Energy Efficiency Analysis and Comparison of Transmission Technologies for an Electric Vehicle

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Abstract— **Electric vehicles are seen as the most promising solution to convert sustainable energy into drive energy. However, there are still some major (technological) challenges, e.g., in terms of maximizing range anxiety, minimizing battery costs and charging time. A possible solution, in order to improve the relative limited range (100-200 km), battery life time and ultimate costs, is utilization, e.g., of a transmission technology between the electric machine and driven wheels. The main research question of this paper is, what are the effects of transmission types (fixed, manual, continuously variable) and parameters (final drive ratio, efficiency) on the overall (battery-to-wheel) efficiency and performance of an electric vehicle?**

A backwards-simulation model from the wheels (load) to the batteries (source) has been developed to simulate an electric vehicle on a representative drive cycle (NEDC, FTP75). The model incorporates rotating (drive train) inertias, different transmission types, regenerative braking, battery efficiency and a shift strategy optimization algorithm based on Dynamic Programming (DP).

The results show that with an optimized shifting strategy compared to a prescribed strategy using a Manual Transmission (MT) a maximum energy saving of 6% is possible. The conventional push-belt Continuously Variable Transmission (CVT) is less energy efficient and needs more battery energy for driving compared to a fixed-gear transmission type. Mainly, caused by the typically lower CVT efficiency. Further, it was found that reducing the final drive ratio improves the overall efficiency. The range in which the electric motor operates was shifted to higher angular velocity resulting in higher efficiency operating points. If the transmission efficiency of all the transmission types are equal assumed, then the CVT is typically the most energy efficient transmission type. For example, an energy saving of 7% is calculated compared to a well-selected fixed-gear ratio transmission type.

I. Introduction

In recent years, due to increase in fossil fuel prices, more severe increasing legislation regarding reduction of green houses gases and the transition to more sustainable energy sources, Electric Vehicles (EVs) are seen as promising solutions and being introduced more rapidly into the market [1], [2], [3], [4]. The drive train technology of EVs is relatively new, costly and leaves room for optimization from different perspectives like safety, comfort and efficiency (topology, technology selection and component sizes). In this paper, the Volkswagen Lupo 3L is selected as baseline vehicle and assumed to be converted to an EV. The Volkswagen Lupo 3L (see, Fig. 1) was chosen, because this particular vehicle is seen as a well optimized compact car with high energy conversion efficiency from tank-to-wheel. Several of these optimizations for the conventional vehicle include usage of light-weight materials, improved aerodynamics and low resistance tires.

Fig. 1: Volkswagen Lupo 3L (Courtesy of Volkswagen)

A. Problem Definition

The main problems of EVs are their relative limited range (100-250 km), battery pack costs (10-20 kEuro), and the relative large charging time of the battery pack (4h-8h). A transmission may address two of these problems. By using a transmission an EV can work at more efficient operation points and draw less power from the battery [5]. This can result in an increased range of the EV, or smaller battery pack. In literature, very few articles can be found studying the effect of a transmission for the performance of an EV. In [6], the authors investigated a two-speed transmission for an EV and concluded that a two-speed transmission improves the efficiency of an EV. However, the effect of changing the transmission parameters and types is still unknown. The goal of this study is to give insight on this topic. Thereby, the underlying main question of the study is: what are the effects of transmission parameters on the efficiency and performance of an electric motor driven vehicle?

In order to answer this question, a numerical-simulation model has been developed. The model includes electric machine, battery efficiency data, parameters for the rotating iner-

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tias of the main drive train components, different transmission types, regenerative braking, and a shift strategy optimization algorithm. The main question has been subdivided into four questions:

1) *What are the effects of **shifting** on the energy efficiency (from battery-to-wheel)?*
   Thereby, three shifting strategies will be researched: (i) no shifting: the transmission will not shift and it will drive a cycle using one (fixed) gear ratio; (ii) pre-scribed (standard) shifting strategy: driving cycles like the New European Drive Cycle (NEDC) also include a standard shift schedule, besides data concerning the velocity and acceleration over time. This strategy represents the shift strategy commonly used by the average driver; (iii) optimal shift strategy: the gear strategy is optimized resulting in the minimum battery energy usage.
2) *What is the influence of a different **transmission technology**, e.g., Manual Transmission (MT) and Continuously Variable Transmission (CVT) on the energy efficiency?*
   Each technology has its advantages and disadvantages. For example, a CVT has an unlimited amount of gear ratios between its upper and lower ratio, yet typically a relative lower efficiency than a MT. A MT has a finite set of gear ratios and additional clutch losses during shifting, yet typically a higher efficiency than a CVT.
3) *What is the effect of the **final drive ratio** on the energy efficiency?*
   The transmission gear ratios for the MT and the CVT are predefined and thus the maximum vehicle velocity is fixed for a given gear ratio. By changing the final drive ratio, the maximum obtainable vehicle velocity is changed for all gear ratios. Moreover, at a given vehicle velocity and gear ratio, a lower final drive ratio allows the Electric Machine (EM) to operate at higher angular velocities, and vice versa for a higher final drive ratio.
4) *What is the effect of the **transmission efficiency** on the energy efficiency?*
   In the simulation model, the different transmission types have a different efficiency. By changing the transmission efficiency insights will be given in the relationship between the transmission type and its efficiency.

B. Outline the paper

The paper is outlined as follows: the simulation model will be discussed in Section II. It starts with the (electric) vehicle modeling. After the vehicle modeling the drive train components (battery, inverter, electric machine, clutch, transmission) will be discussed in Section III. Accordingly, shift optimization problem is discussed in Section IV. In Section V, the simulation results will be presented and analyzed. Finally, the conclusions are described in Section VI.

II. Electric Vehicle Model

The simulation vehicle will be based on the Volkswagen (VW) Lupo 3L (see, parameters as listed in Table I). Many of the specifications are taken from the data sheet specifications, yet the parameters related to the center of gravity, center of drag and moments of inertia are estimated.

A. Vehicle resistances

The stationary resistance forces, denoted as F_R, acting on the vehicle are calculated as the sum of the air, rolling and gradient resistance force, respectively:

$$F_R = \frac{1}{2}\rho_{air} A_f C_d V^2 + C_r m_{tot} g \cos(\alpha) + m_{tot} g \sin(\alpha) \quad (1)$$

In Table I, a description of the used symbols and values in this equation can be found.

TABLE I: Volkswagen Lupo 3L properties

Vehicle base mass, m_{car}	(kg)	830
Frontal area, A_{f}	(m^2)	2.06
Height centre of gravity, h_{g}	(m)	0.55
Height centre of drag, h_{d}	(m)	0.8 h_{g}
Wheelbase, L	(m)	2.319
Distance front to CG, L_{a}	(m)	0.4 L
Distance rear to CG, L_{b}	(m)	0.6 L
Wheel radius, R_{w}	(m)	0.27855
Friction coefficient, C_{r}	(-)	0.006
Drag coefficient, C_{d}	(-)	0.29
Moment of inertia wheels, J_{w}	($\text{kg} \cdot \text{m}^2$)	0.3880
Moment of inertia half-shaft, J_{axle}	($\text{kg} \cdot \text{m}^2$)	0.0144

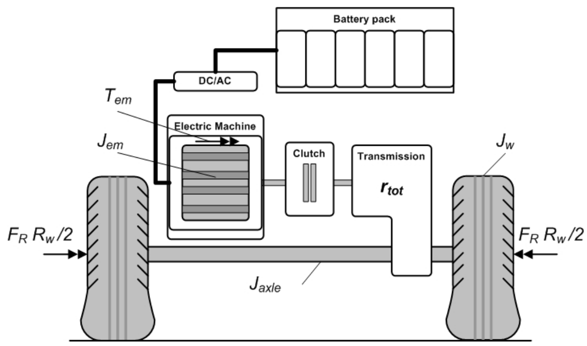


Fig. 2: Electric drive train

The total vehicle mass, denoted as $m_{tot} = m_{car} + m_{bat} = 830 + 300 = 1130$ [kg], consists of the chassis, wheels, body and other parts including the battery pack and is kept constant. According to Newton's Law, the difference between the tractive force at the wheels produced by the electric machine F_D and resistances forces F_R acting on the vehicle, will result in a vehicle acceleration, denoted as $\dot{V}$, with,

$$m_{tot} \cdot \dot{V} = F_D - F_R. \quad (2)$$

In Fig. 2, the electric drive train is schematically shown. The wheel acceleration is simply the velocity of the vehicle divided by the wheel radius, $\omega_w = V/R_w$. The total speed ratio is a multiplication of the gear ratio (r_{trm}) and the final drive ratio (r_{fd}),

$$r_{tot} = (r_{trm} \cdot r_{fd} \mid r_{trm} \in \{r_1, r_2, ..., r_n\} \in \mathbb{R}_0^+ \wedge r_{fd} \in \mathbb{R}_0^+), \tag{3}$$

and is defined as $r_{tot} = \omega_{out}/\omega_{in}$. The acceleration of the rotating parts of the EM is a function of the wheel acceleration, EM velocity, the speed ratio change (i.e., in case of a CVT), and the ratio, or:

$$\dot{\omega}_{em} = (\dot{\omega}_w - \omega_{em}\dot{r}_{tot})/r_{tot}. \tag{4}$$

The moment of inertia of the electric machine J_{em}, drive axles J_{axle} and wheels J_w are considered in the model (see, also [7]). The clutch torque (i.e., in case of a MT required for gear synchronization) is calculated as follows,

$$T_{clutch} = T_{em} - J_{em}\dot{\omega}_{em}, \tag{5}$$

with T_{em} the mechanical output torque of the electric machine. The transmission ratio amplifies the torque output of the transmission, yet the transmission efficiency η_{trm} reduces the net transmission output torque,

$$T_{trm} = T_{clutch} \cdot \eta_{trm}/r_{trm}. \tag{6}$$

The wheel torque after the final drive with a certain ratio r_{fd} and efficiency η_{fd} becomes,

$$T_D = F_D R_w + 2J_{axle}\dot{\omega}_w + 4J_w\dot{\omega}_w = T_{trm} \cdot \eta_{fd}/r_{fd}, \tag{7}$$

used to drive the vehicle and rotating masses of the axle/drive shaft[1] and wheels. In the rest of the paper, the combination of the transmission and final drive efficiency is denoted as $\eta_{trmfd} = \eta_{trm} \cdot \eta_{fd}$. The equations (5) to (7) can be combined to give the tractive force at the wheels,

$$F_D = \frac{1}{R_w}\left(T_{trm}\frac{\eta_{fd}}{r_{fd}} - 2J_{axle}\dot{\omega}_w - 4J_w\dot{\omega}_w\right). \tag{8}$$

The inertia forces is due to the moment of inertias of the EM, axles and the wheels holds,

$$F_{inrt} = \frac{1}{R_w}\left(\frac{J_{em}\dot{\omega}_{em}\eta_{trmfd}}{r_{tot}} + 2J_{axle}\dot{\omega}_w + 4J_w\dot{\omega}_w\right). \tag{9}$$

The torque required from the EM can be obtained after substitution of Eq. (5), Eq. (6), Eq. (8), and Eq. (9) in Eq. (2), and can be expressed as,

$$T_{em} = \frac{R_w r_{tot}}{\eta_{trmfd}}\left(F_R + m_{tot}\dot{V} + F_{inrt}\right). \tag{10}$$

B. Driving cycle

The driving cycles, which will be used as simulation input are the standard European drive cycle NEDC and the US drive cycle FTP75. The time step is $h = 1$ [s] assumed. The adhesive coefficient between the tire and road is 0.8 [-], and the slope of the road is $0°$ assumed.

[1]A front wheel drive vehicle has two drive shafts, each one connects one wheel to the differential

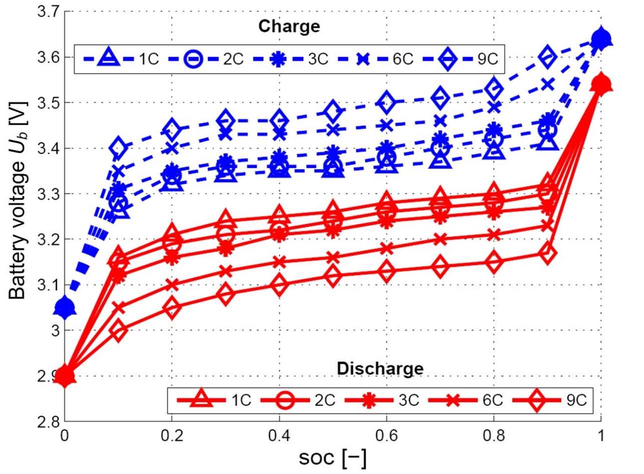


Fig. 3: Battery parameters: battery voltage of a 3.3 Volt $LiFePO_4$ for different C-rates.

III. Component models

The quasistatic efficiency component models for battery, inverter, electric machine and transmission technologies will be discussed next. Accordingly, the optimization problem will be described.

A. Battery

For the battery technology of the vehicle is a lithium polymer battery, i.e., $LiFePO_4$ assumed [8]. The battery voltage, denoted as U_b, of the 3.3 [V] $LiFePO_4$ is shown in Fig. 3 as a function of the State of Charge (SOC) and different charging rates (expressed in C-rate). Table II shows the other parameters used for the battery. The voltage drop is caused by the internal resistance of the battery R during charging (+), or discharging (-) with current I, respectively. The battery voltage is calculated as the difference between the open circuit voltage U_{oc} and the voltage drop,

TABLE II: Battery properties $LiFePO_4$

Battery mass, m_{bat}	(kg)	300
Specific energy	(Wh/kg)	118
Battery cell capacity, Q_0	(Ah)	4.4
Battery cell voltage, U_{bp}	(U)	3.3
Max. discharge current cell, $I_{max,d}$	(A)	70
Max. charge current cell, $I_{max,c}$	(A)	10
No. battery cells in parallel, n_p	(-)	31
No. battery cells in series, n_s	(-)	79
Max. battery pack voltage	(V)	260
Total battery capacity	(kWh)	35.4
State of charge at $t = 0$	(-)	0.5

$$U_b = U_{oc} \pm R(SOC) \cdot |I|. \tag{11}$$

This resistance is a function of the State of Charge (SOC) (temperature and Peukert effects are for simplicity neglected).

The SOC indicates the amount of electric charge in the battery relatively to the nominal battery capacity. The State of Charge, SOC is calculated as follows,

$$SOC(t) = \frac{Q(t)}{Q_0 \cdot n_p}, \text{ with } \dot{Q}(t) = I(t). \tag{12}$$

The maximum (dis-)charged current I_{max} allowed is limited and depends on the battery specifications, $|I| \leq |I_{max}|$. The amount of required batteries in series (n_s) and parallel (n_p) are found by satisfying the maximum motor controller input power. The battery output power can be expressed as,

$$P_b = U_b \cdot I = (U_b \cdot U_{oc} - U_b^2)/R. \tag{13}$$

In practice the maximum output power is lower due to the minimum (or, maximum in case of charging) operation voltages of the motor controller $U_{em,min}$, the battery $U_{b,min}$, or the open circuit voltage divided by two $U_{oc}/2$ [9], and holds,

$$U_{b,min} = \max(U_{em,min},\ U_{b,cell,min},\ U_{oc,cell}/2) \tag{14}$$

B. Inverter

The efficiency of the inverter is considered to be a constant, $\eta_{inv} = 0.95$. The inverter output power during (dis-)charging becomes,

$$P_{inv} = \min(P_b \eta_{inv}, P_b/\eta_{inv}). \tag{15}$$

C. Electric machine

The EM is based on the Siemens 1PV5105WS12. The main properties of the EM are listed in Table III. The electric machine output power depends on its efficiency, which is modeled as quasistatic efficiency map depending on the output speed and torque,

TABLE III: EM properties

Maximum torque @ 0-3000 RPM	(Nm)	115
Maximum power @ 3000-4000 RPM	(kW)	36
Maximum rotor speed	(RPM)	$10 \cdot 10^3$
Rated voltage, U_{em}	(U)	130
Moment of inertia EM, J_{em}	(kg $\cdot$ m^2)	0.0113

$$P_{em} = \min(P_{inv} \cdot \eta_{em}(\omega_{em}, T_{em}), P_{inv}/\eta_{em}(\omega_{em}, T_{em})). \tag{16}$$

D. Clutch model

The purpose of a clutch is to decouple the EM from its load when shifting gears with the manual transmission (MT). Due to the difference in angular velocity of the EM and the input shaft of the transmission, the clutch slips until the two angular velocities match, or

$$|r_{tot} \cdot \omega_{em} - \omega_w| = 0. \tag{17}$$

While slipping, energy is dissipated as heat when shifting up and during downshifts extra energy has to be produced to accelerate the EM. In both cases the clutch provides a loss in power P_{clutch} and this loss should be added to P_{em} when accelerating the vehicle and subtracted from the total power available, e.g., for regenerative braking,

$$P_{clutch} = \left| \left(R_w \cdot F_D + 2J_{axle}\dot{\omega}_w + 4J_w\dot{\omega}_w \right) \frac{r_{tot}(\omega_{em} - \frac{\omega_w}{r_{tot}})}{\eta_{trmfd}} \right|. \tag{18}$$

E. Transmission model

For simplicity, and comparison reasons, the transmission discussed below are assumed to have a constant efficiency (see, Table IV). For the pushbelt CVT initially an average constant value is used from [10]. For the fixed-gear ratio a constant efficiency of 0.95 is assumed (without the final drive efficiency).

TABLE IV: Transmission efficiencies

Final drive efficiency, η_{fd}	(-)	0.95
MT efficiency, η_{trm}	(-)	0.95
CVT efficiency, η_{trm}	(-)	0.85

1) Manual Transmission (MT): The ratios of the manual transmission MT are based on the standard 5-spd. transmission for the Volkswagen Lupo 3L. The gear ratios are multiplied by a factor of two in order to avoid over speeding the electric machine, or $r_{trm} = \{0.14, 0.26, 0.40, 0.54, 0.67\}$. The final drive is $r_{fd} = 0.30$.

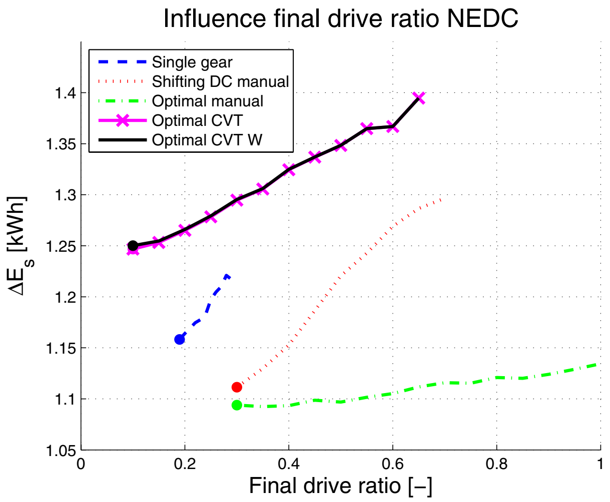
2) Continuously Variable Transmission (CVT): The underdrive and overdrive ratios of the CVT are based on values for a conventional CVT, i.e., 0.40 and 2.40 respectively. The final drive for the CVT is initially selected to be $r_{fd} = 0.095$, which results in the same input torque for any given output torque during overdrive. Moreover, the maximum ratio speed change of the CVT is limited, i.e., $|\dot{r}_{trm}| \leq 1$ is assumed.

IV. Optimization problem

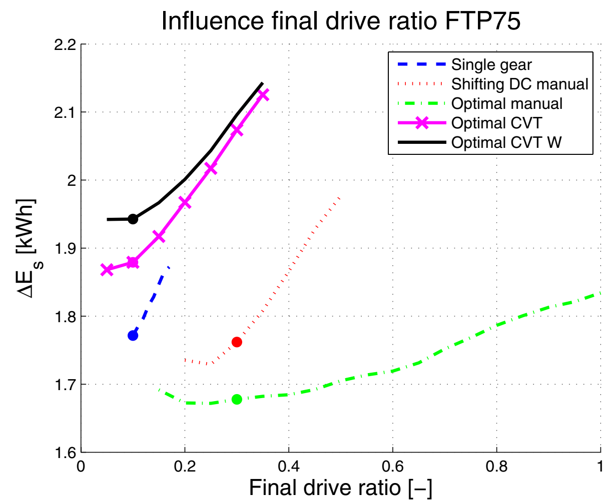
The control design objective for the EV equipped with a MT or CVT is minimization of the battery input power P_s over the whole Drive Cycle (DC) with final time t_f, subject to the different defined (component) constraints (as discussed in the previous sections). The optimization problem becomes,

$$\Delta E_s = \min_{r_{trm}(t)} \int_0^{t_f} \left(P_s(r_{trm}(t), t) \mid V(t), F_D(t)\right) \cdot dt, \tag{19}$$

given the vehicle velocity (cycle) and required traction force at the wheels. Moreover, no shifting is allowed at stationary driving for both transmissions. Shifting with the MT is virtually penalized by the clutch losses. Thereto, shifting with the CVT is penalized by introducing additional energy costs during shifting using an equivalent battery-energy-to-shifting conversion factor [J/-] in order to make a more fair comparison between the two technologies. This factor is for the NEDC and FTP75, 1000 and 2700 respectively. The problem is solved using Dynamic Programming (DP) [11].



(a) NEDC



(b) FTP75

Fig. 4: Final drive ratio

V. Simulation Results

The effect of changing the transmission parameters on the efficiency of the EV will be discussed below.

A. Sensitivity analysis for the final drive ratio

In the Figures 4a and 4b, the battery input energy ΔE_s is computed for both drive cycles. Thereby, the transmission technology is varied. The influence of the equivalent battery-energy-to-shifting conversion factor is also indicated with the character 'W'. The description 'optimal' refers to the optimized shifting strategy. It can be observed that a lower final drive ratio leads to a lower ΔE_s. A higher average angular speed for the EM results in a higher average EM efficiency. Further, for the FTP75, the effects of penalizing shifting of the CVT are significantly larger on the energy consumption. Whereas, it has virtually no effect simulated on the NEDC due to the more mild cycle. Table V lists the optimal final drive ratios for the different transmissions, which are used in the rest of the paper.

TABLE V: Optimized final drive ratios

Single gear NEDC	0.19
Single gear FTP75	0.10
Manual transmission	0.30
CVT	0.10

B. Sensitivity analysis for the shift strategies

In the Table VI, the total battery input energy for both cycles is shown. It can be observed that for both cycles, the minimum energy consumption is realized with the MT. Further, for both cycles regenerative braking can be fully utilized at the front wheels due to the relative low maximum deceleration values. In addition, it was found that shifting during propulsion and braking is profitable in both situations. The optimized shifting strategy tends to shift to lower gear ratios simultaneously increasing the electric machine speed and its efficiency.

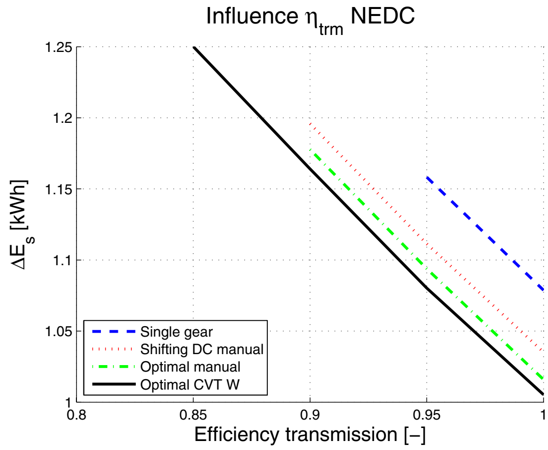
TABLE VI: Total energy NEDC/FTP75

Shift strategy	$\mathbf{\Delta E_s}$**[kWh]**	**Relative** ΔE_s	η_{trm}
Single gear	1.16/1.77	100/100	0.95
Shifting DC manual	1.11/1.76	96/99	0.95
Optimal manual	1.09/1.68	94/95	0.95
Optimal CVT W	1.25/1.94	108/110	0.85

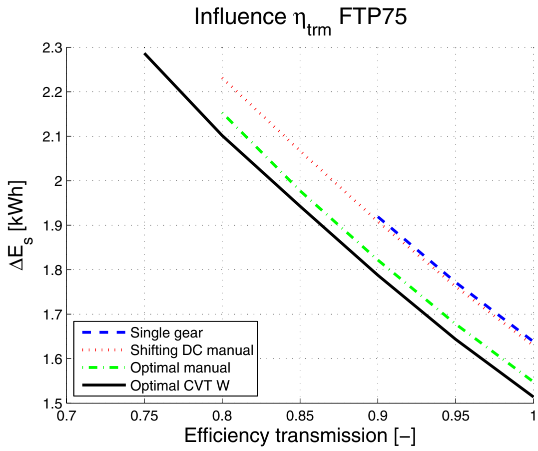
Although, with the CVT in principle the EM can be operated at higher efficiency points, the energy savings are reduced by the higher losses of the CVT compared to the other technologies. In comparison with the Lupo 3L equipped with a small diesel engine, the Lupo simulated as an EV is, as expected, much more efficient. For the optimal manual case on the NEDC (assuming 10.4 kWh/L for diesel), the equivalent fuel consumption is only 100 [km] * 1.09 [kWh] / 10.4 [kWh/L] / 11.8 [km] = 0.9 [L/100km] due to the higher drive train efficiency.

C. Sensitivity analysis for the transmission efficiency

In the Figures 5a and 5b, the battery input energy is computed for both drive cycles as a function of the transmission efficiency. If the CVT efficiency is increased from 85% to 90% with 5%, then the same energy consumption ΔE_s is used compared to the single gear with $\eta_{trm} = 0.95$. An even higher increase in CVT efficiency, i.e., of 8% from 85% to 93% is needed to compete with an optimal operated MT. Table VII lists the normalized energy at $\eta_{trm} = 0.95$, and Table VIII shows the linear approximated energy-to-efficiency sensitivity for the transmission efficiencies between $\eta_{trm} = 0.95$ and $\eta_{trm} = 1.00$. For the same transmission efficiency, it can be observed that the CVT is the most efficient technology (e.g., 7% lower energy required at $\eta_{trm} = 0.95$ compared to the single gear for both cycles). For the NEDC the sensitivity is



(a) NEDC



(b) FTP75

Fig. 5: Transmission efficiency

approximately constant, whereas the energy reduction for an improvement in transmission efficiency is more sensitive for the FTP75 (especially, for the optimal MT, see, Table VIII).

TABLE VII: Normalized energy at $\eta_{trm} = 0.95$

Shift strategy	NEDC	FTP75
Single gear	100	100
Shifting DC manual	96	99
Optimal manual	94	95
Optimal CVT	93	93

VI. Conclusion

The conclusions, following from the four posed research questions in the introduction, are discussed below.

1) Due to shifting (both standard and optimal strategy), the energy consumption can be decreased by relative values between 1%-4% and 5%-6% (compared to an optimized single gear), depending on the drive cycle (FTP75-NEDC) and optimizing the shifting strategy respectively.
2) The difference in energy consumption between the transmission types is relative large. For example, a CVT with optimized shift strategy still consumes about 8%-10% more energy on both driving cycles than an optimized single gear. This is mainly due to the lower CVT efficiency.
3) The energy consumption increases with increase of the final drive ratio. The range of the feasible final drive ratios has a lower and upper limit, which depends on the transmission type and driving cycle. The optimal final drive ratio tends to be at the lower limit. The electric machine is generally more efficient at higher angular velocity.
4) If the transmission efficiency of all the transmission types are equal assumed, then the CVT is typically the most energy efficient transmission type. For example, an energy saving of 7% is calculated compared to a well-selected fixed-gear ratio transmission type. The CVT operates the EM far more often in the high efficiency regions.

TABLE VIII: Sensitivity $\frac{\Delta(\Delta E_s)}{\Delta \eta_{trm}} = \frac{(\Delta E_s(1.00) - \Delta E_s(0.95)}{1.00 - 0.95}$

Shift strategy	NEDC [kWh]	FTP75 [kWh]
Single gear	-1.58	-2.81
Shifting DC manual	-1.61	-3.23
Optimal manual	-1.62	-3.31
Optimal CVT	-1.59	-3.14

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