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Measuring the Efficiency of Reduction Gearboxes for Electric Utility Vehicles during Specific Driving Cycles

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This paper presents the results and the procedure for measuring the efficiency of a single-speed reduction gearbox developed for an autonomous electric utility vehicle. The resulting efficiency of the gearbox was investigated on three different driving cycles, which were selected because their speed profiles most closely matched the expected use of the autonomous vehicle. The required torque for each cycle was obtained from simulations of the vehicle's driving behaviour including its predicted mass and dimensional parameters after a given driving cycle. The results of this research represent the achieved efficiency and average power loss of the gearbox on each driving cycle. The resulting gearbox efficiency was around 50 % in the predominant areas of driving cycles.
Keywords: efficiency, gearbox, powertrain, electric vehicle, driving cycle

Highlights
- *Presentation of a method for measuring the instantaneous efficiency of a separate gearbox during driving cycle.*
- *Study of a gearbox designed for an autonomous electric utility vehicle.*
- *The efficiency of the gearbox can have a significant effect on the electric powertrain efficiency in certain cases.*
- *The gearbox achieved average efficiency values of around 50 % in the predominant driving cycle areas.*

0 INTRODUCTION

One of the most significant disadvantages of car transport is its adverse impact on the environment. For this reason, car manufacturers are under constant pressure from national governments to reduce the pollutants produced by internal combustion engine cars. This pressure has resulted in the search for new solutions to develop internal combustion engine vehicles or new alternative means of propulsion for these vehicles, as hybrid electric vehicles (HEV), plug-in hybrid electric vehicles (PHEV), battery electric vehicles (BEV), etc. Car manufacturers are also under pressure from customers to maintain optimum performance, low consumption (in the case of high mileage electric vehicles (EVs)) and low prices. For this reason, it is now essential to look for possible compromises and ways to achieve the desired result.

One of the ways to reduce emissions or increase the range of electric vehicles, a topic which is currently being addressed by a large number of researchers, is the optimization of individual powertrain components to reduce energy loss in the drive train of the car [1] and [2]. Other researchers say that the so-called Achilles heel of electric vehicles is its batteries [3].

The results [4] show a trend towards improved power train efficiency for all types of conventional cars, with the comparative average power train efficiency for all vehicles in the categories reaching 18.8 % in 2005 and 20.9 % in 2013. In this study, the powertrain efficiency of 37 pairs of conventional vehicles of the same model was compared for the years 2005 and 2013. The eighteen pairs were passenger cars, two pairs were minivans, twelve pairs were sport utility vehicles (SUV) and five pairs were pickup trucks [5].

In general, the tank-to-wheel efficiency of the vehicles with internal combustion engines is 14 % to 33 % for gasoline and 28 % to 42 % for diesel. The powertrain of an electric vehicle has a significantly higher efficiency of 50 % to 80 %. In BEVs, the lossiest components are the electric motor, electric generator and the mechanical transmission if the electric vehicle is equipped with one [6].

In this study, we investigated the efficiency of the gearbox for an autonomous electric utility vehicle, which, according to previous research, together with the efficiency of the electric motor, has a significant effect on the overall energy consumption of electric vehicles [6].

In general, gearbox efficiency is investigated in two basic ways. One way is through the use of mathematical models and advanced simulations, and the other important tool is experimental measurements. Experimental measurements of gearbox efficiency are usually performed at dedicated test benches with electric dynamometers, and experiments of ten measure efficiency over the full range of operating input speed and input torque of the gearbox. In contrast, the energy consumption of electric vehicles is commonly measured and investigated during driving cycles.

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By investigating energy consumption in a driving cycle the effects of individual powertrain components can also be investigated. We ask the question, what is the effect of gearbox efficiency on the energy consumption of an electric vehicle on a drive cycle? We would like to answer it using data from experimental measurement of the gearbox over the full range of input speed and input torque. We would have to simulate driving the vehicle, in which we would specify the data from the gearbox efficiency measurement as one of the simulation parameters.

Therefore, as part of the research and development of electric powertrains for electric utility vehicles, we asked the question, how would it be possible to directly measure gearbox efficiency during driving cycles? Such results could provide us with accurate and valuable data on the effect of gearbox efficiency on the energy consumption of electric vehicles during the driving cycles, while also providing valuable insights for the development of electric vehicle gearboxes. To answer the questions posed and also to provide such experimental measurement data, we experimentally measured gearbox efficiency in our power train laboratory, which allows such experimental measurements to be conducted during driving cycles, and we present this efficiency measurement method along with the results of our measurements.

1 METHODS

In this work, we investigated the efficiency of a gearbox for an electric utility vehicle during driving cycles. Gearbox efficiency is usually given either as a number or as a set of curves which show the dependence on speed or torque. Gearbox efficiency η is calculated as follows:

$$\eta = \frac{P_2}{P_1}\cdot 100\,\% = \frac{\omega_2 \cdot M_2}{\omega_1 \cdot M_1}\cdot 100\,\% = \frac{2\pi \cdot n_2 \cdot M_2}{2\pi \cdot n_1 \cdot M_1}\cdot 100\,\%, \quad (1)$$

where P_2 is the output power of the gearbox, P_1 is the input power of the gearbox, ω_2 is the output angular velocity, M_2 is the output torque, ω_1 is the input angular velocity, M_1 is the input torque of the gearbox and n_2 and n_1 are the speeds of the output and input shafts of the gearbox. The range of values of the efficiency during our experiment is $-1 \leq \eta \leq 1$. The efficiency becomes negative when the vehicle is significantly decelerating, and power is transmitted in the opposite direction to acceleration or steady-state driving.

Efficiency values for a conventional manual passenger car gearbox is typically in the range of 92 % to 97 %. Typical values for truck gearbox are 90 % to 97 %. The efficiency of the individual pairs of spur gears is approximately 99 % to 99.8 %. In the case of bevel gears, the efficiency is estimated at 90 % to 93 % [7]. Load-independent losses are due to gears grinding in the oil bath and grinding of bearings. This groups also consists of losses arising in the seals, and other losses [8].

To perform experimental measurements gearbox efficiency during driving cycles, we obtained input data for the experiment in the form of input and output values for gearbox speed and torque based on the driving resistance forces of the vehicle. For this purpose, we used an advanced vehicle driving simulation on the drive cycles in software Ricardo Ignite [9] to provide the input data for the experiment. This data was then used as input to the control system of the powertrain test bench where we measured the efficiency of the gearbox. During the experiment, we measured and recorded the input and output speeds and torque values of the gearbox. We then processed the measured data in Matlab software to obtain results for the efficiency and power losses of the gearbox during the driving cycles.

We performed experimental measurements on three driving cycles of vehicles designed for close operational use of an autonomous platform for which the measured gearbox is being developed.

1.1 Tested Gearbox

The gearbox for the autonomous battery electric modular platform developed at the Technical University of Liberec was selected for study. Based on our previous research and development in gearboxes for electric vehicles, we developed our own gearbox.

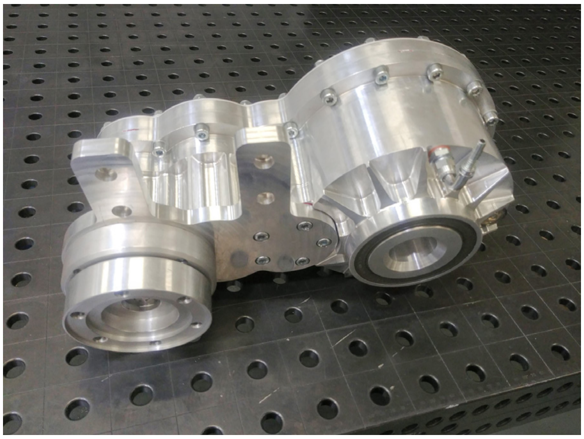


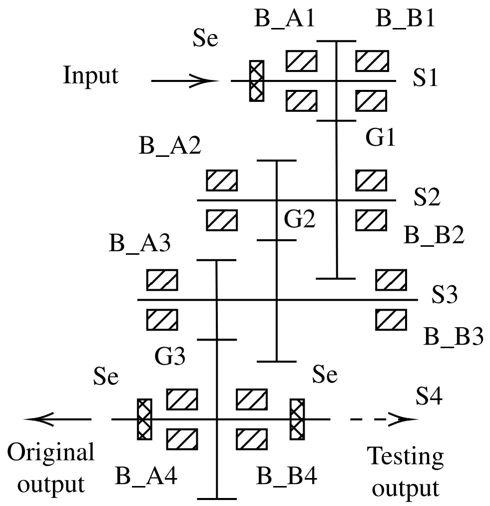
Fig. 1. *Tested gearbox for the autonomous battery electric modular platform*

The vehicle will be driven independently by four electric motors, making it possible to design the transmission without a differential and the use of advanced torque vectoring technology for steering. Because the vehicle has four motors, four identical gearboxes will be fitted, housed in two assemblies, one on each axle (Fig. 1). Each single-speed gearbox has a gear ratio of 11.447 : 1, with a maximum input torque of 150 Nm and a maximum input speed of 6000 rpm.

In designing the gearbox, we were severely limited by the requirement to make one design for all four motors. A gearbox layout is shown in Fig. 2. The gearbox consists of four shafts and six gear wheels, which are partially taken from the Skoda Auto MQ200 automotive gearbox to reduce the production cost and eight bearings produced by SKF. The specific types of bearings used on each shaft are shown in Table 1. The sealing of the oil filling in the gearbox is ensured by shaft seals on the input and output shafts. The material of the housing is aluminium alloy EN AW 6061 T6.

Table 1. *Used bearings*

Shaft	Bearings A	Bearings B
Input shaft	BK3016	6205
First counter shaft	4206 ATN9	HK 3020
Second counter shaft	32005 X	32005 X
Output shaft	6014	NK70/25



B_A - Bearings A | B_B - Bearings B
G - Gear ratio | $S1$ - Input shaft
$S2$ - 1st countershaft | $S3$ - 2nd countershaft
$S4$ - Output shaft | Se - Sealing

Fig. 2. *Tested gearbox layout*

1.1 Simulation of the Vehicle's Driving on the Driving Cycles

The torque required to move the vehicle at each instant was calculated from the parameters of the developed modular platform using Ricardo Ignite [9] simulation software. The main parameters defined in the simulation model are shown in Table 2. The other parameters required for the torque calculation were left at their default values.

Table 2. *Vehicle parameters*

Parameters	Value
Vehicle frontal area [m²]	1.925
Aerodynamic drag coefficient	0.7
Rolling resistance coefficient	0.08
Tire size	225/65 R17, 325/80 R22.5
Vehicle mass [kg]	3000

The driving cycles for our experimental measurements were selected according to two criteria. The first criterion was a speed limit based on the maximum input speed that our dynamometer is capable of producing. The second criterion was selection of a cycle with a speed profile that would match the expected use of the autonomous modular platform, which is expected to be applied, for example, in mines, ports or warehouses. To assess the magnitude of the total efficiency in the gearbox, three types of driving cycles were selected. These are the CARB Heavy Heavy-Duty Diesel Truck (HHDDT) Creep Segment (CARB–HHDDT–CS), Central Business District (CBD) Segment of the Transit Coach Operating Duty Cycle (CBD–SoTCODC) and NREL Port Drayage Creep Cycle (California) (NREL–PDCQC) cycles. Fig. 3 and Table 3 show the basic parameters and speed profiles of the individual driving cycles which the experiment measured [10].

Table 3. *Driving cycle parameters*

Cycle	Time [s]	Distance [km]	Maximal speed [kmh⁻¹]	Avg. driving speed [kmh⁻¹]
CARB-HHDDT-CS	253	0.19	13.19	4.85
CBD-SoTCODC	560	3.29	32.18	25.65
NREL-PDCQC	1330	0.41	20.05	8.36

The data show that the CARB–HHDDT–CS cycle was the shortest of the lowest cycles examined, with the lowest maximum and average speeds. The CBD–SoTCODC cycle attained medium values. By contrast, the NREL–PDCQC cycle attained the highest values for cycle duration, maximum and average speed. For the CBD–SoTCODC cycle, a different tire size was selected because of its maximum speed.

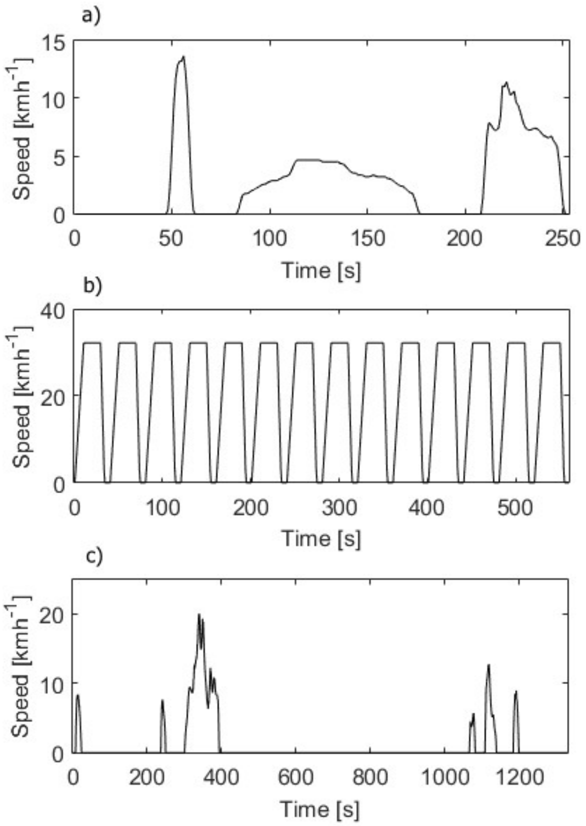


Fig. 3. *Speed profiles of investigated driving cycles, a) CARB Heavy Heavy-Duty Diesel Truck (HHDDT) Creep Segment, b) Central Business District (CBD) Segment of the Transit Coach Operating Duty Cycle, c) NREL Port Drayage Creep Queue Cycle (California)*

1.2.1 CARB Heavy Heavy-Duty Diesel Truck (HHDDT) Creep Segment

The Creep Segment contains the speed profile of the CARB HHDDT driving cycle. The test cycle on a cylindrical dynamometer developed by the California Air Resources Board (CARB) and West Virginia University measures emissions from heavy-duty diesel trucks in slow mode. This type of driving cycle corresponds to the ordinary use of a heavy-duty trucks with a total weight of around 14.000 kg. This driving cycle is used to measure vehicle emissions on a chassis dynamometer [10] and [11].

1.2.2 Central Business District (CBD) Segment of the Transit Coach Operating Duty Cycle

The Central Business District segment of the Transit Coach Operating Duty Cycle (SAE J1376), also known as the Business–Arterial–Commuter Cycle. A cycle consists of 14 repetitions with an acceleration ramp, maintaining speed at 32.18 km/h, followed by deceleration. It simulates passenger boarding and alighting in a business district with frequent stops and heavy traffic [10].

1.2.3 NREL Port Drayage Creep Queue Cycle(California)

This driving cycle was developed by NREL with data from vehicles operating at the Ports of Los Angeles and Long Beach. This cycle is very similar in kinematic intensity to the driving cycle given in the Section 1.2.1.

1.3 Powertrain Test Bench

The powertrain test bench was designed to test and optimize the parameters and long-term testing of the vehicle's powertrain. The test bench is equipped with four asynchronous dynamometers. These dynamometers are divided into pairs for testing the front and rear axles of the vehicle. The first pair of Siemens 136 ADG 288 WP dynamometers located on the front axle, achieves a maximum output of 136 kW at 500 rpm. The second pair of Siemens 111 ADG 286 WP dynamometers representing the car's rear axle, has a maximum output of 111 kW at 500 rpm. The device is controlled by a Simatic S7-300 PLC control system and LabView programming environment. Torque measurement is performed using HBM T10F strain gauge flanges with a measuring range of 5 kNm and sensitivity of 0.1 %, which ensures the accuracy of measurement even with dynamic speed variations.

1.4 Experiment Description

The gearbox was placed between two dynamometers on a unique frame consisting of aluminum profiles and a steel weldment. The input shaft of the gearbox was connected to one dynamometer, which operated in motor mode, using a semi-axle from Skoda Auto. The output shaft of the gearbox was connected in the same way to a second dynamometer located opposite the first. The second dynamometer was operated in torque mode. The gearbox in the test bench is shown in Fig. 4. The method of connecting the input and output of the gearbox to the dynamometers using the Skoda Auto semi-axles imposed a maximum input speed limit of 2000 rpm in our experiment. A maximum input speed of 2000 rpm corresponds approximately to a vehicle speed of 24 km/h (with tire size 225/65 R17, corresponding to off-road tires) or a speed of approximately 36 km/h (with tire size 325/80 R22.5, corresponding to truck tires). However, this limitation on the maximum vehicle speed, considering the driving cycles used, limited us to only the (NREL–PDCQC) cycle, in which the maximum input speed limit on the gearbox would be exceeded. For this cycle, we therefore chose a tire size of 328/80 R22.5, with which the input speed limit was not exceeded.

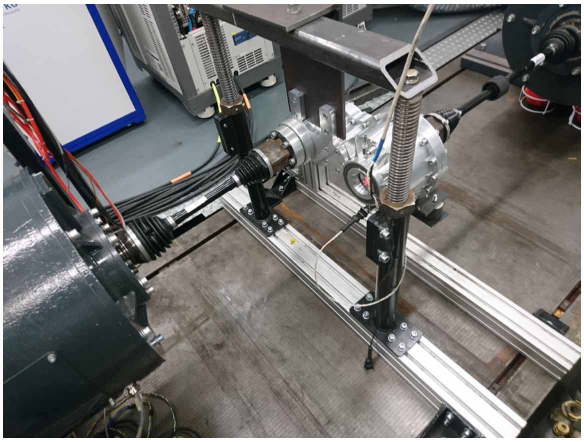


Fig. 4. *Gearbox in the test bench*

First, it was necessary to obtain input data for the control system of the powertrain test bench. As mentioned in the methods chapter, we used the advanced Ricardo Ignite [9] software tool for this purpose. With this software, for each selected driving cycle, we simulated vehicle driving with the parameters listed in Table 2. From the simulation, we obtained the input speed and the total input torque of the gearbox as if the vehicle were driven by a single motor. Since the gearbox was designed for a vehicle powertrain with four motors and four gearboxes, we divided the total input torque obtained by four. We input the obtained simulation data into the control system of the powertrain test bench. We then performed three experimental measurements with the test bench: one measurement for each selected driving cycle, in which the control system, according to the input data from the gearbox, powered the driving cycle as if the gearbox has been placed in a real vehicle.

During the experiment, the control system recorded the input and output speeds and torques of the gearbox from sensors on both dynamometers. The oil temperature in the gear box was measured by the temperature sensor in the gearbox and recorded. The ambient temperature was measured and recorded in the same manner. The speed and torque values were recorded every 0.01 s and the temperature values every 0.1 s.

Measuring the efficiency of a mechanical gearbox over a complete driving cycle is a highly complex process. As mentioned earlier, the values of each input power and output power component were logged every 0.01 s. From each sequential 100 values logged (100 values = 1 s), an average value was calculated for each input power and output power component, which is shown as a circle in the Fig. 5 for illustration purposes in the drive cycle CARB–HHDDT–CS.

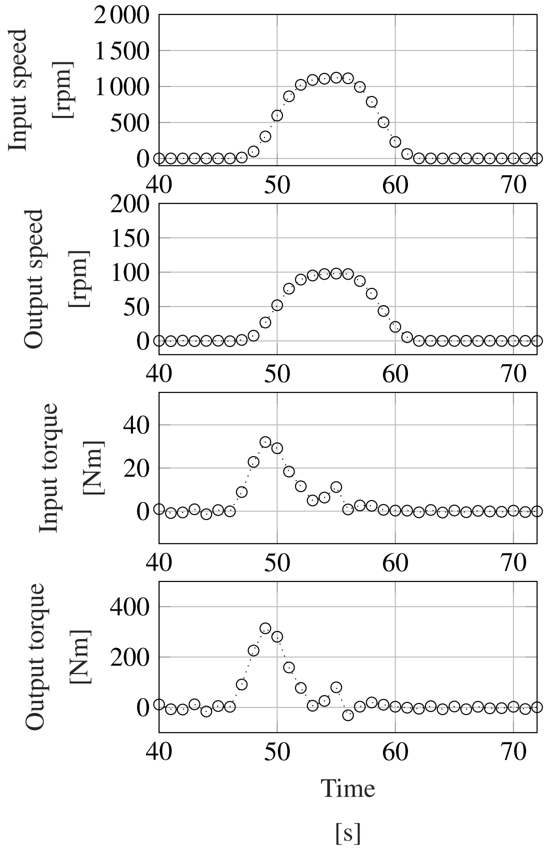


Fig. 5. Detail of parameter processing for the CARB–HHDDT–CS

Subsequently, from the obtained average values of the input and output power components, the values of input power, output power and efficiency were calculated according to the Eq. (1). For illustrative purposes, these values are similarly presented in the following CARB–HHDDT–CS driving cycle detail in Fig. 6.

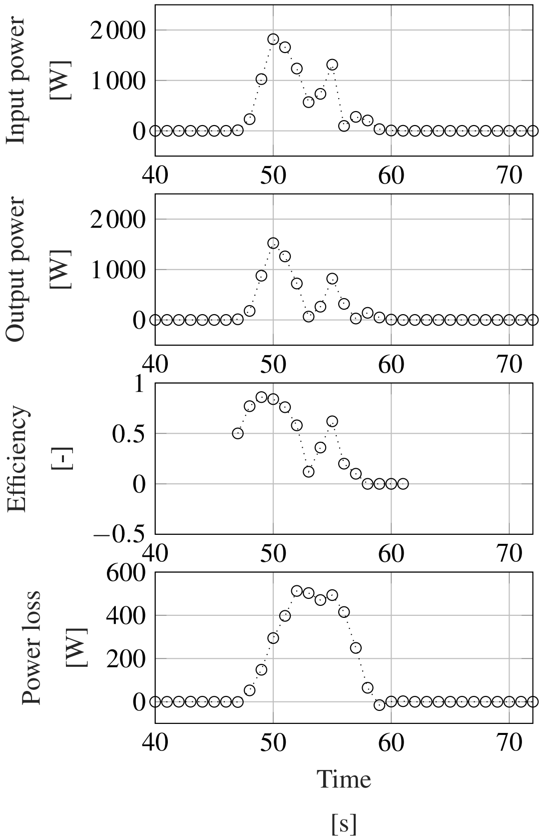


Fig. 6. *Detail of the calculated results of the CARB HHDDT Creep Segment driving cycle measurement*

The areas where the vehicle is not driving were considered as undefined areas. In this condition, the determined efficiency would correspond to the Eq. (2). For these areas, the efficiency was not defined. Simultaneously, the efficiency values in these areas are not shown in the resulting graphs.

$$\eta = \frac{P_2}{P_1} = \frac{0}{0} \Rightarrow \text{undefined.} \quad (2)$$

We would like to note here that the reported gearbox efficiency results will probably still be slightly affected by the semi-axles used to connect the gearbox to both dynamometers. According to the experiment conducted in [12], we can note that the efficiency of a single shaft joint can range about 97 % to 99 %. A further note relates to oil temperature that the experiments were conducted with gearbox oil temperatures in the range of 26 °C to 28 °C and ambient temperatures around 24 °C. The gearbox was filled with gear oil with viscosity class SAE 90.

2 RESULTS

The results of the experiment are shown in the Figs. 7 to 9. The figures show the dependencies of input and output speed, input and output torque, input and output power, efficiency and power loss per unit time. We describe the results for driving cycles CARB–HHDDT–CS, CBD–SoTCODC and NREL–PDCQC below. Fig. 7 shows the results for the CARB–HHDDT–CS driving cycle. In this driving cycle, the gearbox was operated at three speed ranges that correspond to the prescribed vehicle cycle speed profile. The first section was slightly above the 1000 rpm input speed value, the second section was at a lower speed, approximately 150 rpm to 500 rpm, and during the third section, the gearbox input speed was in the range of 500 rpm to 1000 rpm. The gearbox input torque corresponding to the torque required by the vehicle to handle the specified speed profile of the driving cycle rose briefly during the first section to a maximum value of 30 Nm. During the second section, it was fairly constant mostly around the value of 10 Nm. During the third section, three torque peaks can be observed in the range of 15 Nm to 30 Nm at changes in the vehicle's speed. The gearbox efficiency values peaked at 70 % to 96 % in all three sections, with the efficiency results varying over time during the cycle due to the relatively dynamic input torque profile. The instantaneous power loss values, which correspond to the value of dissipated power in the gearbox during the cycle, reached a maximum value of 502 W in the first section. In the second section, the values were around 100 W, and in the third section, a maximum value of 408 W was reached. The average power loss was 67.9 W. Fig. 8 shows the results for the CBD–SoTCODC. The speed profile of this driving cycle consisted of fifteen sections, in which the input speed of the gearbox increased to 1841 rpm each time after a defined ramp and then dropped again to zero. The gearbox was loaded with a maximum input torque of 44.5 Nm in each section. The maximum gearbox efficiency value was 94 %.The maximum power loss value achieved at a steady speed of 911.3 W, and the average power loss value during the cycle was 481 W. Fig. 9 shows the results for NREL–PDCQC. Six relatively short speed starts and stops with different profiles were performed during this cycle. The maximum input speed of the gearbox was 1677 rpm. The maximum input torque was 36.5 Nm. The maximum gearbox efficiency value was 94.8 %. The maximum value of power loss in the gearbox was 818.75 W, and the average value of power loss during the cycle was 32.88 W.

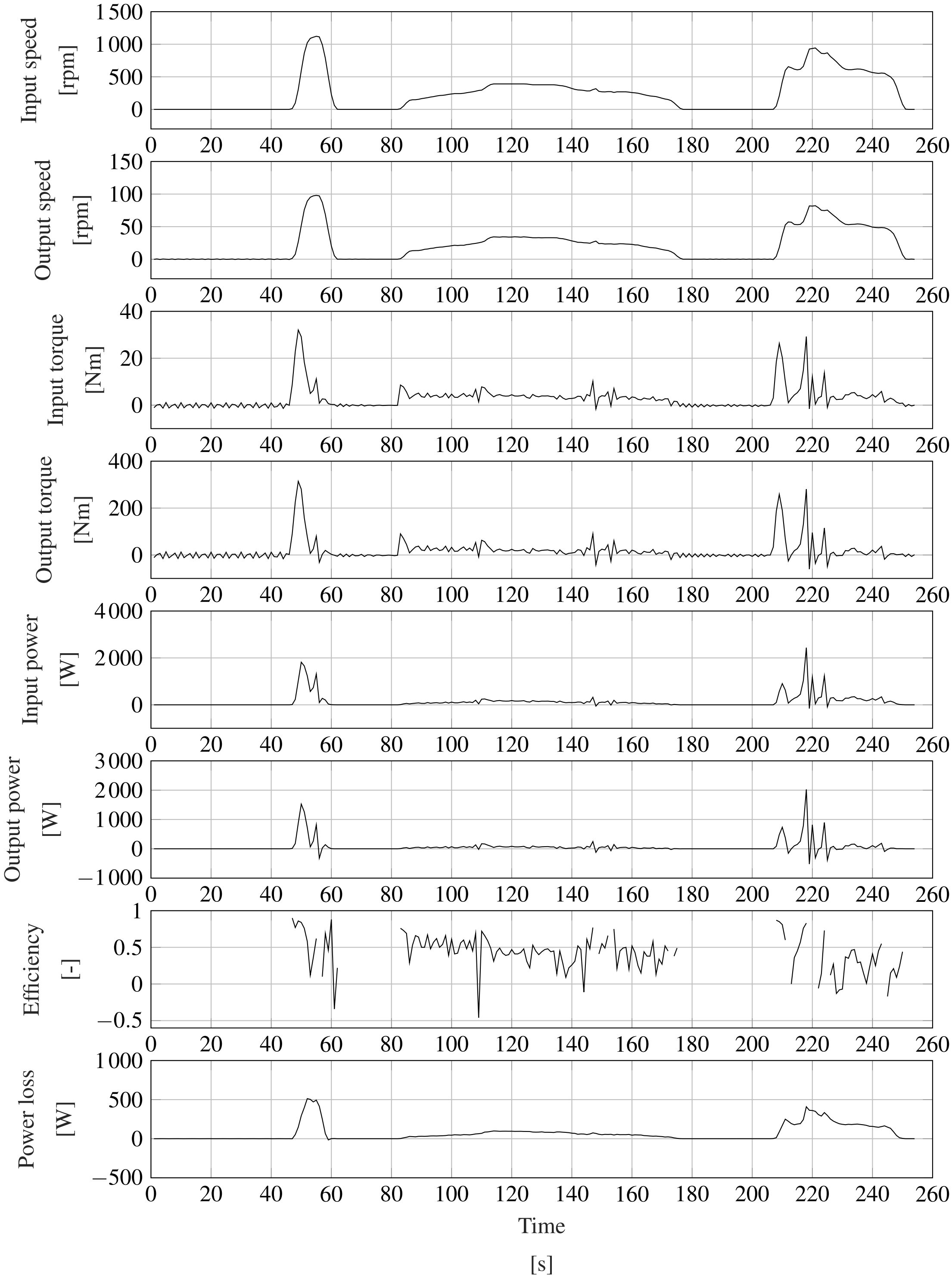


Fig. 7. *Input, output and calculated data for CARB Heavy Heavy-Duty Diesel Truck (HHDDT) Creep Segment*

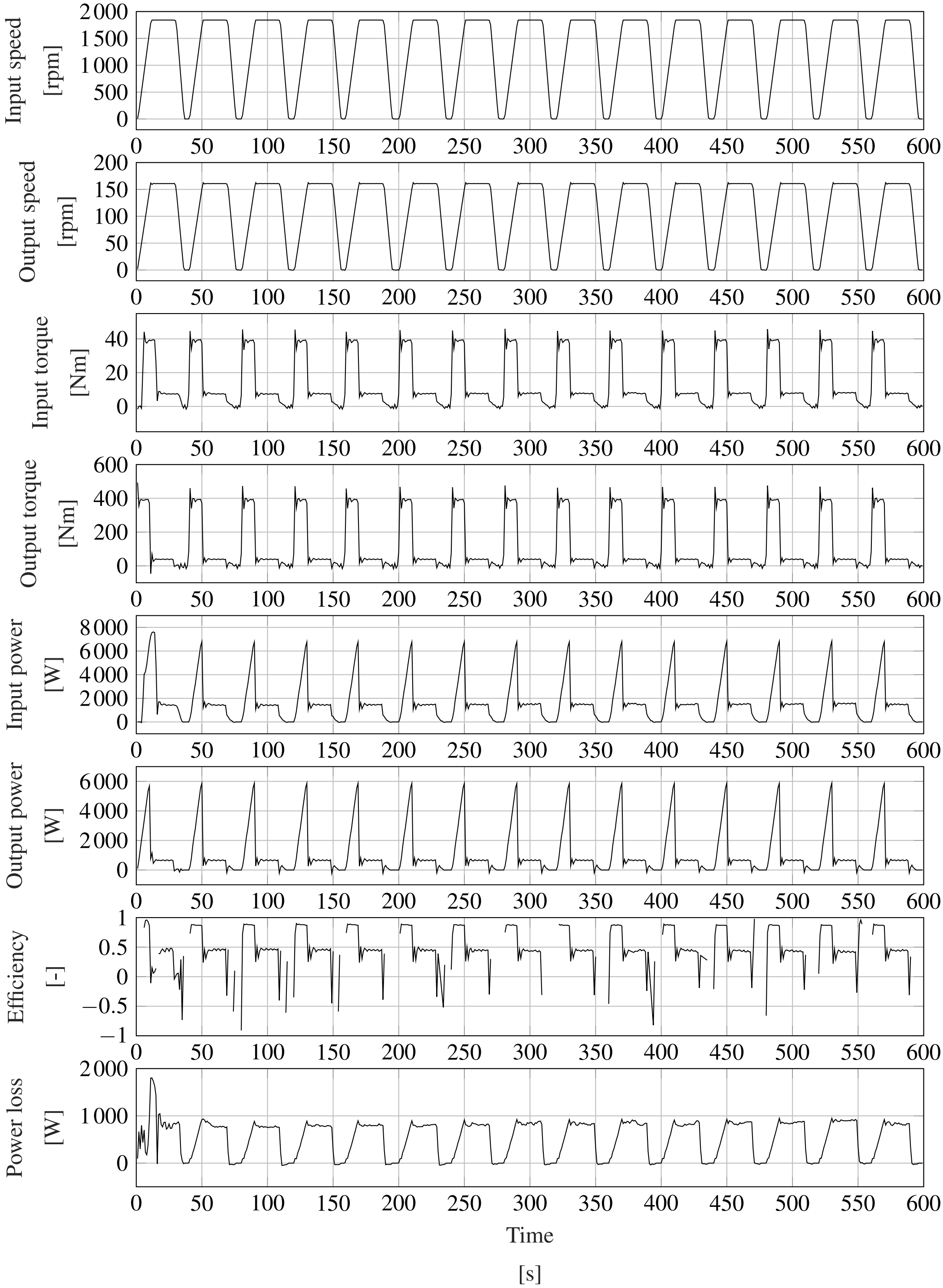


Fig. 8. *Input, output and calculated data for Central Business District (CBD) Segment of the Transit Coach Operating Duty Cycle*

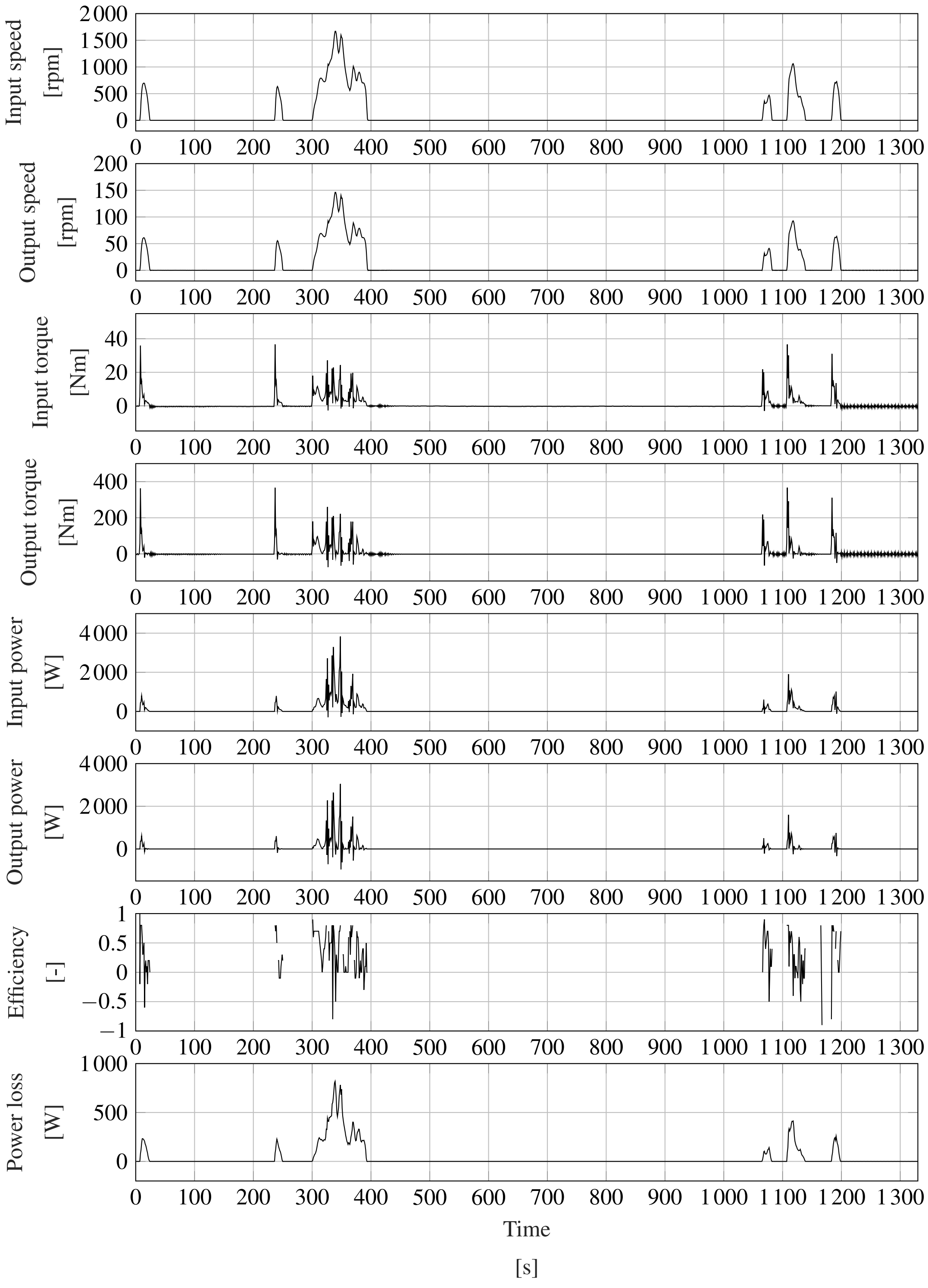


Fig. 9. *Input, output and calculated data for NREL Port Drayage Creep Queue Cycle (California)*

3 DISCUSSION

We presented a method for directly measuring the gearbox efficiency of an autonomous electric utility vehicle during driving cycles to obtain data on the effect of gearbox efficiency on the energy consumption of electric vehicles. Such data has a high potential for use in the future research and development of electric utility vehicle powertrains. Methods for measuring the overall efficiency of electric powertrains during driving cycles are already common, however, if we want to focus on research and development of separate optimisation of the efficiency of mechanical of powertrain, we should have an adequate understanding of the effect of the separate gearbox efficiencies.

To obtain data on the efficiency of the gearbox during driving cycles, we measured the efficiency of three driving cycles which simulated the driving operation of working utility vehicles using the described method. The method consists of measuring the mechanical power input to the gearbox and the mechanical power output of the gearbox during the driving cycle on a powertrain test bench.

The resulting values of achieved transmission efficiency during all measured driving cycles varied quite significantly depending on the input torque and input speed. At higher values of input torque, achieved transmission efficiencies of around 94 % could be observed from the measured data. However, in areas with lower input torque values, the achieved gearbox efficiencies were more around 50 %. These areas predominate in the measured driving cycles for the vehicle under consideration, and thus the gearbox was in areas of relatively low efficiency for the majority of the time. The achieved efficiency of around 50 % thus appears significantly low compared to the frequently reported gearbox efficiency values of 92 % to 97 % [7] sometimes reported in electric powertrain efficiency research articles. We explain such relatively low resulting gearbox efficiency values mainly for two reasons. The first reason is likely to be the characteristics and driving profiles of the driving cycles of the working vehicles used. For the assumed vehicle, the driving cycles represented relatively low load and therefore average gearbox input torque requirements on over the whole cycle. The gearbox was thus predominantly operated in areas of low input torque. The second reason may be the internal components of the gearbox are from a conventional gearbox Skoda Auto MQ200. It is likely that the gears in the gearbox are optimised predominantly for higher input torque at which the components may have the potential to achieve higher efficiencies. Both aspects can be indicated by the maximum efficiency values achieved during our measurements, with values above 90 % in areas with higher input torque values.

The low gearbox efficiency values achieved undoubtedly raise the question of the cause of such low values. We assume that detailed measurements of individual gearbox subsystems will be required to obtain an adequate answer. Nevertheless, it might still be feasible to estimate some approximation of the cause. The main sources of gearbox losses are generally oil churning, seal friction, gear mesh and bearing friction [13]. Generally, the losses can be split into two groups: load-dependent losses and load-independent losses [14]. Given the possible observation of significant changes in the achieved efficiency with changes in input torque, our results could suggest that some of the load-dependent losses are more likely to cause the relatively low gearbox efficiency achieved. Among the main sources of gearbox losses mentioned above, gear mesh and bearing friction could be considered.

From the resulting instantaneous power loss values, we conclude that the efficiency of electric vehicle gearbox can have a significant effect on the energy consumption results of electric powertrains measured during driving cycles. This is particularly the case in situations similar to the one we measured for the electric utility vehicle gearbox on drive cycles for specialized work vehicles, in which the vehicles moved at very low speeds with low loads.

By comparing the instantaneous gearbox efficiency curves of the individual driving cycles, it can be observed that the instantaneous efficiency measurements during driving cycles required frequent dynamic changes in input torque (in our case, CARB–HHDDT–CS and NREL–PDCQC cycles).

Consequently, the resulting gearbox efficiency data show oscillations and are challenging to evaluate adequately. In contrast, the results from the driving cycle CBD–SoTCODC, which has a relatively straightforward speed and input torque profile, were relatively coherent and more beneficial.

4 CONCLUSIONS

We asked the question of how the efficiency of the gearbox could be directly measured during driving cycles and considered how gearbox efficiency affects the energy consumption of electric vehicles during these driving cycles. We investigated the effect of gearbox efficiency on the driving cycle through experimental measurements in a dedicated powertrain laboratory using a gearbox designed for an autonomous electric utility vehicle.

In conclusion, the method of measuring gearbox efficiency on the driving cycles presented here provide an adequate basis for further research and development of electric vehicle gearboxes. Moreover, the resulting instantaneous gearbox efficiency values of driving cycles presented for specialized work vehicles, which can often be operated at low speeds with low loads, highlight the perhaps sometimes overlooked fact that in certain cases, it is the gearbox efficiency component which can have a significant effect on the overall efficiency of an electric powertrain, and as a consequence, on the energy consumption of electric vehicles. This method can be further developed with more detailed efficiency measurements based on input speed and input torque and parallel design modifications in the gearbox to maximize its efficiency during the driving cycles of specialized electric utility vehicles, with consequent results of lower energy consumption in real working operation.

5 ACKNOWLEDGEMENTS

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