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Virtual Analysis and Optimization of Fuel Consumption for Diesel-Powered Buses

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Research Article

Abstract

In this study, fuel consumption test cycles and commonly used SORT cycles for buses were discussed. An exemplary bus model was extensively modeled in terms of power systems. In addition to main components such as the engine, transmission, torque converter, and axle, the drags of engine-powered components such as alternators, engine cooling fan, and air conditioning compressor were integrated into the powertrain model to ensure that the calculated consumption comes as close as possible to the real-life values. The tire model, which determines the quality standard in vehicle simulation, was also discussed in detail. During the modeling of these components, necessary parameters were obtained through analyses and tests conducted on the sample vehicle using the CAN bus and added sensors.

After the completion of the model, real-life tests were conducted to validate the virtual analysis results. Once the model was validated, virtual studies continued to reduce the vehicle's consumption. Particularly during these studies, reducing consumption and enhancing performance through automatic transmission optimization were emphasized. When virtual models validated with SORT simulations are optimized with route-specific virtual analysis, public vehicles will minimize environmental impacts by reducing carbon emissions and at the same time perform more efficiently.

Keywords: Fuel consumption, Fuel Economy, SORT, Performance, Simulation, Test, Optimization

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1. Introduction

Fuel consumption is the most important parameter in transportation. Transportation service companies and bus manufacturers are working to decrease fuel consumption.

They are trying to use alternative fuel options that could reduce operation costs. CNG, LNG, hydrogen, and battery electric are some alternatives. CNG buses consume 23-24% more fuel than diesel buses under similar operating conditions but have a 52% lower fuel cost, making them more environmentally friendly and cost-effective [1]. Also, passenger load, road grade, and congestion levels significantly impact fuel consumption and emissions in urban buses, with CNG buses showing a 60% lower NOx emission rate than diesel buses [2].

Other parameters affect fuel consumption. Service routes, vehicle models, average travel speed, and loaded weight significantly influence fuel efficiency in public buses, with driver behavior and daily schedules also playing a role [3,4].

In the automotive industry, fuel consumption tests for buses and public transportation are optimized with simulations, providing advantages in terms of cost and time. The traditional method needs transporting of the vehicle, fuel consumption, and engineering time for the test. So, it can be time-consuming and costly. This situation brings about the necessity of working with more limited data by limiting the number of test iterations. Fuel consumption simulations decrease the costs by moving these processes to the virtual environment and allow obtaining more comprehensive data by increasing test iterations. For these reasons, it was decided to work on the optimization of vehicle parameters by calculating the fuel consumption of the vehicle with 1D Simulations.

There are established procedures and equipment to test the fuel consumption of diesel buses. The fuel consumption of buses is usually tested using specific procedures to provide accurate and comparable data for public transport buses produced by different manufacturers. SAE (American Society of Automotive Engineers) has a standard test procedure called J1526 that can be used to evaluate the fuel consumption of different vehicles [4]. The Altoona Bus Testing and Research Center conducts fuel economy tests to achieve this goal [7, 8]. Large cities have established their testing procedures apart from these general procedures. Orange County Bus (OC Bus) Cycle (Figure 1) and Manhattan Driving Cycle (Figure 2) are some of these procedures [6]. These tests involve the use of a suitable operating cycle to measure fuel consumption and are performed in a controlled environment using specific testing equipment, facilities, and personnel [7].

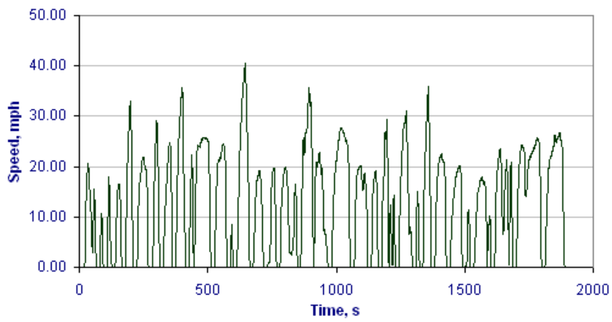

Fig.1. Orange County Bus Cycle [9]

OC Bus Cycle is a dynamometer test for heavy commercial vehicles. It was developed by West Virginia University (WVU) based on the driving patterns of urban public transportation buses in Los Angeles, California. The OC Bus Cycle should not be confused with the Orange County Garbage Transportation Cycle, also developed by WVU. Vehicle speed during the cycle period is shown in Figure 1 [9].

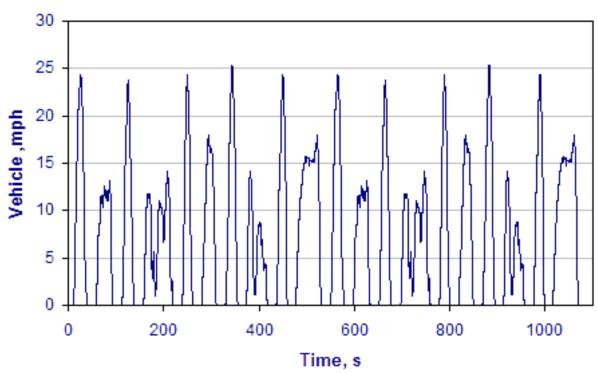

Fig. 2. Manhattan Driving Cycle [10]

Manhattan Bus Cycle is a dynamometer test for city buses. This cycle developed based on observed real-life driving patterns of urban transit buses in New York City's midtown Manhattan. Frequent stops and very low speeds characterize the cycle. Vehicle speed during the Manhattan driving cycle is shown in Figure 2. Some parameters of the cycle are shown in Table 1.

Table 1. Selected parameters of the Manhattan Test Cycle [10]

Duration (s)	1089
Maximum Speed (km/h)	40.88
Average Speed (km/h)	11

SORT cycles, the most widely used protocol other than these cycles, are an initiative of the International Union of Public Transport (UITP) and aim to standardize the measurement of energy consumption and performance of buses in urban public transportation systems [11]. This protocol, developing since 2004, is designed to compare the energy consumption of buses and to encourage the development of more efficient and attractive public transport systems [11].

In these cycles, parameters such as speed, acceleration, duration, and distance are predetermined, and the vehicle is expected to provide these parameters as much as possible. This article's tests and virtual analyses were performed according to this protocol.

This protocol consists of three cycles: SORT 1 Heavy Urban (Figure 3), SORT 2 Easy Urban (Figure 4), and SORT 3 Easy Suburban (Figure 5).

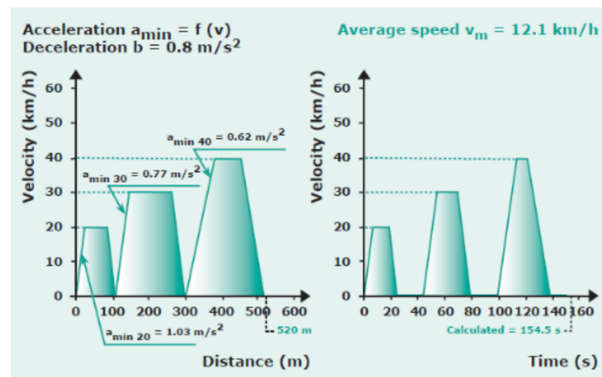

Fig. 3. SORT 1 Cycle - Heavy Urban [11]

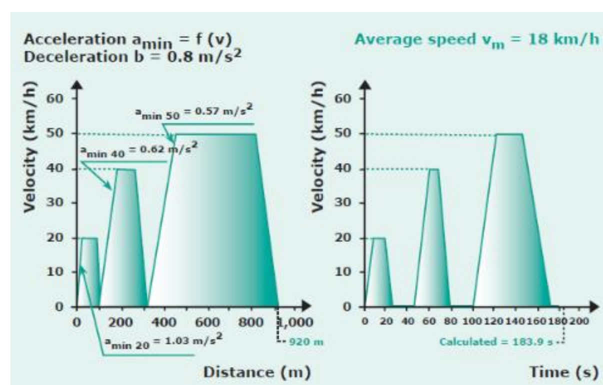

Fig. 4. SORT 2 Cycle - Easy Urban [11]

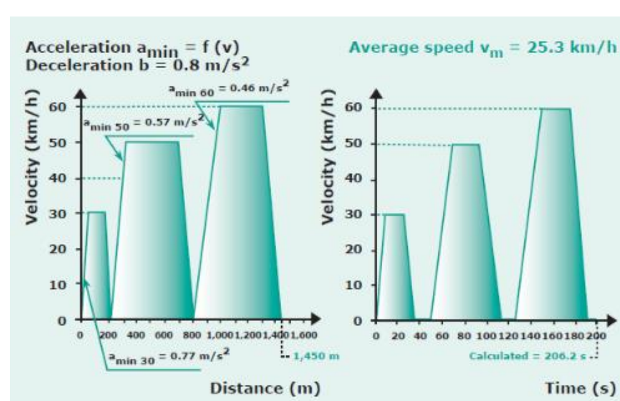

Fig. 5. SORT 3 Cycle - Easy Suburban [11]

2. Methodology

2.1. Vehicle Model

The 1D simulation model used (Figure 6) consists of sub-models that work together, such as control strategy, control units, and electromechanical components. PTControl contains the overall drive control strategy and manages the electrical and mechanical energy distribution of the drive system. Each control unit controls one type of electromechanical component to ensure the energy distribution planned by PTControl.

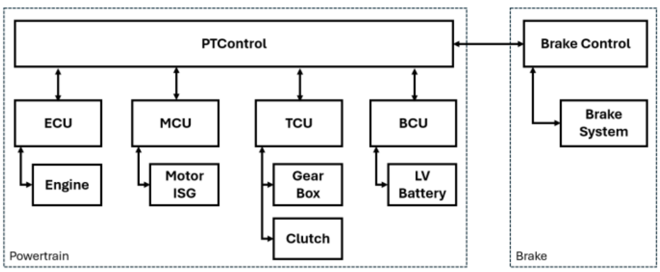

Fig. 6. Structure of Powertrain Model

A 12m Low Floor bus was modeled for analysis. The Powertrain model (Figure 7) is based on a diesel engine and automatic transmission with a torque converter. Components are modeled with 1D-2D Look-up tables.

Table 1. Powertrain details of Bus

Engine Type	6 Cylinder Inline Diesel
Engine Displacement	6.7L
Emission	Euro 6
Transmission	6 Speed Automatic
Clutch	Torque Converter

2.1.1. Diesel engine model and fuel consumption calculations

The diesel engine model uses look-up tables to describe the torque characteristic produced depending on engine speed and load.

Dyno measurements shared by the engine supplier are defined in these tables. These measurements give the maximum torque corresponding to the relevant rpm. The torque produced according to the accelerator pedal position is determined by interpolation from this table and used in the model.

To make the engine power as accurate as possible, instantaneous engine power and torque are calculated regarding ambient conditions. The calculation of torque according to ambient temperature and pressure is shown in Eq. 1.

$$Trq_{Eng}^{corr} = fac(T_E\,,\,P_E) \cdot Trq_{Eng} \tag{1}$$

The calculation of the correlation factor is given in Eq.2.

$$fac(T_E\;,P_E) = \frac{1}{\left(\frac{P_{ref}}{P_E}\right)^{C_P} \cdot \left(\frac{T_E}{T_{ref}}\right)^{C_T}} \tag{2}$$

Fuel consumption calculations were made using the BSFC Map in 2D look-up tables (Figure 7). The x-axis is the rotation speed of the diesel engine, and the y-axis is the torque output.

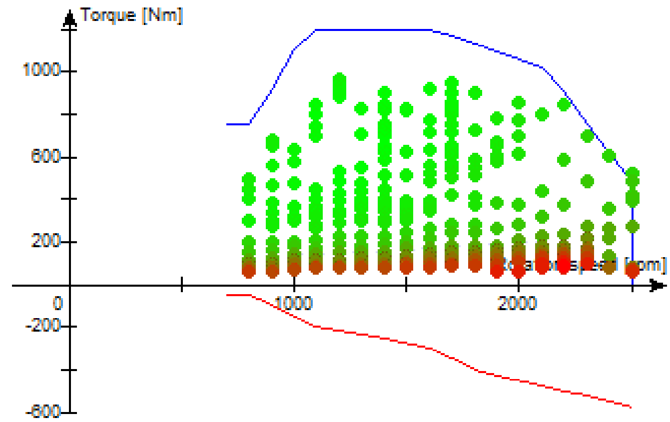

Fig. 7. BSFC Map of diesel engine.

Fuel consumption was calculated depending on fuel density, specific mass flow rate, and engine output power (Eq. (3)). The absolute value of engine power was used to ensure positive volume flow in the case of negative engine torque.

$$\dot{v}_F = \frac{\dot{m}_F^{spec}(\omega_{Eng}, Trq_{Eng}) \cdot \; |P_{Eng}|}{\delta_F \cdot 3.6 \, \cdot \, 10^9} \tag{3}$$

During SORT tests, there are situations where the engine is running at idle while the vehicle is in a static state. To calculate the consumption in this case, lines of code written in C were added to the program. To determine the nominal friction losses, the engine was run at idle for a while and CAN recording was taken with the Vector device (Figure 8) during this period. This record was created by interpreting the CANalyzer file and the program code was added.

Fig. 8. Vector CAN Interface device.

2.1.2. Clutch model

The hydrodynamic or Föttinger torque converter increases torque by reducing the rotation speed. The input and output torque of the converter are calculated depending on the converter factor and input speed.

$$T_{In} = k_{In} \cdot \omega_{In}^2 \quad (4)$$

$$T_{Out} = k_{Out} \cdot \omega_{Out}^2 \quad (5)$$

$$k_{In}, k_{Out} = f(s) \quad (6)$$

$$s = \frac{\omega_{Out}}{\omega_{In}} \quad (7)$$

The connection between the input and output torque of the converter is described by the torque ratio and is shown in Eq. (8).

$$\mu = \frac{T_{Out}}{T_{In}} = \frac{k_{Out}}{k_{In}} \cdot s^2 \quad (8)$$

The transmission behavior of the converter can be modeled using the converter factor and torque ratio. To examine the behavior of the transmission against torque and rotation speed, acceleration tests were performed on a vehicle with the same transmission and torque converter. During the test, recordings were made by connecting the Testman device (Figure 9) to the transmission control unit.

Fig. 9. Testman device.

These datasets were interpreted, and the following graphs (Figure 10, Figure 11) were created.

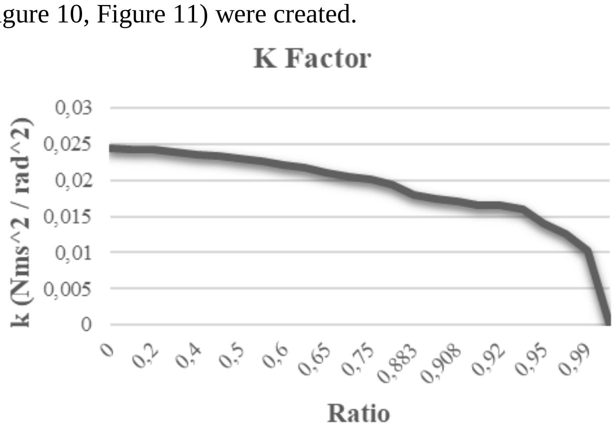

Fig. 10. Converter K Factor parameters.

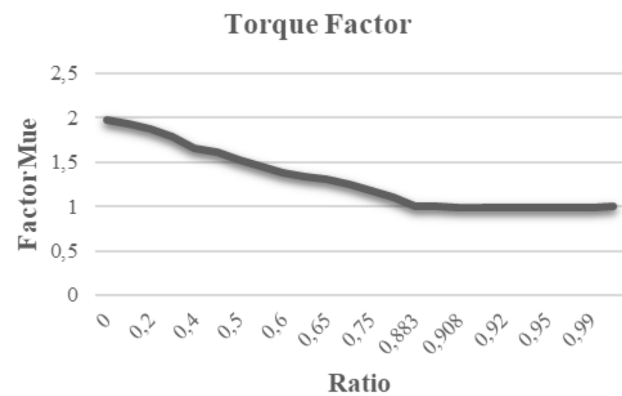

Fig. 11. Converter Torque Factor parameters.

2.1.3. Transmission and TCU model

While defining the transmission in the vehicle model, a simple reducer was simulated using a 1D look-up table. Gears, transmission ratios, and transmission efficiencies are listed in the table. The parameter of transmission efficiency coefficient acquired from the study of Irimescu et al was used [12].

Table 2. Transmission gear ratios and efficiency coefficients.

Gear No	Ratio	Efficiency
1	3.364	0.96
2	1.909	0.96
3	1.421	0.96
4	1	0.96
5	0.72	0.96
6	0.615	0.96
R	4.235	0.96

The TCU model (Figure 10) controls the converter and lockup clutches according to the torque transmitted from the engine, the torque requested by the driver, and the speed, and orders the transmission to shift to the appropriate gear. The TCU determines the requested torque based on the accelerator pedal position and vehicle mass.

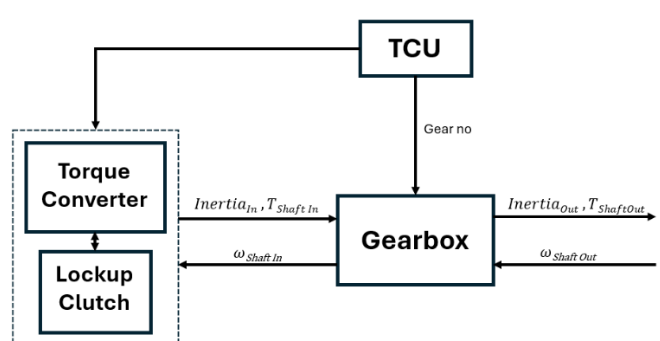

Fig. 12. Transmission Control Unit model.

The torque and rotation speed coming out of the transmission are calculated by the differential ratio and transferred to the wheels.

2.1.4. Tire and wheel model

The quality of a vehicle model is often evaluated by the quality of its tire model. The task of the Contact Point Interface based tire model is to calculate tire forces and torques at the tire-road contact point. So, this model only describes the torques at the contact point between the tire and the road.

Tire loads are calculated in the vehicle model and used as input in the tire model. The sideslip angle and longitudinal slip need to be calculated with this tire model. This needs to be related to an equation like the one below.

$$\alpha, s = f\left(\overline{v}^{(P)}, v_{Belt}\right) \text{ with } v_{Belt} = f\left(\Omega_{Rim}, R_{Belt_{eff}}\right) \quad (9)$$

The simulation uses the following definition for longitudinal slip calculation.

$$s = \frac{\Omega_{Rim} \cdot R_{Belt_{eff}} - v_x^{(P)}}{v_x^{(P)}} \quad (10)$$

To also calculate backward movement, the formulation has been modified as follows.

$$s = \frac{\Omega_{Rim} \cdot R_{Belt_{eff}} - v_x^{(P)}}{\left|v_x^{(P)}\right|} \quad (11)$$

In this equation, F_x and longitudinal slips have the same sign. When the sign is positive, the vehicle is pushed forward, and when the sign is negative, the vehicle is pushed backward. The following formula is used for slip angle calculation.

$$\alpha = atan \frac{v_y^{(P)}}{\left|v_x^{(P)}\right|} \quad (12)$$

The turn slip calculation is as follows.

$$\varphi_t = \frac{\psi_{Rim}}{\left|v_x^{(P)}\right|} \quad (13)$$

2.2. Virtual Analyses Test Model

To perform the test in a virtual environment, a smooth and sufficiently long asphalt road was defined. As stated in UITP's SORT standard, checkpoints were created at certain intervals, and speed trapezes (Table. 3.) were formed at these points, as specified in the relevant SORT document, and the waiting time specified in the relevant cycle after each trapeze was entered as an additional parameter to the driver.

Table 3. Design of the 5 SORT Trapezes. [10]

V (km/h)	Distance Between of Cone 1-4 (m)	Position of Cone 2 (m)	Position of Cone 3 (m)	Distance of Const. Speed Cone 2-3 (m)
20	100	15	80.7	65.7
30	200	45	156.6	111.6
40	220	100	142.8	42.8
50	600	170	479.4	309.4
60	650	300	476.4	176.4

The results of virtual analyses are shown in Table 4.

Table 4. Virtual analyses test results.

Cycle of SORT	Fuel Consumption (L/100km)
1	42.923
2	35.788
3	32.148

2.3. Real-Life Tests

Like virtual analysis tests, real-life tests were conducted according to UITP's SORT standards. While making measurements, Racelogic Vbox 3i telemetry device (Figure 13) and Kistler Fuel Flow Meter (Figure 14) were used.

Fig. 13. Racelogic Vbox 3i.

Fig. 14. Kistler Fuel Flow Meter.

The tests were applied on dry asphalt and a maximum of 1% longitudinal grade. To eliminate the wind and grade effect tests were applied in two directions on the same road. The results are averages of the two directions of the same cycle and they are shown in Table 5.

Table 5. Real-life test results.

Cycle of SORT	Fuel Consumption (L/100km)
1	43
2	36
3	32

2.4. Optimization Study

The optimization study started after the virtual model to validated by real-life tests. The SORT 1 cycle repeated from 1500 to 1000 engine rpm on the virtual analyses. As a result, a decrease in average speed and fuel consumption to observed.

The correlation between reducing fuel consumption and engine upshift speed decreased significantly below 1200 engine RPM. The transfer rate changed to achieve the required performance because the drive of accessories, such as the alternator, is made on the engine crankshaft. For this reason, accessories power losses increased when the engine rpm decreased. Optimum performance achieved taking into consideration these results at 1250 engine rpm.

3. Conclusions

In this study, exemplary bus powertrain components were modeled for virtual analyses, and fuel consumption and performance were calculated. After the modeling real-life tests were applied to validate the virtual model. Optimization was applied to reduce fuel consumption and better performance when the virtual model was validated.

As a result of the optimization study carried out to reduce fuel consumption, it was observed that fuel consumption decreased as the engine speed at which the gear was changed decreased. However, when we lowered this value below a certain level, serious losses occurred in the vehicle's acceleration ability and final speed. For this reason, reducing the speed can be done enough to provide the minimum acceleration values specified in the SORT tests. As the speed is reduced, the climbing ability will also decrease, but since the slope parameter was fixed in the SORT tests during optimization, the climbing ability effect was ignored.

When virtual models validated with SORT simulations are optimized with route-specific virtual analysis, public vehicles will minimize environmental impacts by reducing carbon emissions and at the same time perform more efficiently. This method enables rapid iterations in the design process, optimizing the vehicle development process and ensuring more efficient use of resources.

Nomenclature

Trq_{Eng}	:	Engine Torque (Nm)
T_E	:	Environment Air Temperature (K)
P_E	:	Environment Air Pressure (mbar)
C_T	:	Temperature correction coefficient
C_P	:	Pressure correction coefficient
T_{ref}	:	Reference temperature (K)
P_{ref}	:	Reference pressure (bar)
$\dot{v}_F$	:	Volumetric fuel flow (m^3/s)
$\dot{m}_F$	:	mass flow (kg/s)
ω_{Eng}	:	Engine rotation speed (rad/s)
P_{Eng}	:	Engine power (kW)
δ_F	:	Fuel density (kg/m^3)
T_{In}	:	Torque Inlet (Nm)
T_{Out}	:	Torque Outlet (Nm)
ω_{In}	:	Rotation speed inlet (rad/s)
ω_{Out}	:	Rotation speed outlet (rad/s)
k_{In}	:	k factor inlet
k_{Out}	:	k factor outlet
μ	:	Torque factor
$\vec{v}^{(P)}$	:	Velocity vector in the tire-road contact point (m/s)
Ω_{Rim}	:	Rim rotation speed (rad/s)
v_{Belt}	:	Velocity of the tire belt in the tire-road contact point (m/s)
$R_{Belt_{eff}}$	:	Effective rolling radius (m)
φ_t	:	Turn slip (1/m)
α	:	Side slip angle (rad)

Conflict of Interest Statement

The authors declare that there is no conflict of interest in the study.

CRediT Author Statement

Ahmet Murat Yılmazlar: Conceptualization, Writing-original draft, Validation, Data curation, Formal analysis
İmdat Taymaz: Supervision, Editing

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