of Padua, Stradella San Nicola 3,36100 Vicenza, Italy; nicola.andretta.1@phd.unipd.it

2-Istituto per le Tecnologie della Costruzione, Consiglio Nazionale delle Ricerche (CNR), Corso Stati Uniti 4, 35127 Padova, Italy; antonio.rossetti@itc.cnr.it

Abstract.

The hybrid hydromechanical transmission must be sized by fully exploiting its strengths: the energy contribution of the hydraulic source and the possibility of managing the thermal source at the most convenient conditions.
Therefore, in this paper we propose a simulation-based optimisation which searches the design parameters values of both transmission and hydraulic source while minimizing the fuel consumption along a typical route.
A second objective function involving emissions was also considered. The difficulty of managing numerous chemical species with different dangers has been overcome by summarizing them in a single parameter: the damage to human health. In this way the concept of emission correspond with the concept of damage to health.
A study was carried out on a 12 m class urban bus.
The results of this study allow us to evaluate the effects of the hybridization of a hydromechanical transmission and the effects of the engine management criteria, not only on the optimal values of the design parameters, but also on fuel consumption and damage to human health.

Keywords. Hybrid hydromechanical transmission, Power-Split hydraulic hybrid, minimum fuel consumption criterion, minimum emissions criterion.

1. Introduction

Powertrain hybridization is a technique implemented by the automotive industry in recent years to reduce fuel consumption. Hydraulic hybridization, less widespread than electric hybridization, has some characteristics that make it interesting in some specific applications. In fact, the high energy density and low specific power of electric batteries direct electric hybridization towards light vehicles, while the high specific power of hydraulic accumulators direct hydraulic hybridization towards applications characterized by frequent starts and stops, such as heavy vehicles [1, 2] and urban transport vehicles [3-8]. Hydraulic hybridization has been applied both to the classic hydrostatic transmission, showing significant consumption savings, and to the hydromechanical transmission [9]. The latter promises further efficiency benefits, although implementation of the hybridization could not be without problems, since this aspect was not faced thoroughly by literature. The need for sensors to measure the engine power and for a suitable control system to managing the engine and transmission leads to an increase in costs. This is a topic for future research development in this area. Additional hydraulic components, however, should not create reliability and cost issues, as it is a mature technology.

Ivantysynova patented the hybrid hydromechanical transmission in 2008 [10]. Van de Ven [11], Kumar and Ivantysynova [12], Kwon and Ivantysynova [13] studied it in the following years.

The hydromechanical transmission shows some advantageous features: it can recover the energy from braking, or it can level the engine operation to recharge the hydraulic source; as a continuous transmission, it allows the engine to operate at a speed independent of that of the wheels, leaving full freedom for its management. Thus the engine can be operated at speeds that minimize fuel consumption or emissions. However, the emissions are many and have opposing genesis. This creates difficulties in a management procedure and forces some compromises.

Some authors consider the emissions within the objective function as the sum of all emissions, implicitly considering them all equally dangerous [14-15]; Dollar et al. [16] only take into account NOx emissions, and Johri and Filipi [17] only NOx and PM, considering them the most dangerous, but neglecting the others. Masih-Tehrani considers emissions weighted on the basis of standard limits [18].

The authors proposed, instead, another way to take into account the emissions, which involves unifying emissions through a common parameter: the damage to human health caused by each of them [19]. In this way all emissions are summarized in a single parameter, which therefore becomes the basis for a procedure completely similar to that for fuel consumption [20].

The hydromechanical transmission, therefore, offers the possibility of producing the same power while producing fewer emissions. This aspect has a certain importance in light of the increasingly stringent regulations on emissions.

However, hybridization and engine management cannot be fully exploited without both entering into the transmission sizing procedure, which, given the complexity of the problem, must be transformed into an optimization process. Many authors in the literature have addressed sizing in these terms [21-26].

This work proposes an optimal sizing procedure that involves both the transmission with its components and the engine with its management criterion. The procedure, starting from some assigned data (engine, transmission layout, maximum vehicle speed, maximum torque at the wheels and typical load cycle), identifies the value of the design variables (gear ratios, displacements of the hydraulic units, size of the hydraulic source) which minimize fuel consumption or emissions.

The aims of this study are to quantify the advantage of hybridization, and to evaluate the influence of the management criterion on design variables, consumption, and emissions.

2. THE HYDROMECHANICAL HYBRID TRANSMISSION

The hydromechanical transmission taken as the basis for the study is the classic Output Coupled (hereinafter referred to as OC) [9]. The hybrid transmission (hereinafter HOC) is derived from the latter by adding the hydraulic source and the related valves (Figure 2). The layout of both schemes is based on the one proposed by Kwon (13).

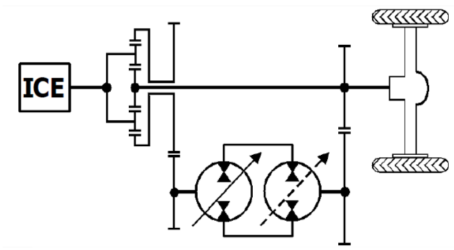


Figure 1. Scheme of the OC transmission.

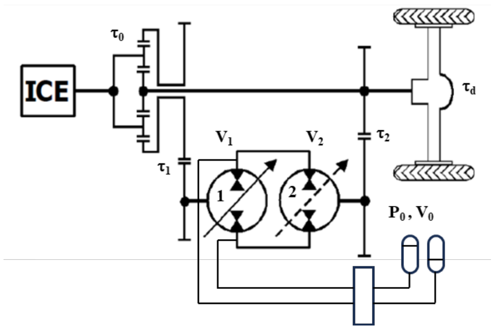


Figure 2. Scheme of the hybrid hydraulic transmission HOC.

The OC transmission works in the following modes.

- Power additive mode: the engine power is divided into the two branches of the transmission, the mechanical branch and the hydraulic branch; the two powers are then mixed in the final spur gear.
- Full mechanical point: all the power passes through the mechanical branch, since the epicyclic ring is stationary. It is the highest efficiency operation.
- Power recirculation mode: in order to increase the speed of the vehicle beyond the Full mechanical point speed value, the Unit 2 must invert its displacement, and transform from a motor to a pump. Therefore, some of the power flows back

through the hydrostatic group, in which the two machines have reversed their operation.
- Braking mode: Unit 1 is put at zero displacement while Unit 2, thanks to the inertia of the vehicle, transforms into a pump, raising the pressure of the low pressure branch. The safety valve laminates the flow rate, sending it to the other branch, creating the braking resistance. A mechanical brake integrates the hydraulic one to guarantee braking service in all conditions.

The HOC transmission has the same three modes of operation with some differences.

Starts and restarts occur solely through the power supplied by the accumulator, activated by the distributor; when the pressure in the accumulator reaches the pre-charge value it is no longer able to provide power and so the engine, previously at idle, is accelerated and the displacement of Unit 1 is increased. The braking phase occurs by setting the displacement of Unit 1 to zero and inverting the accumulator distributor. In this way the inertia of the vehicle transforms Unit 2 into a pump which charges the accumulator.

The sizing method of the OC transmission proposed by Blake [27] was used here in the optimization procedure.

3. ENGINE MANAGEMENT CRITERIA

The continuously variable transmission separates the engine angular velocity from the vehicle speed. This allows the engine to be managed according to various criteria. The first and most immediate is that of minimum fuel consumption.
In the engine map, the set of minimum points for BSFC (Brake-specific Fuel Consumption) can be identified and expressed as a function of the power

$$BSFC|_{min} = f(P|_{min}) \tag{1}$$

This set is defined as Optimal Operating Line (OOL) and can be used in the control system to force the engine to produce the required power at the minimum fuel consumption velocity [20], as shown in Figure 3.

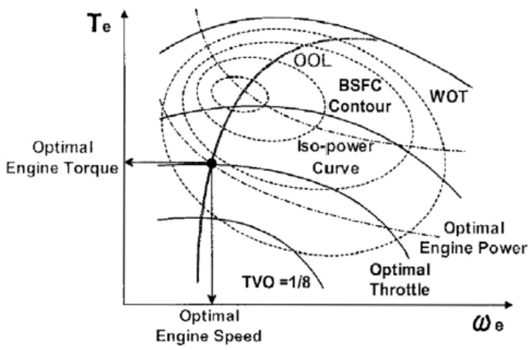


Figure 3. Engine map: iso-BSFC lines, iso-power lines, and optimal operating line [20].

A similar set of minimum is also possible for emissions, although they are divided into four groups (CO, H, NOx, and PM) and have different genesis. As the authors suggested in [19], the four groups of emissions can be unified through the concept of damage to human health that each of them produces. The damage to human health is quantified by the average number of years of life lost by a population due to premature death or disability caused by exposure to pollutants, and measured in DALY unit. The damage caused by the emission of a substance can be assessed according to the Jolliet methodology [28]:

$$D(T,n) = m(T,n) \cdot cf \cdot sd \tag{2}$$

where D [DALY] is the damage to human health due to the emission having mass m [kg]; cf [kg_{ref}/kg] is the characterization factor of the substance belonging to a certain impact category, sd [DALY/kg_{ref}] is the specific damage for that impact category. Summing the damage of all the emissions, a total damage is defined. In this way, a line of minimum total damage can be identified and used in the control system for minimal emission operation.

4. TRANSMISSION DESIGN AS AN OPTIMIZATION PROBLEM

The optimal design procedures are based on the classic optimization problem, which in more general terms can be generally stated as:

Find $x = \begin{bmatrix} x_1 & x_2 & .. & x_n \end{bmatrix}^T$ *minimizing* $f(x)$*; subject to the constraints* $g_j(x) \le 0 \; j = 1..m$ *and* $l_j(x) = 0 \; j = 1..p$

where:

- $x_1 \quad x_2 \quad .. \quad x_n$ are the n^{th} free optimization variables (design variables);
- the equality constraints $l_j(x) = 0 \; j = 1..p$ are the design parameters;
- the inequality constraints $g_j(x) \le 0 \quad j = 1..m$ are the design constraints;
- $f(x)$ is the objective function of the problem.

For the case of the hybrid hydromechanical transmission, the objective function f(x) can be the engine fuel consumption or the total damage to human health.

The design variables are:

• the gear ratio of the planetary gear τ_0 (see Figure 2)

• ger ratio of the ordinary gears τ_1, τ_2, τ_d

• the displacement of the hydraulic units V_1, V_2

• the precharge pressure and the volume and of the accumulator p_0, V_0.

The design parameters are:

• engine maximum power and speed

• maximum vehicle speed

• maximum torque at wheel

• wheel radius.

The design constraints are:

• maximum pressure of hydraulic circuit

• maximum speed of hydraulic machines as a function of displacement.

The load cycle should consider a typical speed profile for the vehicle with starts and stops, and maximum power peaks.

5. THE CASE STUDY: THE CITY BUS

The study was conducted on a 12m class city bus, whose main characteristic data are shown in Table 1.

Table 1. Main data of the vehicle.

Engine- Turbodiesel	Max Power/Speed 228 kW/2200 rpm Max Torque/Speed 1300 Nm/1200 rpm Displacement/Cylinders 8600 cc/6
Vehicle Mass/max passengers	11000 kg/45x70kg
Wheel Radius	0.5281 m
Maximum Vehicle Speed	90 km/h
Maximum Pulling Torque	25000 Nm

6. THE VEHICLE MODEL

The city bus was modeled in the Amesim environment [29]. The complete model is composed by the sub-models of the engine, the hybrid transmission HOC (according to the layout in Figure 2), the load, and the sub-models of the control systems. The vehicle loads, i.e. the rolling friction and the aerodynamic friction, were respectively modelled according to $F_f = f\ m_v\ g$ and $F_{aer} = 0.5\ v^2 A\ \rho\ c_d$; where for the rolling resistance coefficient, the drag coefficient, frontal area the following values were respectively assumed: $f = 0.01$, c_d=1,17, $A = 7$ m^2. The mass m_v is equal to the constant mass of the vehicle and passengers (see Table 1), plus the transmission variable mass as detailed in point 6.4.

As regards the losses in the gears, a constant efficiency equal to 0.98 was assumed for each spur gear, while for the planetary gear, an efficiency equal to 0.97 was assumed for each pair of engaged wheels.

The non-hybrid hydromechanical transmission model (OC, Figure 1), which is taken here as a reference for comparisons, was obtained from the previous hybrid transmission model, leaving the accumulator directional valve always closed.

6.1. *Control strategy*

The transmission control strategy involves separating the energy contributions of the two energy sources: the starting and pickup are assigned to the hydraulic source, the rest to the thermal source. Therefore two control systems in sequence were adopted: the first one, based on the vehicle speed error signal, manages only the displacement of Unit 2 to pursue the desired speed; the second one, on the basis of the same signal, acts sequentially first on the displacement V_1 and then on the displacement V_2.

In OC transmission only the second control works.

6.2. *Minimum fuel consumption and minimum emission lines*

The OOL line was obtained from the engine map by crossing the isopower hyperbolas with the isoconsumption curves (Figure3). Figure 4 shows the map with the OOL line.

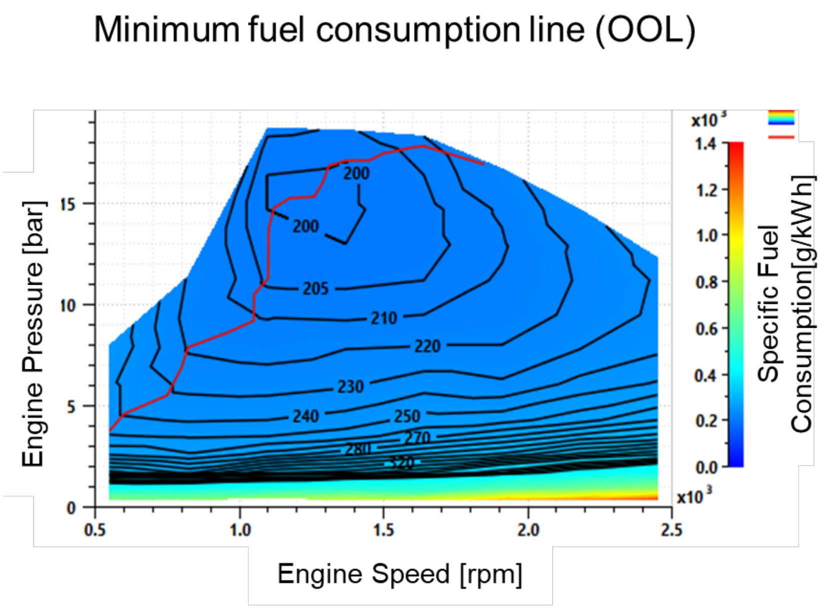
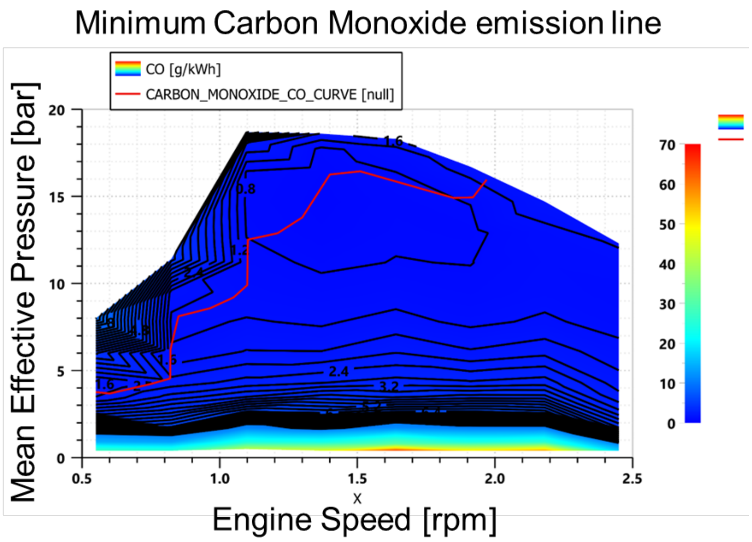
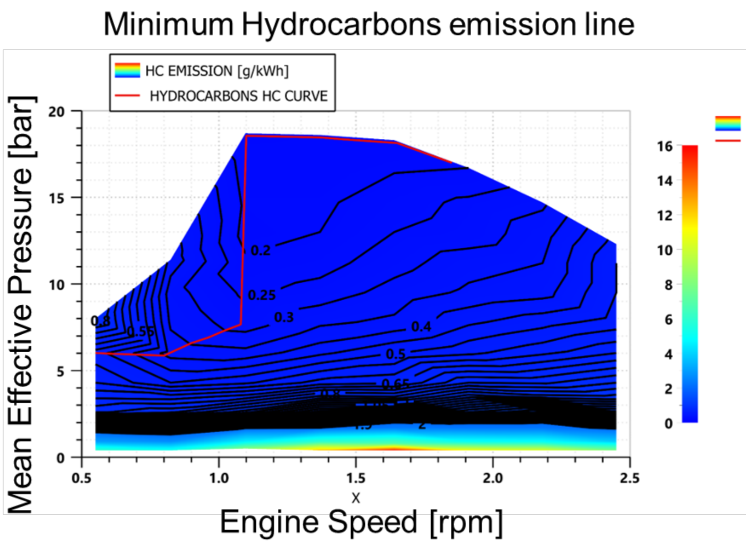


Figure 4. Engine map: iso-BSFC lines, and optimal operating line.

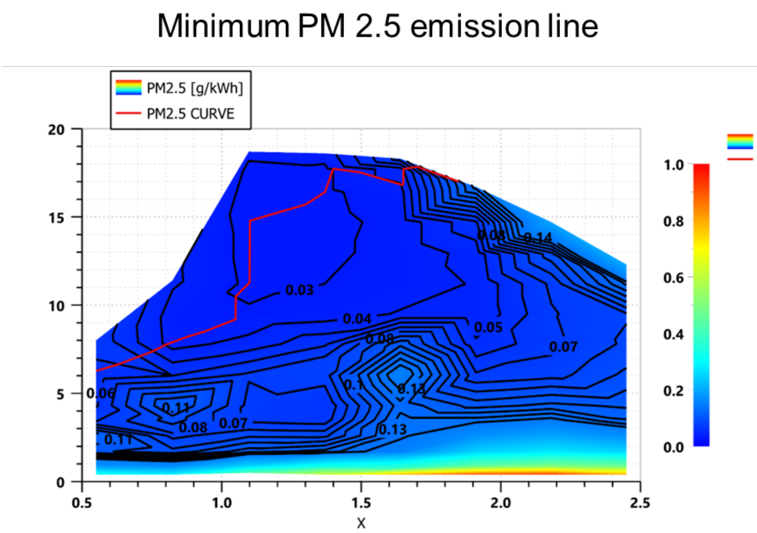
Figure 5 shows the maps of the four emission groups: NO, HC, PM and NOx. For each of them, the minimum emission line was obtained by crossing the isopower and iso-emission curves. The four isocurves are distant from each other, especially that of the minimum NOx, effectively preventing the formation of a single minimum line.



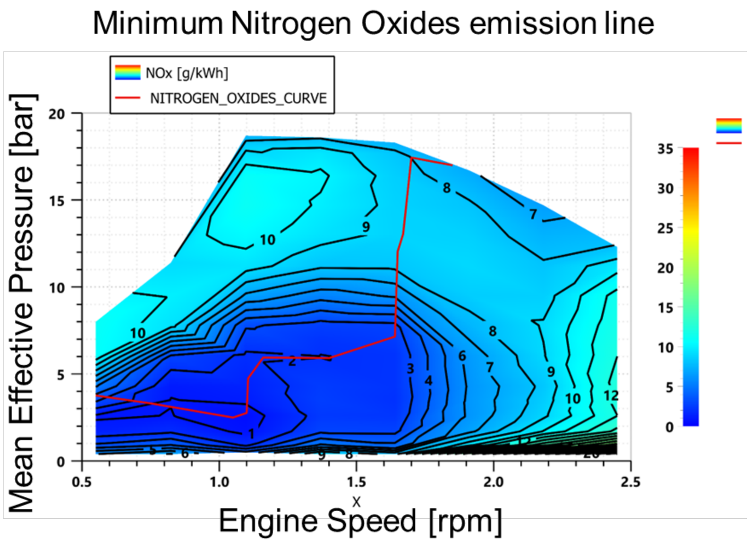
A)



B)



C)



D)

Figure 5. Emission maps and minimal emission lines: A) CO, B) HC, C) PM, D) NOx.

The total damage to human health is the sum of the damage caused by each emission calculated with the procedure illustrated in Section 3. Also in this case the line of minimum total damage (MTDL) can be identified by crossing the isopower curves with the isodamage curves (Figure 6). The two lines, OOL and MDL, are distant from each other and allow two different engine managements.

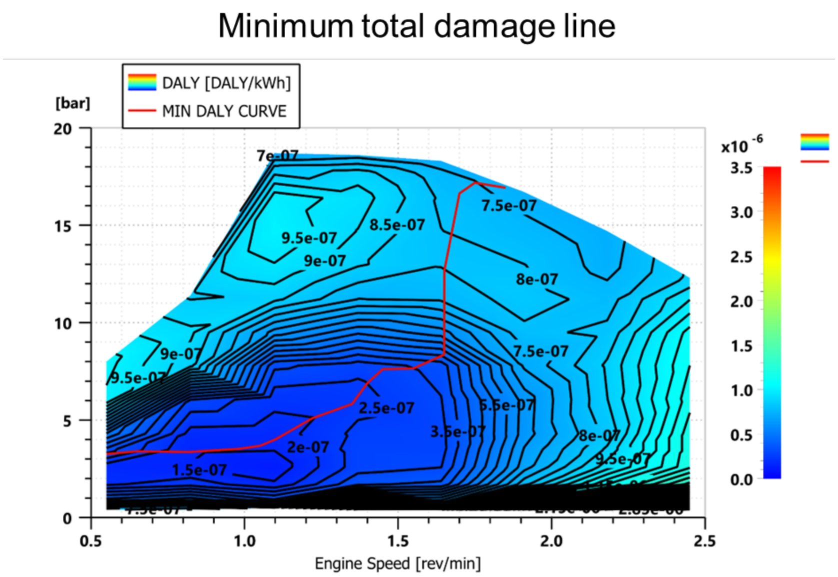


Figure 6. Total damage map and minimal total damage line (MTDL)

6.3. Hydraulic Pump and motor modelling

The accuracy of the models of the hydraulic units affects the reliability of global simulation results. Therefore the volumetric and hydromechanical losses were evaluated using scaling laws starting from values experimentally obtained from a reference machine, as suggested in [25, 30].

Linear scaling laws are based on the assumption that scaling units operate at the same maximum pressures and maximum speeds. Volumetric losses can be summarized by the flow rate flowing between two cylindrical surfaces, which depends on cube of the linear dimensions, as shown by the Poiseuille's law. Torque losses can be summarized by the torque generated by the friction force occurring between two flat surfaces in relative motion, which linearly depends on the linear dimension [30].

Therefore, the losses for a generic unit can be expressed as a function of the losses of the reference unit as follows:

$$Q_{loss} = Q_{loss\_ref} \cdot \lambda^3 \quad (3)$$

$$T_{loss} = T_{loss\_ref} \cdot \lambda^2 \quad (4)$$

where Q_{loss_ref} and T_{loss_ref} are the reference volumetric and hydromechanical losses expressed by polynomials as a function of pressure, speed and displacement fraction, while λ is the scale parameter.

$$\lambda = \sqrt[3]{\frac{V}{V_{ref}}} \quad (5)$$

6.4. Size and weight of transmission and accumulators

The transmission and accumulators weights were also evaluated in the transmission model. It is very difficult to predict the weight of a hydromechanical transmission since geometric, layout and mechanical resistance factors are involved in its definition. In this work a law was adopted that expresses the approximate weight of the transmission in this way:

$$m_{tr} \cong m_0 + (m_1 + m_2) \quad (6)$$

Where m_o = 600 kg is the mass of a commercial transmission of equal power [31] minus the weights of its two hydraulic units. m_1 and m_2 are the masses of the hydraulic units taken from the catalog of a well-known manufacturer. This transmission weight has been reduced by the powershift transmission weight supplied with the bus in question (Voith Diwa 6 model) equal to 350 kg. The final weight of the transmission also includes the weight of the oil, which varies between 20 and 35 kg. Both the displacements of the hydraulic units and the size of the accumulators refer to the catalog data offered by well-known manufacturers. Table 2 shows the displacement, maximum velocity and weight data for the pumps and motors; Table 3 shows the dimensions and weights of the accumulators.

In the vehicle model, a special control rejects cycles in which at least one of the two units runs at speeds higher than that recommended by the manufacturer.

Table 2. Pump and motor characteristics for 500 bar maximum operating pressure.

V_1 - cm^3	56	71	90	110	125	145	175	210	280
n_{1max} - rpm	3900	3600	3300	3250	3150	3000	2800	2650	2550
m_1 - kg	38	50	60	84	88	106	115	152	160
V_2 - cm^3	80	107	140	160	200	250	255	500	
n_{2max} - rpm	6150	5600	5150	4900	4600	3300	2650	2400	
m_2 - kg	36	46	61	62	78	100	170	210	

Table 3. Standard bladder accumulators - maximum pressure 400 bar.

V_0 - L	10	13	20	32	50	2x50	2x50+20
m_{acc} - kg	41	49	71	104	137	274	345

6.5. Mission profile

The vehicle model was subjected to the standard WLTP cycle for hybrid vehicles, limited to the low and medium segments of the conventional cycle for class 3 vehicles (first 1000 s in figure 7) [32]. These two segments show the characteristics of a city route with stops and starts and with speed peaks close to the maximum. Furthermore, sudden accelerations allow the engine to reach maximum power.

In the simulations the vehicle speed follows the mission profile with a maximum error of 1.8%.

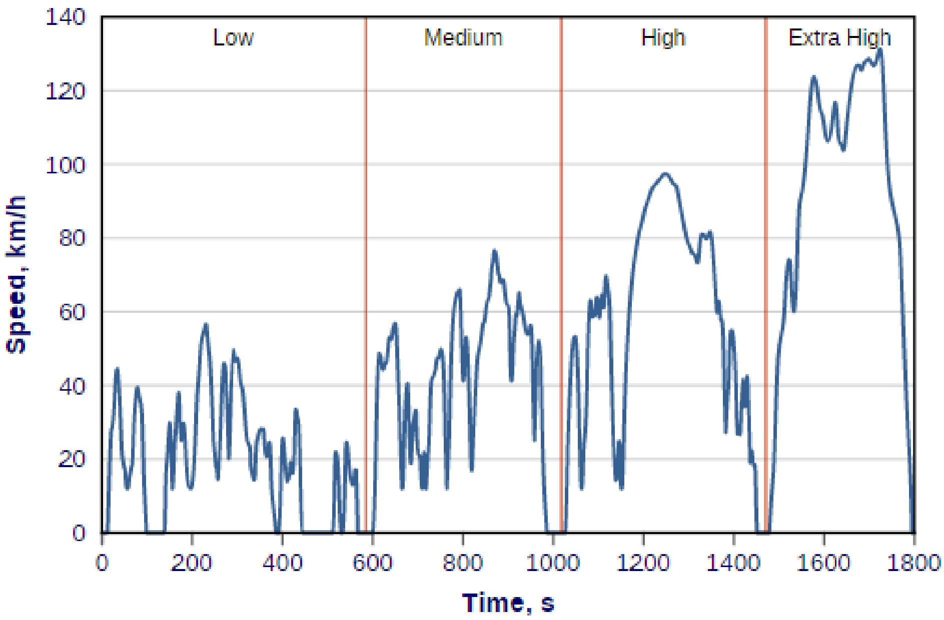


Figure 7. Mission profile based on WLTP cycle for class 3 vehicles.

7. SIMULATIONS AND OPTIMIZATIONS

As already mentioned, the bus models were used to size the transmissions OC and HOC as seen in Section 4. Two engine management criteria were considered. The first one requires the engine to follow the minimum consumption line (OOL, Figure 4), therefore in the following it will be indicated with MinC criterion; the second criterion requires the engine to follow the line of minimum total damage (MDL, Figure 6), hereinafter indicated as MinD criterion. The optimizations performed are summarized in Table 4.

Table 4. Optimizations carried out in this study.

	MinC	MinD
OC	x	
HOC (continuous variables)	x	x
HOC (discrete variables)	x	x

The design variables for each of the optimizations are as follows:

- OC transmission: τ_o, τ_1, τ_2, τ_d, V_1, V_2

- HOC transmission: τ_o, τ_1, τ_2, τ_d, V_1, V_2, p_0, V_0

The accumulator starts and ends the cycles at the same pressure, with a deviation of less than 1%. In this way, comparisons between different solutions are not distorted by a net contribution of energy due to the hydraulic source.

The search for the optimum was carried out using the genetic algorithm, a method particularly suitable for solving problems not well suited for standard optimization algorithms, including problems with discontinuous objective function, nondifferentiable, stochastic, or highly nonlinear.

The algorithm parameters were assumed based on literature recommended values. Population size was assumed equal to 30 and 40 respectively for the OC and HOC optimizations (recommended value 4.5* number of design variables); the reproduction rate was assumed 80% to assure a fast convergence limiting the risk of a local minima (recommended values 50- 85%); the number of generations was assumed equal to 30, with recommended value over 10.

The machine time to reach the solution was approximately 10 hours for each of the optimizations on a Windows I9 machine.

8. RESULTS AND DISCUSSIONS

Figure 8 shows the power contributions of the hydraulic source and the thermal source in a part of the work cycle. The first source only contributes to the start as long as the gas pressure in the accumulator allows it. After which only the thermal source will provide the required power. During the deceleration phase, the engine is idled and the kinetic energy of the vehicle is used to recharge the accumulator. During the energy sources shift, there is a power deficit which causes a small deceleration of the vehicle.

Figure 9 shows the evolution of the objective function, fuel consumption, and displacements during the optimization process of the OC transmission with MinC criterion.

Similar evolutions occurred for the HOC transmission.

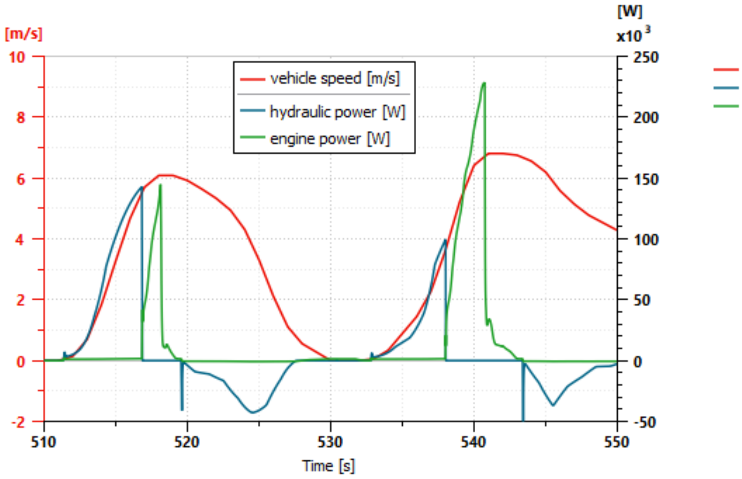


Figure 8. Hydraulic and engine power contributions during acceleration and braking phases.

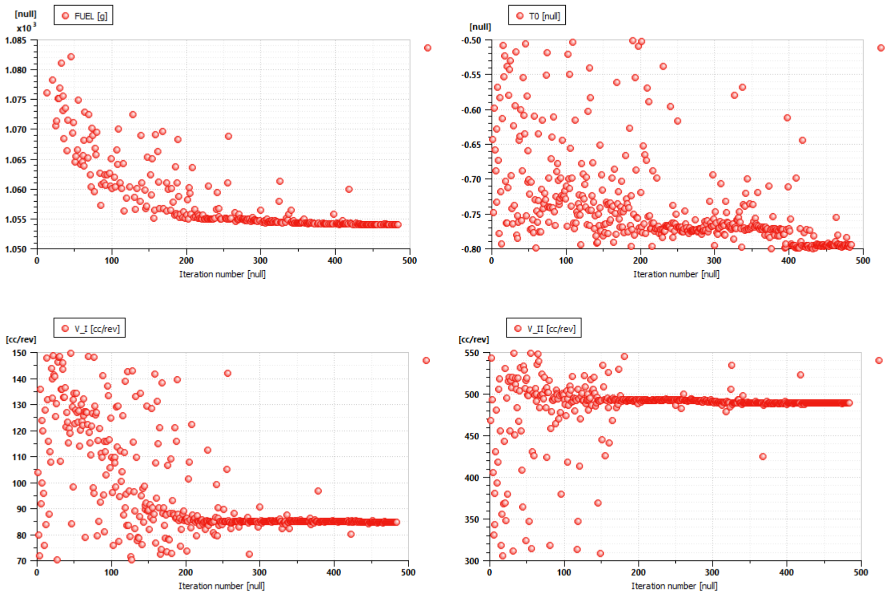


Figure 9. Evolution of the objective function (minimum fuel consumption) and the design variables during the optimization of OC transmission with MinC criterion.

8.1. Effect of the hybridization

Table 5 compares the optimization results of the OC and HOC transmissions both subjected to MinC criterion.

Table 5. Optimal design variables for OC and HOC transmissions (MinC)

	τ_0	V_1	V_2	p_0	V_0	τ_1	τ_2	τ_d	m_{fuel}	damage
	/	cm^3	cm^3	bar	L	/	/	/	g	DALY
OC	-0.849	84.7	489	/	/	1,12	0.84	8.2	1054	3.80E-6
HOC	-0.828	94.9	510	255	129.3	1.01	0.8	8.3	916	3.23E-6
Δ %									-13%	-15%

As can be seen, hybridization, i.e. the addition of the two accumulators, does not significantly change the optimal values of the design variables, only the displacement of the Unit 1 increases by approximately 10%, indicating that transmission and accumulators are practically independent functional groups.

Hybridization, however, increases the average transmission efficiency from 0.74 to 0.89, and leads to marked reductions in fuel consumption and damage, equal to 13% and 15% respectively. In a previous study [33], the authors found on a reach stacker a fuel consumption reduction due to hybridization of 21%.

The energy contribution of the hydraulic source was found to be equal to 24% of the total energy entering the transmission, while the additional weight due to the accumulators, approximately 750 kg, produces an increase in fuel consumption of 3%.

Note the high value of the accumulator pre-charge pressure. This is explained by the transmission management criterion adopted here, which provides for starting via the accumulator alone, without the contribution of the engine.

8.2. Effect of the management criterion on HOC transmission

Figure 10 shows the operating points in the engine map for a HOC simulation with the engine managed according to MinC criterion (red dots) and MinD criterion (black dots). The two trends are consistent with those of the OOL (Figure 4) and MDL (Figure 6), a sign that the control system acts correctly in forcing the engine towards those two conditions. In the medium-high speed area the two curves differ more from each other and therefore here the difference in damage caused is more marked.

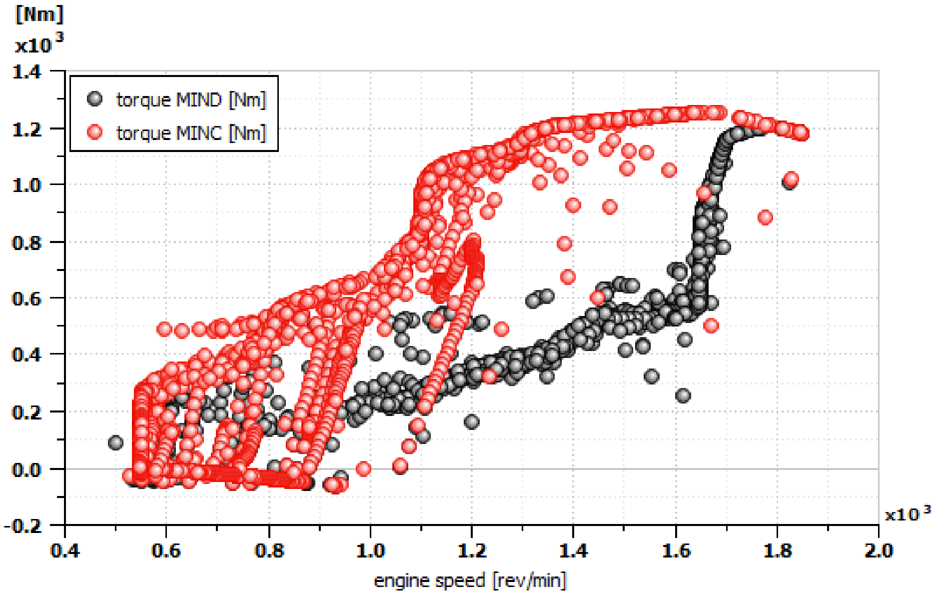


Figure 10. Operating points of the engine operated according to MinC criterion (red points) and MinD criterion (black points).

Table 6 shows the optimal parameters provided by the optimization procedure for the HOC transmission managed with MinC and MinD criteria.

Table 6. Optimal design variables for HOC transmission with the engine managed at MINC and MIND criteria.

	τ_0	V_1	V_2	p_0	V_0	τ_1	τ_2	τ_d	m_{fuel}	damage
	/	cm^3	cm^3	bar	L	/	/	/	g	DALY
MinC	-0.828	94.9	510	255	129.3	1.01	0.8	8.3	917	3.24E-6
MinD	-0.849	116	514	270	90	0.82	0.80	8.2	947	2.43E-6
Δ %									+3 %	-25%

The comparison shows that the management criterion moderately influences the optimal values of the design variables, only the Unit 1 displacement increases significantly. What is evident, however, is the difference in consumption and damage: the minimum damage criterion, compared to an increase in consumption of 3%, produces a reduction in the damage produced by emissions of approximately 25%.

8.3. Effect of component discretization

We now want to examine how and to what extent the choice of components from the catalogue, i.e. which vary in discrete steps, influences the optimal values of the design variables and the performance of the HOC transmission. Therefore the optimization processes were repeated considering the displacements and the volume of the accumulator varying according to the catalog values reported in Tables 2 and 3.

Table 7. Optimal design variables for HOC transmission with the engine managed at MinC and MinD conditions and discrete values for displacements and accumulator volumes.

	τ_0	V_1	V_2	p_0	V_0	τ_1	τ_2	τ_d	m_{fuel}	damage
	/	cm^3	cm^3	bar	L	/	/	/	g	DALY
MinC	-0.796	110	500	256	120	0.89	0.8	8.5	917	3.24E-6
MinD	-0.65	110	500	279	100	0.97	0.71	9.5	953	2.46E-6
Δ %									+4%	-24%

For both criteria, both the displacements and the volume of the accumulator settle on the values closest to those obtained in Table 6, without this causing significant deviations in consumption and damage. This concept is reiterated in Table 8, in the continuous-discrete comparison of the design variables.

Table 8. Continuous-discrete comparison for the management criteria MinC and MinD

	τ_0	V_1	V_2	p_0	V_0	τ_1	τ_2	τ_d	m_{fuel}	damage
	/	cm^3	cm^3	bar	L	/	/	/	g	DALY
MinC										
cont	-0.828	94.9	510	255	129.3	1.01	0.8	8.3	917	3.24E-6
disc	-0.796	110	500	256	120	0.89	0.8	8.5	917	3.24E-6
Δ %									0.0%	0.0%
MinD										
cont	-0.849	116	514	270	90	0.82	0.80	8.2	947	2.43E-6
disc	-0.65	110	500	279	100	0.97	0.71	9.5	953	2.46E-6
Δ %									+0.6%	-1.2%

From these comparisons it can be deduced that the usual practice in hydraulic design also applies in delicate cases like this one, i.e. proceeding with the theoretical calculation of the size and then adopting the closest catalog value.

9. CONCLUSIONS

In this work the hybrid hydromechanical transmission was investigated with particular reference to its sizing.

The sizing involved both the transmission and the engine in order to exploit the advantage afforded by the continuous transmission of engine management according to the most convenient criteria.

The criteria adopted here are the minimum fuel consumption criterion and the minimum emissions criterion. This second criterion was achieved by unifying emissions into a single parameter: the damage to human health.

Given the complexity of the transmission and the high number of variables, sizing was transformed into a mathematical programming problem.

The proposed sizing procedure was applied to a 12 m class city bus, of which a model including engine, transmission, control systems and load was prepared. The model was inserted into an optimization process that used the genetic algorithm as a search strategy for the optima.

For this vehicle, the study highlighted some points that can be summarized as follows:

- the hybridization of a powersplit transmission leads to a decrease in consumption of approximately 13% and a decrease in damage to human health by 15%.

-The optimal values of the transmission variables are not significantly affected by hybridization, indicating that the transmission and accumulators are practically independent functional groups.

-The management criterion of minimizing damage to human health, compared to an increase in consumption of 3%, reduces the damage caused by emissions by approximately 25%.

-The use of catalog components, i.e. with discretely variable dimensional values, does not lead to a substantial reduction in efficiency compared to the optimal solution obtained with continuous variation.

It is worth underlining that the results obtained are to be considered valid for this type of vehicle. For other vehicles, subject to different duty cycles, the results may be different.

10. NOMENCLATURE

cf	characterization factor of a substance	
	belonging to an impact category	$[kg_{ref}/kg]$
D	damage to human health	[DALY]
m	emission mass	[kg]
m_{acc}	accumulator mass	[kg]
m_{tr}	transmission mass	[kg]
n,ω	engine rotational speed	[RPM, rad/s]
P	power	[kW]
p_0	accumulator precharge pressure	[bar]
Q_{loss_ref}	experimental volumetric losses of a reference	
	hydraulic unit	[L/min]
Q_{loss}	volumetric losses of a generic hydraulic unit	[L/min]
sd	specific damage of a impact category	$[DALY/kg_{ref}]$
T	torque	[Nm]
T_{loss}	volumetric losses of a generic hydraulic unit	[L/min]
T_{loss_ref}	experimental hydromechanical losses of	
	a reference hydraulic unit	[L/min]
v	vehicle speed	[m/s]
V	maximum unit displacement of a hydraulic unit	$[cm^3]$
V_1	maximum unit displacement of Unit 1	$[cm^3]$
V_2	maximum unit displacement of Unit 2	$[cm^3]$
V_0	accumulator volume	[L]
λ	scaling parameter	[-]
τ_0	standing gear ratio of the planetary	[-]
	gear (see Figure 2)	[-]
τ_1	gear ratio of the Unit 1 gear (see Figure 2)	[-]
τ_2	gear ratio of the Unit 2 gear (see Figure 2)	[-]
τ_d	rear axle gear ratio (see Figure 2)	[-]

acronyms

BSFC	Brake-specific fuel consumption	[g/kWh]
MTDL	minimum total damage line	
MinC	minimum fuel consumption engine management criterion	
MinD	minimum health damage engine management criterion	
HOC	Hydraulic hybrid Output Coupled	
OC	Output Coupled	
OOL	Optimal operating line in the engine map	

Funding: This research was funded by the University of Padua through the DOR 2023 fund.

11. References

[1] Wang J., Yang Z., Liu S., Zhang Q. and Han Y., "A comprehensive overview of hybrid construction machinery". Advances in Mechanical Engineering 2016, Vol. 8(3) 1–15) 2016. DOI: 10.1177/1687814016636809

[2] Chen, Q.; Lin, T.; Ren, H.; Fu, S. "Novel potential energy regeneration systems for hybrid hydraulic excavators". Math. Comput. Simul. **2019**, 163, 130–145.

[3] R. Ramakrishnan, Somashekhar S. Hiremath and M. Singaperumal, "Theoretical investigations on the effect of system parameters in series hydraulic hybrid system with hydrostatic regenerative braking". Journal of Mechanical Science and Technology 26 (5) (2012) 1321~1331- DOI 10.1007/s12206-012-0321-y

[4] Ramdan M I and Stelson K A 2016 "Optimal design of a power-split hybrid hydraulic bus" Proc IMechE Part D: J Automobile Engineering, Vol. 230(12) 1699–1718. DOI: 10.1177/0954407015621817

[5] Prado Barbosa T P, Rodrigues da Silva L A, Pacheco Pujatti F J, Horta Gutierrez J C 2022 "Hydraulic hybrid passenger vehicle: Fuel savings possibilities" Mechanics Based Design of Structures and Machines An International Journal Volume 50, 2022 - Issue 1 https://doi.org/10.1080/15397734.2020.1714447

[6] Bender, F.A.; Bosse, T.; Sawodny, O. "An investigation on the fuel savings potential of hybrid hydraulic refuse collection vehicles". Waste Manag. **2014**, 34, 1577–1583.

[7] Wen Q, Wang F, Xu B and Sun Z "Improving the Fuel Efficiency of Compact Wheel Loader with a Series Hydraulic Hybrid Powertrain" IEEE Transactions On Vehicular Technology, Vol. 69, No. 10, October 2020.

[8] Karl Pettersson, Kim Heybroek, Per Mattsson & Petter Krus (2017) "A novel hydromechanical hybrid motion system for construction machines", International Journal of Fluid Power, 18:1, 17-28, DOI: 10.1080/14399776.2016.1210423

[9] Renius KT and Resch R 2005 "Continuously variable tractor transmissions" ASAE Disting. Lect. 29 (2005).

[10] Ivantysynova M M, Carl, B A and Williams, K R 2008 "Power split transmission with energy recovery" US Patent 20080081724

[11] Van de Ven J D, Olson M W, and Li P Y 2008 "Development of a Hydro-Mechanical Hydraulic Hybrid Drive Train with Independent Wheel Torque Control for an Urban Passenger Vehicle" Proc. of International Fluid Power Exhibition, Las Vegas, Nevada, United States.

[12] Kumar R and Ivantysynova M 2010 “Investigation of various power management strategies for a class of hydraulic hybrid powertrains: Theory and experiments” Proc. of the 6th FPNI PhD Symposium, p. 87–99.

[13] Kwon, H.; Ivantysynova, M. “System Characteristics Analysis for Energy Management of Power-Split Hydraulic Hybrids”. *Energies* **2020**, *13*, 1837. https://doi.org/10.3390/en13071837

[14] Hu, X.; Zhang, X.; Tang, X.; Lin, X. “Model predictive control of hybrid electric vehicles for fuel economy, emission reductions, and inter-vehicle safety in car-following scenarios”. Energy 2020, 196, 117101.

[15] Tang, X.; Chen, J.; Liu, T.; Qin, Y.; Cao, D. “Distributed Deep Reinforcement Learning-Based Energy and Emission management Strategy for Hybrid Electric vehicles”. IEEE Trans. Veh. Technol. 2021, 70, 9922–9934.

[16] R. A. Dollar, L. Thibault, M. Laraki and A. Sciarretta, “Eco-Driving of Compression-Ignition Vehicles to Minimize Nitrogen Oxide Emissions” in *IEEE Transactions on Control Systems Technology*, vol. 30, no. 5, pp. 2084-2099, Sept. 2022, doi: 10.1109/TCST.2021.3133867..

[17] Johri, R.; Filipi, Z. “Optimal energy management of a series hybrid vehicle with combined fuel economy and low-emission objectives”. Proc. IMechE Part D J. Automob. Eng. 2014, 228, 1424–1439.

[18] Masih-Tehrani, M.; Ebrahimi-Nejad, S.; Dahmardeh, M. “Combined fuel consumption and emission optimization model for heavy construction equipment”. Autom. Constr. 2020, 110, 103007.

[19] Rossetti, A.; Andretta, N.; Macor, A. “On the Use of the Disability-Adjusted Life Year (DALY) Estimator as a Metric to Optimally Manage ICE Emissions”. Energies 2022, 15, 4386. https://doi.org/10.3390/ en15124386.

[20] R. Pfiffner*, L. Guzzella, C.H. Onder. “Fuel-optimal control of CVT powertrains. Control Engineering Practice” 11 (2003) 329–336.

[21] Katharina Baer, Liselott Ericson & Petter Krus “Framework for simulation-based simultaneous system optimization for a series hydraulic hybrid vehicle”, International Journal of Fluid Power, DOI: 10.1080/14399776.2018.1527122

[22] Uebel K, Raduenz H, Krus P and de Negri V J 2018 “Design optimisation strategies for a hydraulic hybrid wheel loader” Proceedings of the BATH/ASME Symposium on Fluid Power and Motion Control FPMC2018 September 12-14, 2018, Bath, UK

[23] S. Ahn, J. Choi, S. Kim, J. Lee, C. Choi and H. Kim. “Development of an integrated engine-hydro-mechanical transmission control algorithm for a tractor” Advances in Mechanical Engineering 2015, Vol. 7(7) 1–18 DOI: 10.1177/1687814015593870

[24] Li C T and Peng H 2010 “Optimal Configuration Design for Hydraulic Split Hybrid Vehicles” 2010 American Control Conference Marriott Waterfront, Baltimore, MD, USA June 30-July 02, 2010 pag. 5812-5217.

[25] Andretta N 2022 “Design methods for hybrid hydro-mechanical transmissions for heavy-duty vehicles”. Ph.D. Dissertation. Department of Management and Engineering, course in Mechatronics and Product Innovation Engineering. University of Padua 2022.

[26] N Andretta *et al* 2022 *J. Phys.: Conf. Ser.* **2385** 012031 **DOI** 10.1088/1742-6596/2385/1/012031

[27] Blake C, Ivantysynova M, Williams K. “Comparison of Operational Characteristics in Power Split Continuously Variable Transmissions”. SAE 2006 Commercial Vehicle Engineering Congress & Exhibition, October 2006, *SAE Techical Paper series 2006-01-3468*

[28] Jolliet, O., Margni, M., Charles, R., Humbert, S., Payet, J., Rebitzer, G., et al. IMPACT 2002+: A New Life Cycle Impact Assessment Methodology. 2003. doi:10.1007/BF02978505

[29] *Amesim*; 2021-2022; Siemens Industry Software NV: Plano, TX, USA, 2021.

[30] L. Shang, M. Ivantysynova “An Investigation of Design Parameters Influencing the Fluid Film Behavior in Scaled Cylinder Block/Valve Plate Interface” FPNI2016-1540, V001T01A027. https://doi.org/10.1115/FPNI2016-1540

[31] Mercati S, Panizzolo F, Profumo G 2016 “Power split Hydro-mechanical Variable Transmission (HVT) for off-highway application”, 10th International Fluid Power Conference (10. IFK) March 8 - 10, 2016, Vol. 3, pp. 69-80.

[32] https://dieselnet.com/standards/cycles/index.php

[33] Andretta N, Rossetti A and Macor A 2019 “Optimal design of a hybrid hydromechanical transmission for a reach stacker” AIP Conference Proceedings 2191, 020010 (2019); https://doi.org/10.1063/1.5138743.