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Estimating real-world energy savings of an electric city bus with in-wheel motors versus central drive configuration

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Summary

A model-based in-principle comparison has been made of the energy demand of two different driveline configurations of a typical 12m electric bus: a configuration with a central driving electromotor and a configuration with two in-wheel motors. For this, the mechanical drivetrain losses occurring in propeller shaft, differential and final gear have been estimated and compared with representative data from literature. Energy demand was estimated for a range of relevant driving cycles. Results show that the lower drivetrain losses with the in-wheel motor version result in an 8 to 11 percent lower energy demand.

Keywords: Bus, EV (electric vehicle), Efficiency, Wheel-hub motor, Simulation

1 Introduction

Electrification of mobility is a key element in policies that aim to improve air quality in densely populated urban areas. Electrification of public buses has been an important path towards achieving these objectives. Currently, most electric buses in operation apply one central electromotor that is connected to the driving axle in the manner previously developed for conventional ICE (internal combustion engine) powered buses.

At the same time, several companies (such as Ziehl-Abegg, e-Traction and Hyundai Motors) have been advocating and producing in-wheel driveline solutions. Because the latter solutions eliminate the use of a propeller shaft, gearboxes and differential they promise a significant reduction in energy demand thus lowering the total cost of ownership of these vehicles.

Unfortunately, very little data can be found in the literature that will substantiate these claims. And even if such data were to be found, they would apply to vehicles with different road loads and they would have been determined either in simplified (SORT) test cycles or from real-world monitoring (with all the variability and uncertainties typical of such monitoring). For this reason, it was decided to make a model-based in-principle comparison of the energy demand of a typical urban bus in both driveline configurations. This report describes this work and discusses the outcome of the calculations.

2 Methodology

It was decided to perform a comparison between a central drive and in an in-wheel motor configuration for a typical 12m urban bus. The main road load characteristics of this bus are summarized in Table 1.

Table 1: Typical road load characteristics of a 12m urban bus

Frontal area [m2]	8.4
Vehicle mass, empty [kg]	13250
Vehicle mass in calculations [kg]	14500
Air drag resistance coefficient Cd [-]	0.7
Rolling losses resistance coefficient Crr [-]	0.006
Tyre type	275/70R22.5

For the central drive configuration, a drivetrain layout was used that resembles that of the VDL CITEA SLF-120 E electric bus. In this bus a central motor is connected directly (without gearbox) through a propeller shaft with a conventional rear axle with differential.

As energy flows from the battery towards the wheels of the bus (and back in the case of brake energy recuperation) losses occur in the battery, in the power electronics system (i.e. in the DC/AC invertor), in the electromotor and in the mechanical transmission to the wheels.

In this study it was assumed that both central drive and in-wheel motor configurations have the same battery pack (similar build-up and cell internal resistance). For the capacity of this battery pack a typical value of 220 kWh has been chosen. It was further assumed that the energy lost in the power electronics system was small and a constant fraction of the total energy transmitted. Of course, the performance characteristics of the central drive motor and in-wheel motors are different. The max torque, speed and power specifications for the central drive motor are based on those of the Siemens 1DB2016 electromotor [1]. The speed ratio between motor and wheels for a central drive bus was given a representative value 6.12:1. The corresponding specifications for the in-wheel motor were derived from this ratio and are shown in Table 2.

Table 2: Central drive respectively in-wheel motor specifications

	Central drive	In-wheel
Maximum power [kW]	160	80
Maximum torque [Nm]	2500	7650
Maximum speed [rpm]	3500	572
Best point efficiency[%]	96	93

Table 2 also mentions the best point efficiency of the electromotor (including that of the power electronics). In this study it was assumed that a permanent magnet synchronous machine (PMSM) would be applied in both drivelines. Manufacturers of state-of-the-art PMSM claim peak efficiencies up to 96 %. As the in-wheel motor has a lower power rating and because it has to deal with more severe design constraints (in terms of packaging and cooling) its efficiency tends to be lower. For their 90 kW PD18 4250, Protean mention a 93 % driving efficiency [2]; for their 240 kW motor, e-T-action claim a 94 % peak efficiency [3]; for their 182 kW motor Ziehl-Abegg mention a 92 % maximum efficiency [4]. All of these in-wheel motors are outer rotor type machines. With in-wheel motors of the inner rotor type there will be additional efficiency losses due to the built-in fixed transmission. The values selected for this study are therefore considered to be representative (and maybe even slightly advantageous to the in-wheel driveline configuration).

2.1 Modelling approach

To build the corresponding vehicle model, use has been made of the Simcenter/Amesim modelling environment [5]. This environment provides a library of models for electric and mechanical subsystems that can be integrated into larger (driveline) component systems. In Amesim, such an integrated system is referred to as a supercomponent. Subsystems and supercomponents can in turn be integrated into a complete driveline respectively vehicle model.

One of the subsystems in this vehicle model is the electric machine. In Amesim a default electromotor component model is available with a corresponding generic efficiency map. The shape of the efficiency map can be modified (through homothetic transformation) by changing the position of its reference operating point (that is the operating point with highest efficiency) and by modifying the actual value of this highest efficiency. For the central drive motor, the position of the reference point is shown in Figure 1. This figure also shows the maximum speed and torque levels mentioned in Table 2.

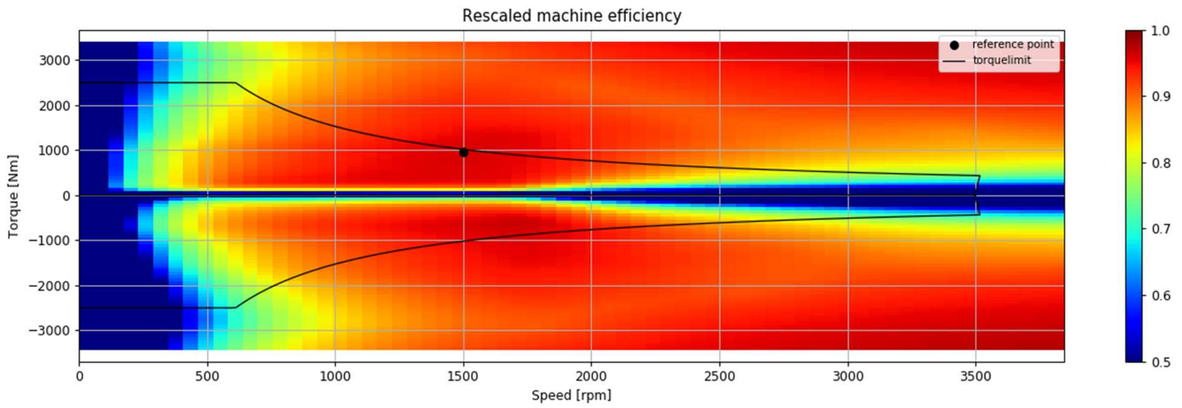

Figure 1: Central drive motor efficiency map (including power electronics losses).

Similarly, Amesim allows to define a regenerative braking strategy. For this study it was assumed that braking is default performed by the electromotor (up to the electromotor torque limit); where the brake torque exceeds this limit, the remainder of the brake torque is provided by the service brakes. With this assumption the energy recuperation is the maximum possible within the design constraints. This implies of course that service brakes are also part of the in-wheel motor design. Similarly, it is assumed that the above mentioned battery pack size of 220 kWh does not pose a limitation to the power input to the battery.

2.2 Mechanical driveline loss modelling

Very often the drivetrain mechanical efficiency is assumed to be constant and in a range between 95 and 98 percent [6]. These average levels are however representative for typical truck applications, where the vehicle is operating most of its time at much higher speeds than with urban buses. In this study an effort was made to take into account the variation in efficiency as a function of motor speed and torque. Mechanical (friction) losses will occur in the following driveline components: in the electromotor, in the propeller shaft, in the differential, in the end gear reduction and finally in the wheel bearings.

For each of these components a corresponding Amesim submodel or supercomponent was constructed. The main details of this modelling are presented below:

- The mechanical losses occurring in the electromotor (as well as the power electronics losses) are assumed to be included in the electromotor efficiency map (as explained above).
- The propeller shaft supercomponent calculates the losses that occur in both cardan joints of this shaft. This calculation is based on the work by Morecki as mentioned in [7]. Sliding friction losses that occur when the (splined) shaft length changes with changing shaft angle are not taken into account (as they are typically much smaller than the cardan losses).
- Separate supercomponents have been created for the differential and for the end gear reduction. The losses in these supercomponents result from those in the different plain and roller bearings that are part

of these components. To calculate the losses in the plain bearings the corresponding standard Amesim module (‘TRBJ01A – journal bearing’) was used [5]. To calculate the losses in the roller bearings, a new submodel was created that follows a general bearing friction calculation method proposed before by a well-known bearing supplier [8]. Representative sizing of the bearings was based on experience and on information/pictures found in [9]. Oil churning losses that would occur in the differential have been neglected.

- Wheel bearing losses have not been calculated separately. They are assumed to be part of the rolling resistance losses.

As an illustration, Figure 2 shows the supercomponent that was built in Amesim to calculate the roller bearing losses. As indicated, the total bearing losses are the superposition of static friction losses, seal friction losses and load dependent losses (as in [8]).

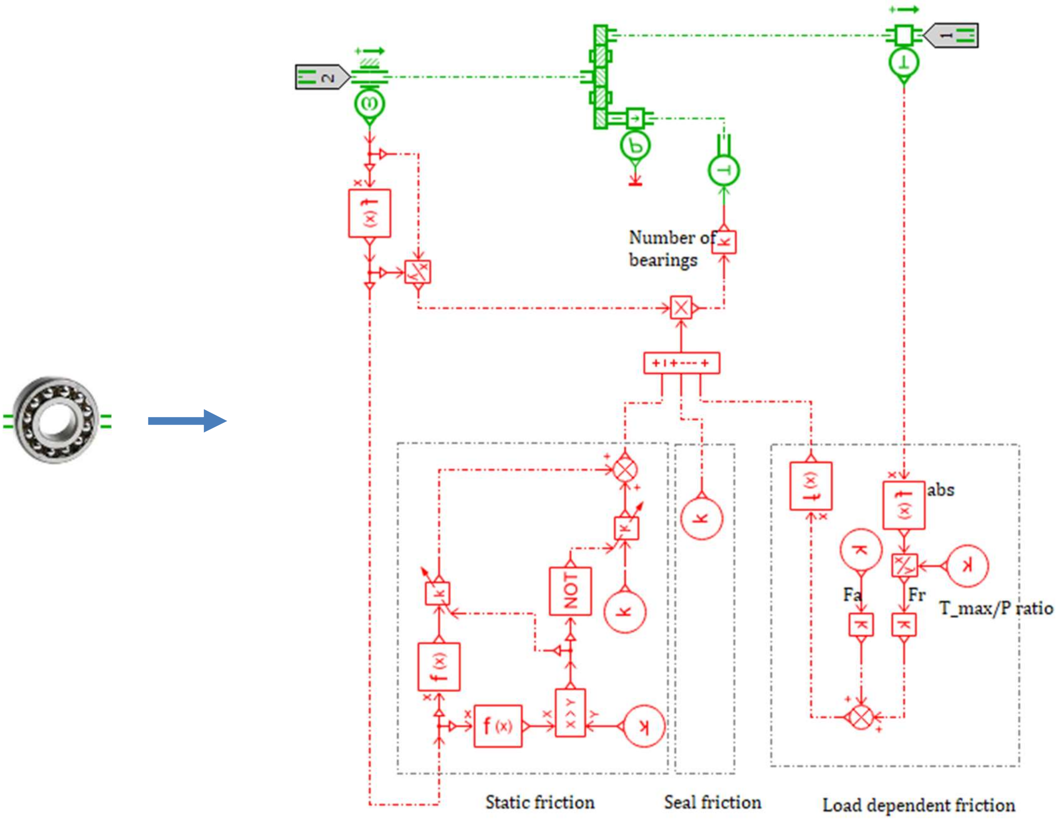

Figure 2: Central drive motor efficiency map (including power electronics losses).

Table 3 gives an overview of the actual losses that have been modelled for the different new supercomponents.

Table 3: List of components where mechanical losses occur.

Supercomponent	Location and type of friction loss	Occurrence (#)
Driveshaft	Plain cross joint bearing	4x2
	Slide mechanism	1
Differential	Roller bearing between rack shaft and bel house	2
	Contact rack/pignon	1
	Roller bearing between diff. body and bel house	2
	Plain bearing between sun wheel and diff. body	1
	Plain bearing between output shaft and diff. body	2
End gear reduction	Contact teeth drive gear – intermediate gears	2x2
	Plain bearings intermediate gears – bel house	2x4
	Contact intermediate gears – driven gear	2x2

2.3 Further modelling assumptions

For this in-principle comparison the effect of road gradients has been neglected. Further, ambient driving conditions were kept constant: wind speed was set at zero and ambient temperature and pressure were set at 20 ˚C and 1013 mbar respectively. Finally, it was assumed that the propeller shaft is always ideally aligned (this implies that when the vehicle mass changes with passenger load, the suspension system automatically adjusts the vehicle distance to the road). As a result of this assumption, cardan joint losses do not occur.

3 Qualitative validation of mechanical driveline losses

No detailed experimental data were available on the mechanical losses occurring in the central driveline. To assess the representativeness of the predictions, a comparison has been made with detailed experimental data found in the literature [10] on the mechanical efficiency of the driveline of a medium tactical vehicle. Although this driveline also has a central drive configuration, it is not identical to the one modeled in this study (and not all of its details are mentioned). That means that the results can only be compared trend wise. The result of this comparison is shown in Figure 3.

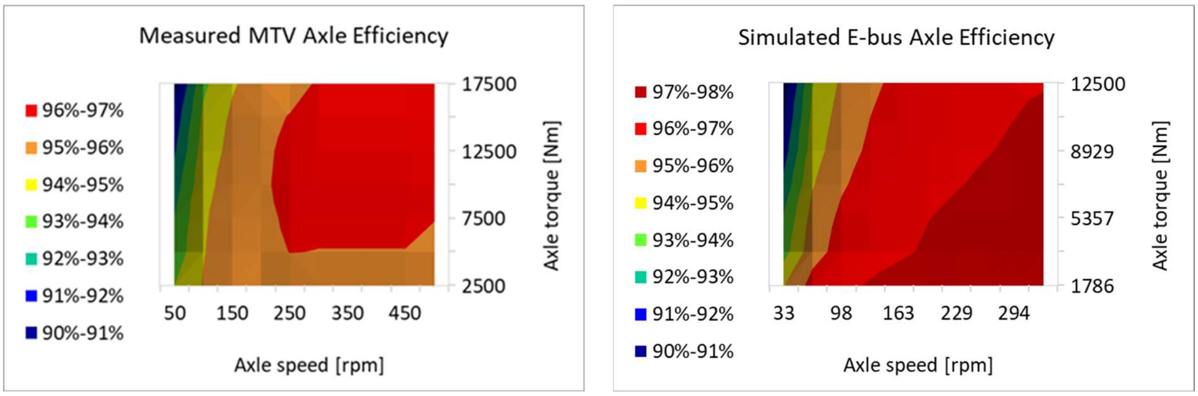

Figure 3: Comparison between simulated Central Drive (CDR) efficiency and measured efficiency from literature [10]

Obviously, both efficiency maps show a similar pattern. At low speed and torque conditions (a situation that occurs frequently with urban buses) the efficiency tends to drop significantly. The calculated mechanical efficiency in the range of higher torque and speed, typical of heavy-duty trucks, are in line with the values mentioned in the literature for those applications [6]. In view of these results, it was concluded that the mechanical efficiency model is expected to result in realistic predictions.

4 Results and discussion

Figure 4 shows the driveline efficiency at different (constant) vehicle speed levels for the central drive configuration (CDR) and for the in-wheel configuration (IW). This efficiency is defined as the ratio (in percent) of the energy supplied by the battery to the energy supplied to the wheels.

The IW driveline clearly has a higher efficiency. For comparison, also the efficiency of a theoretical central drive without mechanical losses is shown (CD). The lower energy efficiency of the IW-variant compared to this CD-variant originates with the lower electric efficiency of the former.

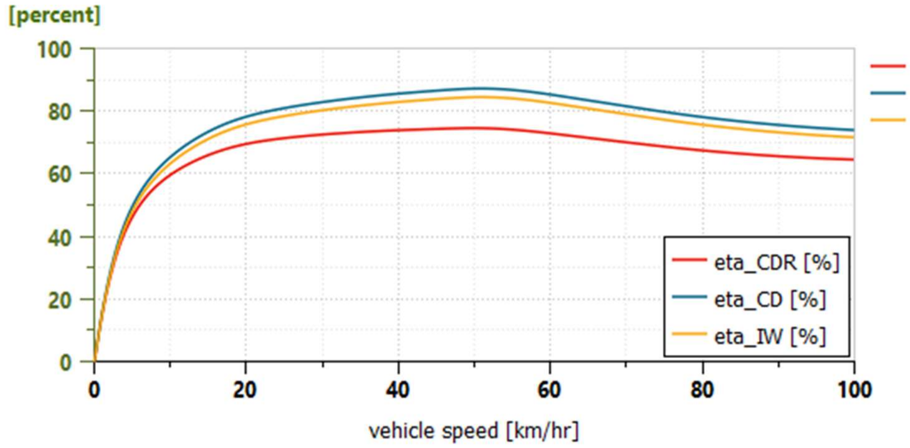

Figure 4: Calculated driveline efficiency versus vehicle speed for central drive with (CDR) and without (CD) losses versus in-wheel motor (IW) drive, all at 14500 kg vehicle mass

For the CD configuration, top efficiency of 88 % is reached at 50 km/h vehicle speed. This efficiency falls off towards 75 % at 100 km/h. For the CDR configuration the corresponding efficiencies are 73 % respectively 65 %. For the IW configuration they are 85 % and 72 %. These efficiencies are much lower than the maximum efficiencies. This is mainly because the torque required for driving at constant speed is relatively low.

Figure 5 shows the evolution of the State-Of-Charge (SOC) of the CDR, CD and IW drivelines (all at 14500 kg vehicle weight) in the Dutch Urban Bus cycle (TNO, [11]). Obviously, the in-wheel motor configuration has a significantly lower SOC-decrease.

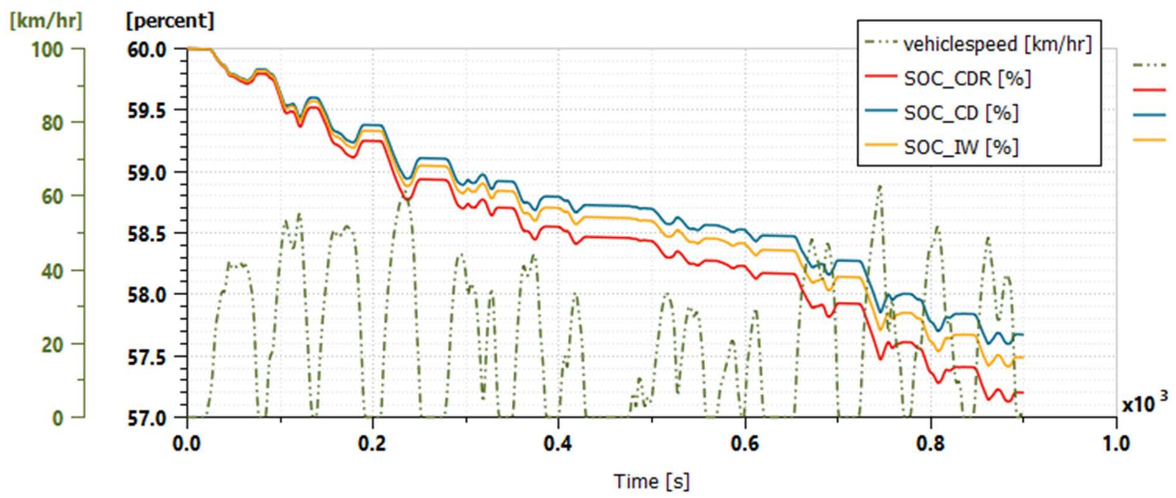

Figure 5: Battery State of Charge (SOC) decrease in the Dutch Urban Bus driving cycle. Results are for a bus with central drive with (CDR) and without (CR) mechanical losses and for a bus with in-wheel motor (IW).

Similarly, the energy consumption for both vehicle configurations has been calculated for an additional set of driving cycles. This set includes some initial SORT-cycles as they were defined in [12]. The corresponding results are shown in Table 3. The energy consumption shown in this table corresponds to the nett energy supplied by the battery pack to the driveline per km displacement.

Table 3: Cycle energy consumption for different driveline configurations.

Cycle energy consumption [kWh/km]	Central drive	Central drive without friction losses	In-wheel	In-wheel
Vehicle weight [kg]	14500	14500	14500	14200
Dutch Urban Bus cycle [11]	1.04	0.86	0.93	0.91
Braunschweig cycle [11]	1.00	0.85	0.90	0.88
Initial SORT heavy urban cycle [12]	1.21	1.00	1.10	1.08
Initial SORT suburban cycle [12]	1.21	1.03	1.12	1.09

From Table 3 it is clear that the IW driveline consistently has a lower energy demand than the CDR driveline. These calculation results indicate that the energy savings resulting from the application of in-wheel motors is expected to be in the range of 8 to 11 percent.

At the same time, in the literature claims of up to 16.5 % higher efficiency have been mentioned (based on benchmarking tests where vehicle mass was kept constant) [4].

To put these claims into perspective, again, also the energy demand for a hypothetical "ideal" central driveline without mechanical friction losses has been calculated. This energy demand would be equivalent to that of an IW-configuration that would use an electromotor with 96 % maximum efficiency (i.e. 3 % better than we assumed until now). These results indicate a maximum theoretical benefit of 15 to 18 percent. The abovementioned efficiency benefit of 16.5% therefore seems to be on the high side of what can be expected.

At the same time, to be fair, in reality additional losses occur with the central driveline due to cornering and due to propeller shaft angle variation (linked to vertical vehicle dynamics). The calculated benefit of 8 to 11 percent in Table 3 does not take this into account.

Furthermore, in all of the calculations discussed until now, it was assumed that the change in driveline configuration did not affect the vehicle weight. However, the in-wheel motor configuration tends to have a lower weight than the central driveline variant. For the bus configuration considered here, this weight benefit is estimated to be in the range of 300 kg. Hence, the different calculations for the IW-configuration were performed again, now for a total vehicle weight of 14200 kg. The results are again shown in Table 3. The weight reduction results in an additional predicted efficiency benefit of the order of 2 %.

Based on all of the above, it can be concluded that the shift from a central drive towards an in-wheel driveline configuration has a considerable advantage in terms of energy consumption. In addition, a switch to in-wheel motors gives more freedom in the interior vehicle design. Another typical advantage is the possibility for independent wheel torque control (this is however considered to be less relevant for urban bus applications).

Nevertheless, some additional comments are in place to put these advantages in perspective:

- In a central drive configuration, adding a (two or more speed) transmission would allow to have a better match between vehicle operation and electromotor efficiency map. Such transmission would be difficult to integrate in an in-wheel motor driveline.
- In-wheel drivelines are characterised by a higher un-sprung mass and higher wheel inertia; these characteristics tend to worsen comfort and can have a negative impact on road holding.
- The complex subsystems of the in-wheel motors are more subject to harsh environmental conditions and strong vibrations, raising some concerns about durability. Similarly, there are concerns about higher maintenance costs.
- Finally, in-wheel motor drivelines tend to be more expensive than equivalent central driveline solutions, in particular because of their higher complexity and smaller production numbers.

Based on these observations it is clear that there is a need for a more holistic comparison between the two options.

Acknowledgments

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