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Simulation of 8x8 Heavy Duty Truck for Evaluating Effects of Torque Converter Characteristics on Vehicle Performance

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1. Introduction

Heavy duty trucks are frequently used to serve in transportation, military or fire extinction missions. Last few decades have witnessed the increasing attention to technologies of heavy duty vehicle powertrain systems. These technologies are focused on both conventional powertrain [1] and hybridization [2] of conventional vehicles. Powertrain system modeling and simulation of conventional heavy duty trucks have the potential to improve the existing system performance. The main goal of modeling and simulation of powertrain is to improve the traction performance and fuel economy by redefining each system component’s properties and detecting its suitability of vehicle operating conditions.

Mathematical model of vehicle powertrain system can have carried out by dynamic [3] or quasi static [4,5] approach based on longitudinal vehicle dynamics [6,7,8] equations. When a part of system is interested for the determination of dimensioning or strength, the powertrain model can be arranged for one or a couple components of all system [9]. But the calculation of efficiency (overall), consumption (energy, fuel) and performance (traction, acceleration) of vehicle are needed to set up full model of powertrain system. For instance, analysis of vehicle and powertrain dynamics with an automatic gearbox was implemented [9] for the determination of vehicle velocity and engine speed versus time. Speed and moment requirements of automatic gearbox components such as sun gear and pinion gear are determined the analysis either. Energy efficiency analyses of a vehicle was simulated [10] by using powertrain model for all system according to vehicle velocity profile named drive cycle. Tao et al. [11] focused on automatic transmission model and applied evaluation models for heavy duty vehicle performance. Meng et al. [12] investigated the control of automatic transmission by using engine to wheel powertrain model for heavy duty vehicle. Input parameters are characteristic map of engine and hydrodynamic torque converter characteristics.

8x8 heavy duty truck has two power module at the rear side of vehicle including diesel engine, hydrodynamic torque converter and automatic transmission. The hydrodynamic torque converter is a major component among of all which affects driving performance [10]. Required power is transferred from engines to transmissions and engine torque is also increased by torque converter. The increasement can arrange according to engine operational area. For better vehicle performance and fuel economy, torque converter parameters are determined correctly. Correct torque converter parameter determination means greater operational area of the converter curves inside the engine torque curve. To show the effects of torque converter, two different designs that have different operational area, are integrated into powertrain model individually. The study compares the results of traction performances according to a drive cycle.

Drive cycles can be utilized to predict vehicle performance, fuel consumption and exhaust emissions. New European Drive Cycle (NEDC), Urban Dynamometer Driving Schedule (UDDS) and Federal Test Procedure (FTP-75) are used as standard drive cycle for reaching and testing legislative boundaries about vehicle design [10,5]. Heavy Duty Urban Dynamometer Driving Schedule (HD-UDDS) is developed for heavy duty vehicles. The load on the wheels are calculated to simulate driving resistance on the vehicle such as rolling, aerodynamic and acceleration resistance and powertrain simulations are performed on HD-UDDS cycle. Essentially, this paper is aimed to investigate the effects of torque converter characteristic on the result of vehicle performance and fuel consumption by the drive cycle analysis of the 8x8 heavy commercial vehicle powertrain.

2. Powertrain of 8x8 heavy duty vehicle

The powertrain components have been consisted on two diesel engines, two automatic gearboxes with torque converters, drive shafts, summation box, transfer box, four differential and eight reduction units in axles which are shown in Fig. 1. Each diesel engine has maximum 425 kW power at 1600 rpm and 2800 Nm torque at 1100 rpm. Seven speed automatic gearbox is located after hydrodynamic torque converter. Output drive shafts from the automatic gearboxes are connected with each other into the summation box. Propulsion is transferred after summation box via drive shaft to a transfer case and it transmits the torque to differentials. Differential distributes torque to the wheels and planetary system inside the wheel hub increases it as the last reduction from engine to tire road contact.

2.1. Torque converter alternatives for powertrain

Torque converter assembly is composed of three major components [10]. Impeller is the input element of converter which is attached directly to the engine and turns at engine speed. A stator changes the direction of flow inside the torque converter and torque is transferred to the turbine as a result of the induced flow from the impeller and stator [13]. Characteristic of torque transferring depends on dimensions of each component, shape of blades and volume of torque converter.

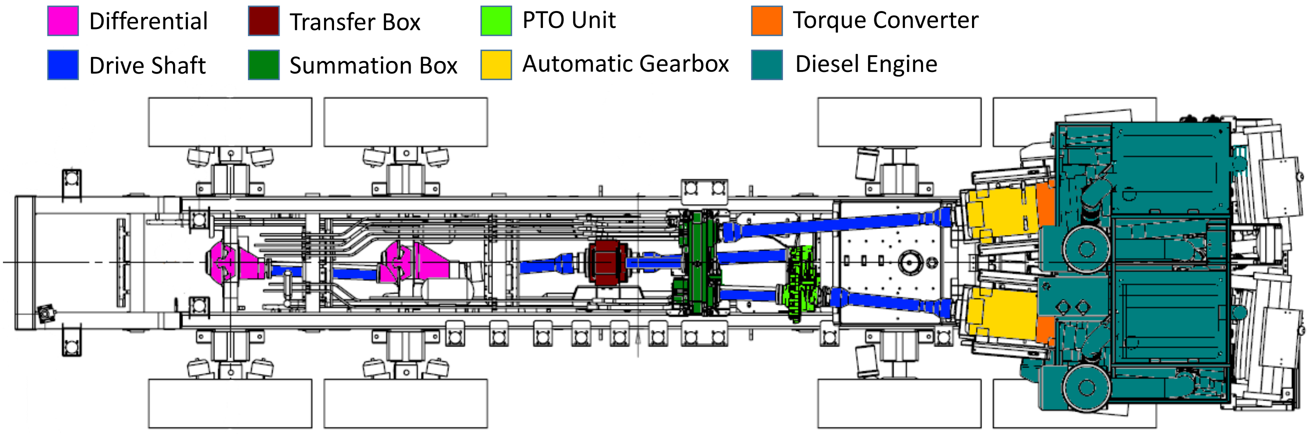


Fig. 1 8x8 Heavy duty vehicle powertrain

Different transferring characteristics of torque converter effect the torque ratio i_m and capacity factor k. Both of them are the function of torque converter speed ratio i_n. Capacity factor is also depending on fluid density, torque coefficient and effective diameter of torque converter. Torque ratio, speed ratio and impeller test moment $M_{P,test}$ are achieved by constant test speed $n_{p,test}$. Constant test speed was arranged at 1000 rpm in the test bench. Capacity factor is described by the function of speed ratio as:

$$k\left(i_n\right) = \frac{M_{P,test}}{n^2_{p,test}}. \tag{1}$$

Impeller moment M_P, which is called as the input torque of the torque converter, is changed by impeller speed (n_p), which is called as input speed of the torque converter. The equation is given as:

$$M_P = k\left(i_n\right) n_p^2. \tag{2}$$

The steady-state turbine moment-output moment of torque converter M_T and turbine speed-output speed of torque converter n_T can be presented as follows:

$$M_T = M_P i_m, \tag{3}$$

$$n_T = n_P i_n. \tag{4}$$

Output torque M_T is transferred at output speed n_T of torque converter from an engine to the automatic gearbox unit smoothly. The torque converter unit is also multiples the input torque especially in low engine speeds according to its characteristics of capacity factor and torque transfer ratio. Therefore, two different type hydrodynamic torque converters are integrated into powertrain model separately for comparison of the performance results of the vehicle. Torque converter alternatives called Design 1 and Design 2, have different torque ratio i_m and capacity factor k. The characteristics of the two design options are shown in Fig. 2 as function of speed ratio i_n of torque converter. The main differences of converter alternatives are explained as the Design 2 has lower torque ratio (1.673) than Design 1 (1.829), however, test moment of impeller and capacity factor are higher in Design 2 torque converter.

Experimental impeller speed and numerically calculated impeller speed (Eq. (3)) are compared for the torque converter of Design 1 in Fig. 3. Speed characteristics of both numerical and experimental results show progressive form at increasing turbine speed at the range of 2200 rpm and 1700 rpm. This comparison for Design 1 torque converter is considered as verification of moment transfer from engine to gearbox.

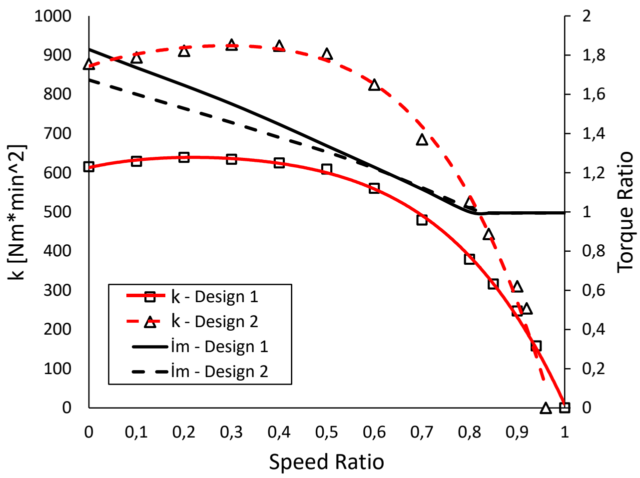


Fig. 2 Torque converter characteristics of two different types (Design1 and Design 2)

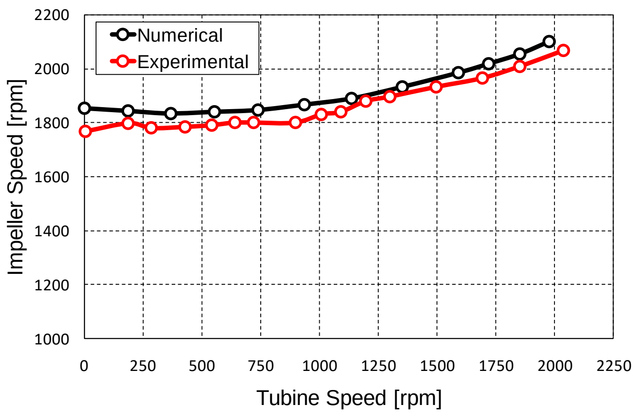


Fig. 3 Comparison of impeller speeds at design 1

Design 1 and Design 2 are caused different moment output from torque converter although same engine are using at the powertrain system. Moment outputs at torque converter of both designs are calculated in Eqs. (3-4). Output moments are compared in Fig. 4 for both Design 1 and Design 2 alternatives. The comparison shows clearly that the output moment is higher in Design 2 than Design 1 because of the difference of their torque ratio (i_m).

Torque ratio and capacity factor are affected on operational impeller moment inside the engine moment curves also. Engine operational area are restricted with impeller moments at maximum and minimum speed ratio (i_n=0 and i_n=1). Engine torque-speed characteristic and operational area of torque converter designs are shown in Fig. 5. Moment requirements that caused from tire road interactions are supplied by restricted area (black for design 1 and red for design 2).

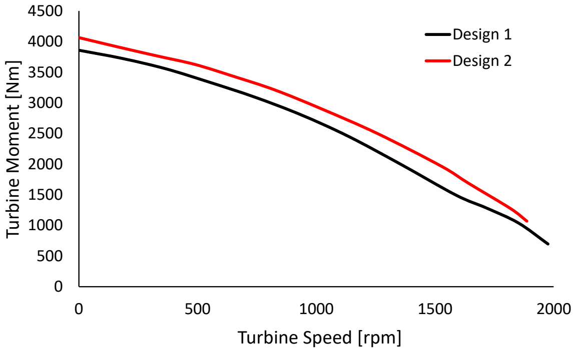


Fig. 4 Output moments of torque converter designs

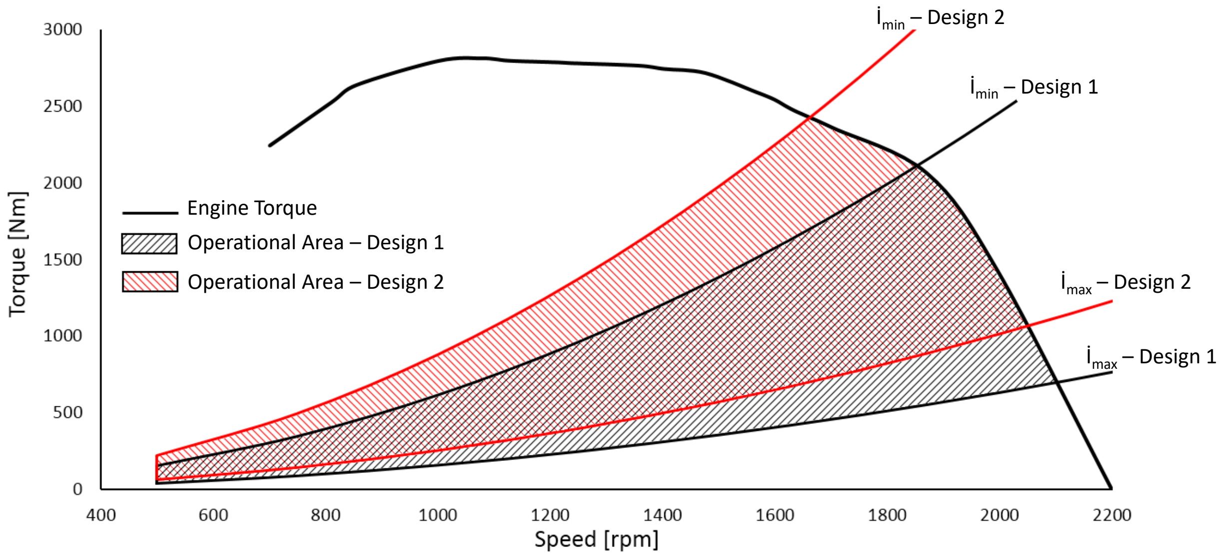


Fig. 5 Operational area of torque converter designs inside engine torque curves

2.2. Longitudinal driveline dynamics of vehicle

The modelling of powertrain is especially useful for determination of performance characteristics such as grading and accelerating, but they are strongly depending on the measurements for experimental verification [3]. An intermediate model which includes comprehensive theoretical model and simulation on a drive cycle without experiment is preferred in heavy duty vehicle industry because of financial anxiety. The powertrain model is presented in this paper, based on the following assumptions:

- The model taking into account the longitudinal motion only.
- Slipping of tire is ignored and adhesive coefficient of the tire-ground contact assume sufficient for traction and barking.
- Each element is regarded as discrete system.
- System vibration and damping are ignored.
- Dynamic tire radius assumed constant.

Engines of the vehicle have to produce required power in order to overcome resistance forces and losses of propulsion system for the continuity of motion. Resistance forces consist of rolling resistance force F_R and air resistance forces F_L in no grade road conditions. Additionally, with vehicle move up grade, grading resistance F_{St} and under acceleration conditions, acceleration force F_a occurs. Those resistance forces can be expressed by:

$$F_R = m\ g\ \ f_R\ , \qquad (5)$$

$$F_{St} = m\ g\ sin\,\delta\ , \qquad (6)$$

$$F_L = \frac{1}{2}\,\rho\ A\,C_W\ V^2\ , \qquad (7)$$

$$F_a = \lambda\ m\ a\ , \qquad (8)$$

where m is total mass of vehicle, g is the gravity of acceleration, f_R is the rolling resistant coefficient, δ is the slope angle of the road, ρ is the density of air, C_W is aerodynamic coefficient of the vehicle, A is projection area of vehicle, V is the velocity of vehicle, λ is the coefficient of the effect of rotating masses and a is acceleration of vehicle.

In two power module (engine-torque converter-gearbox) turbine moment (Eq. (3)) is transferred to gearbox depends on drive ratio τ_{GB} at the same time for maximum traction effort. Output moments of gearbox M_{GB} are summed inside the summation box. Summation box principle is shown in Fig. 6. Gearbox output shafts connected the gears with speeds of n_{GB1} and n_{GB2} and moments of M_{GB1} and M_{GB2} (red in Fig. 6) which is gear teeth numbers represented as z_1 and z_2. Gear ratio of summation box τ_{SB} is represented in Eqn. 9.

$$\tau_{SB} = \frac{z_3}{z_1} = \frac{z_3}{z_2}\ , \qquad (9)$$

where z_3 is the output gear teeth (green in Fig. 6). At the output speed of n_{SB}, expression of output moment of summation box M_{SB}:

$$M_{SB} = 2\ \tau_{SB}\ M_{GB}\ . \qquad (10)$$

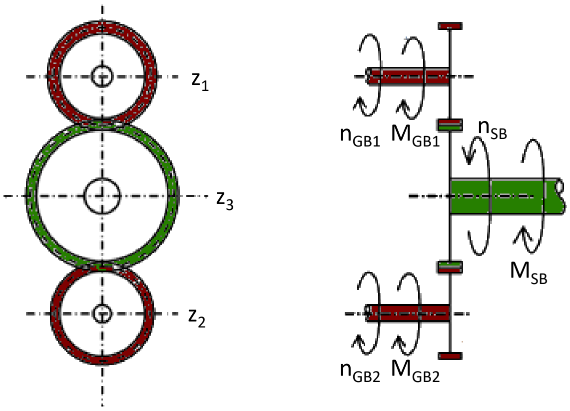


Fig. 6 Summation box principle (red; input gears, green; output gears)

Vehicle performance is determined by traction force diagram which is calculated torque and speed transfer from the engine to wheels. Using with torque converter output moment M_T, at various gear stages (represented by subscript i) moments on traction tires F_{Ti} are defined as:

$$F_{T_i} = \frac{M_T \ \tau_{GB_i} \ \tau_{SB} \ \tau_{TB} \ \tau_A \ \eta_T}{r_{dyn}}, \tag{11}$$

where τ_{GB} is the ratio of transfer case, τ_A is the ratio in axles (including differential ratio and reduction on wheel hub) and η_T is the overall mechanical efficiency of powertrain model. Mechanical efficiency which is expressed in below includes gearbox η_{GB}, summation box η_{SB}, transfer case (η_{TB}) and reduction in the axles η_A. Parameters and coefficients about vehicle are shown in Table 1.

$$\eta_T = \eta_{GB} \ \eta_{SB} \ \eta_{TB} \ \eta_A. \tag{12}$$

Table 1

Parameters of vehicle

Symbol	Value	Unit
m (loaded)	43870	kg
m (unloaded)	25000	kg
g	9,81	m/s^2
ρ	1,228	kg/m^3
A	8	m^2
C_W	0,55	-
r_{dyn}	728,7	mm
η_T	0,86	-

Drive ratios of powertrain components and their mass moment of inertia are presented in Table 2. Speed and torque reduction are implemented in 7 speed transmission τ_{GBi}, summation box τ_{SB}, transfer box τ_T and inside of four driven axles τ_A (differentials and wheel hubs). Torque converter reduction ratio is defined in Fig. 2 before and it is not shown in Table 2. Driven axles have same gear ratio as 5.46.

Table 2

Drive ratios and mass moment of inertia of powertrain components

	Gearbox						
	1. Gear	2. Gear	3. Gear	4. Gear	5. Gear	6. Gear	7. Gear
Drive Ratio	5,6	3,43	2,01	1,42	1	0,83	0,59
Mass Moment of Inertia [kgm^2]	1,18	1,02	0,92	1,04	1,58	1,47	2,73
	Torque Converter	Summation Box	Transfer Box	Axles			
				1. Axle	2. Axle	3. Axle	4. Axle
Drive Ratio	-	1,06	0,97	5,46	5,46	5,46	5,46
Mass Moment of Inertia [kgm^2]	2,07	1,22	1,41	0,35	0,49	0,47	0,35

To complete defining the traction force curve, conversion of angular velocity of torque converter output n_T to vehicle speed at various gear stages V_i can be expressed by:

$$V_i = \frac{2 \ \pi \ n_T \ r_{dyn} \ 3.6}{60 \ \tau_{GB_i} \ \tau_T \ \tau_{SB} \ \tau_A}. \tag{13}$$

3. Longitudinal driveline dynamics of vehicle

Traction force curve of heavy duty vehicle are calculated by Eqs. (11) and (13) for seven speed gearbox. Maximum tractive force curve is determined by the assumptions of:

- Gear shifting times are depending on theoretically usage of maximum traction force.
- Gear shifting process is applied in pre-determined engine speed that is related to intersection point of traction curves in each gear numbers.

Maximum tractive force area for grading and accelerating is defined as the area between traction curves and ressistive forces on tires in straight motion. Rolling and air ressitance (Eqs. (5, 7)) forces are effected on tire in straight motion of vehicle. Vehicle can use the area as reserve forces for accelerating and climbing which are the most crucial performance characteristics of a vehicle. Fig. 7 shows the maximum traction forces for usage of design 1 torque converter, reserve forces area and ressistive forces. Maximum reachable speed of vehicle is also determined by finding the point of equal maximum force and ressistive force value (black and red line).

Acceleration capacity of vehicle is expressed by the derivation of speed by time. When reserve forces are used for accelerating the dynamic equilibrium of the acceleration is written

$$a = \frac{dV}{dt} = \frac{F_T - F_R - F_L}{m \ \lambda_i}, \tag{14}$$

where λ_i is the coefficient of the factor of rotating masses as mentioned before which is changed with gear shifting. The definition of λ_i is given by:

$$\lambda_i = 1 + \frac{\theta_{red_i}}{m \ r_{dyn}}, \tag{15}$$

where $\theta_{red\ i}$ is the reduction mass moment inertia of all powertrain components to tire rotating axis and r_{dyn} is the dynamic rolling radius of tire. With the consideration of all gear ratios and components, $\theta_{red\ i}$ is evaluated

$$\theta_{red_i} = 2\theta_E \tau_{GB}^2 \tau_{SB}^2 \tau_{TB}^2 \tau_A^2 + 2 \cdot \theta_{TC} \tau_{GB}^2 \tau_{SB}^2 \tau_{TB}^2 \tau_A^2 + \\ + 2\theta_{GB} \tau_{GB}^2 \tau_{SB}^2 \tau_{TB}^2 \tau_A^2 + \theta_{SB} \tau_{SB}^2 \tau_{TB}^2 \tau_A^2 + \\ + \theta_{TB} \tau_{TB}^2 \tau_A^2 + \theta_A + 8\theta_R, \tag{16}$$

where θ_E, θ_{TC}, θ_{GB}, θ_{SB}, θ_{TB} and θ_R are presented the mass moment of inertia of engine, torque converter, gearbox, summation box, transfer box and tires respectively. θ_R is taken 81,9 kgm^2 for each tire. The values of gear ratios and mass moment inertias are presented in Table 2.

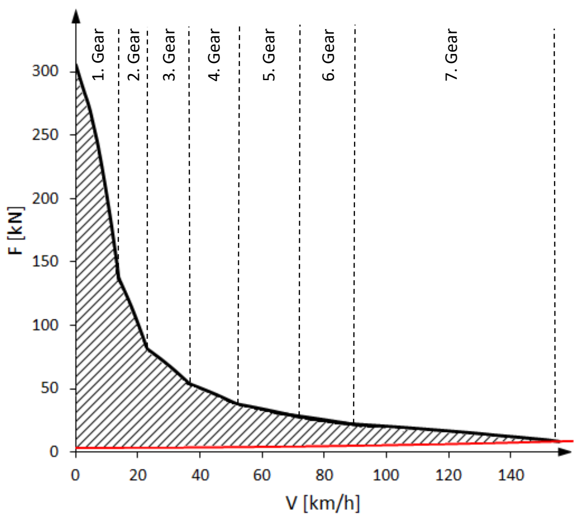


Fig. 7 Reserve forces on traction tires (black; maximum traction force curve, red; ressistive forces-F_R and F_L)

Effect of the integration design 1 and design 2 type hydrodynamic torque converters to powertrain model on accelerations of vehicle are calculated by Eq. (14). Acceleration capacities of the vehicle for loaded and unloaded operating conditions in various vehicle speeds are presented in Fig. 8. Results are clearly showed that acceleration capacity of unloaded operation is always higher. Torque converter type design 2 yields better performance on vehicle acceleration.

Acceleration time results of the heavy duty vehicle are calculated by using reserve forces and arranged by Eq. (14). Times for acceleration of the vehicle are expressed as:

$$t = \int_0^{V_{max}} \frac{m\lambda_i V}{F_T - F_R - F_L} dV, \tag{17}$$

where V_{max} is the maximum velocity of vehicle that is determined with solution of maximum traction force and resistance force curves illustrated in Fig. 7, F_T is the maximum traction force of vehicle which is shown as example for torque converter of design 1. F_R represents rolling resistance force and F_L is the air resistance force. Results are shown in Fig. 9 for both torque converter types and operating load conditions. Difference of acceleration times between the usages of torque converter types are clear especially in loaded operation. Depending on the exerting torque from design 1 and 2 types of torque converters, accelerating time is higher with using of Design 2 than Design 1.

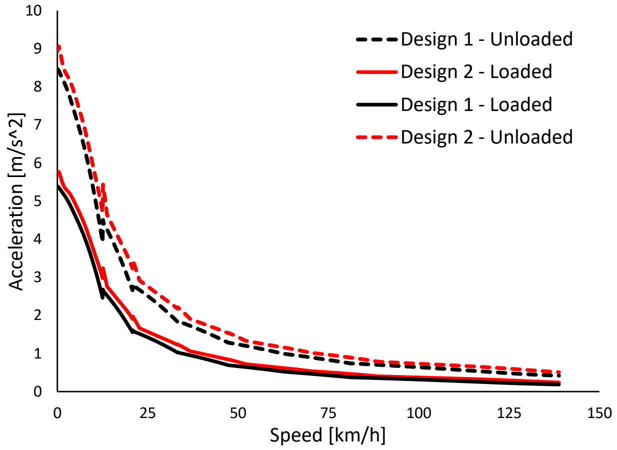


Fig. 8 Acceleration performance results of the vehicle

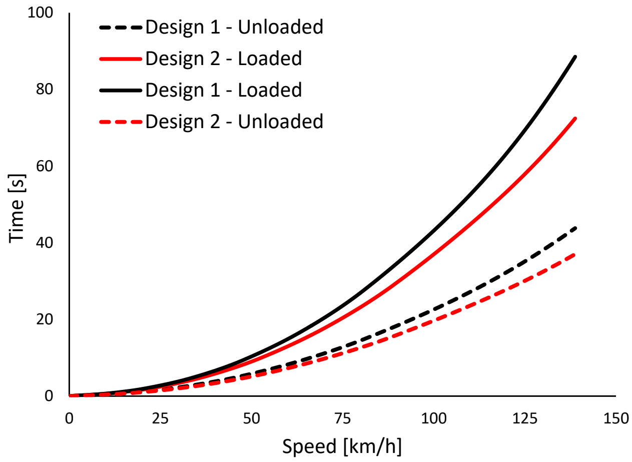
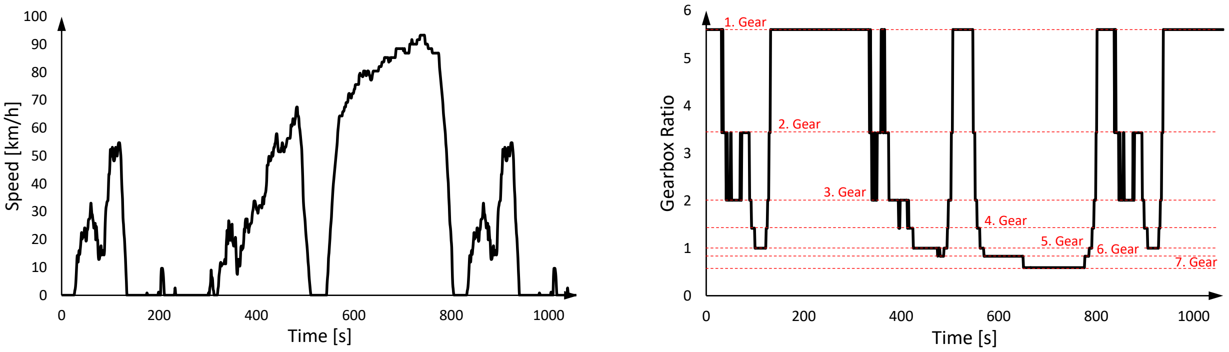


Fig. 9 Acceleration times of the vehicle

4. Drive cycle simulation and discussions

HD-UDDS drive cycle is as standardised driving pattern, which has been developed by The United State Environmental Protection Agency (EPA) Urban Dynamometer Driving Schedule (UDDS) for chassis dynamometer testing of heavy duty vehicles [8]. It is mostly uses for the calculation of fuel consumption and exhaust emissions. Driving patterns are composed of specific speed trajectories over time which are included constant speed periods, acceleration and deceleration phases [14]. HD-UDDS drive cycle are used in the vehicle industry for determination of vehicle performance in real world driving conditions for heavy duty vehicle. In this work the HD-UDDS are chosen for the case study. Speed profile of HD-UDDS drive cycle is shown in Fig. 10, a and required gear ratio on transmissions are illustrated in Fig. 10, b. Characteristics of the cycle are expressed as; duration is 1060 seconds, distance is 8.9 km, average speed is 30.4 km/h, maximum speed is 93.3 km/h and idle time are 353 s for 14 number of stops. Gear ratio is calculated by using Eq. (13) and traction force curve on tires which is illustrated in Fig. 7. Gear positions of seven speed gearbox are determined according to differences between the maximum reserve forces and total resistance force on vehicle traction tires. Required torque on summation box is split two equals for transmission unit. Devices before the summation box (gearboxes, torque converters and engines) are worked synchronously during the drive cycle.



a b

Fig. 10 Speed profile of drive cycle HD-UDDS (a) and related gearbox ratios during drive cycle (b)

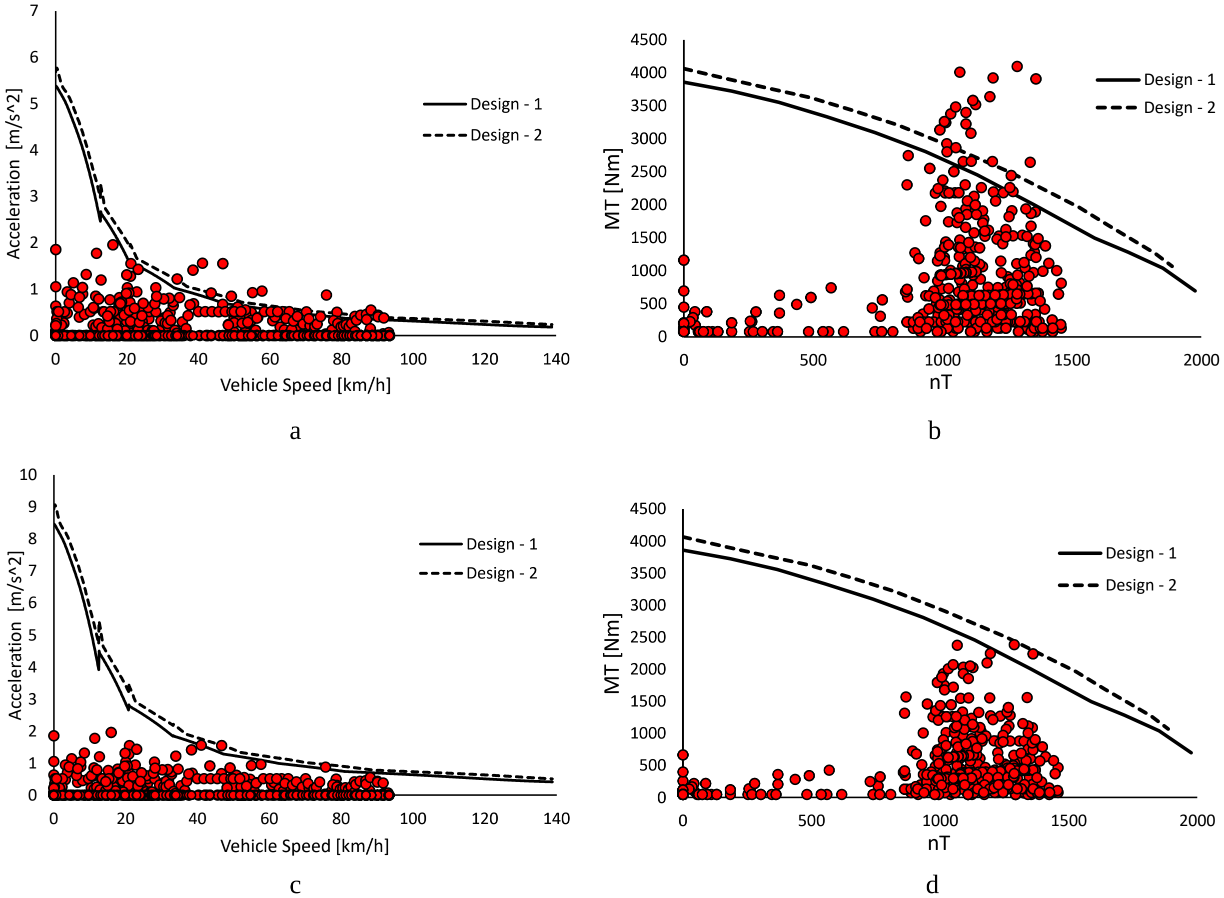


Fig. 11 Simulation results of acceleration requirements for loaded (a) and unloadaed (c) vehicle and moment requirements for loaded (b) and unloadaed (d) vehicle HD-UDDS

To compare the effects of torque converter on heavy duty vehicle powertrain model, drive cycle simulations were performed by using HD-UDDS cycle. Required moment on torque converter $M_{required,TC}$ during the test cycle is defined:

$$M_{requied,TC} = \frac{\left(F_R + F_L + F_a\right) r_{dyn}}{\tau_{GB_i}\ 2\ \tau_{SB}\ \tau_{TB}\ \tau_A\ \eta_T}. \tag{18}$$

Speed requirement at the torque converter $n_{required,TC}$ is expressed as:

$$n_{required,TC} = \frac{V_{cycle\cdot}\ 60\ \tau_{GB_i}\ \tau_T\ \tau_{SB}\ \tau_A}{2\ \pi\ r_{dyn}\ 3.6}, \tag{19}$$

where V_{cycle} is the speed of vehicle that is defined by test cycle.

Acceleration requirement because of HD-UDDS speed profile can be calculated by Eq. (14). Output moment of torque converter and maximum acceleration capacity of vehicle were given in Figs. 4 and 9. before. Acceleration and moment requirement points in related speed must be smaller than capacitive curves of them. Achievement of test cycle speeds is ensured by that. Fig. 11 shows the acceleration and moment requirement and their comparisons with torque converter designs (Design 1 and Design 2) for both loaded and unloaded conditions of the vehicle. In loaded operating condition that is showed in Fig. 11, a and b, required acceleration and moment higher than their maximum capacities. It shows that the vehicle cannot reach the speed values of test cycle when vehicle is fully loaded. Difference between torque converter selections are clearly shown in unloaded vehicle operation. Torque converter moment output for design 2 seems enough to achieve test cycle speeds in Fig. 11, d. However, design 1 of torque converter moment output is under two required operating points.

Additionally, fuel consumption of a heavy duty vehicle is crucial as well as vehicle performance. Fuel consumption of the vehicle can be investigated by using specific fuel consumption data of engine which is shown in Fig. 12. Consmption data is given for engine speed and power in g/kWh unit.

Engine operating points during the test cycle speed profile were found by using Eqs. (2), (3) and (4) with the torque convertes data which is given in Fig. 2. Engine moment requirements conversation from the required moment of torque converter which was calculated by Eq. (18) and shown in Fig. 11, can be considered the same procedure with creating power output curve of the torque converter.

The operating points give moment requirements M_{Eng} and speed requirements n_{Eng} of the engine. The formulation of power requirements P_{Eng} of the engine is expressed as:

$$P_{Eng} = \frac{M_{Eng} n_{Eng}}{9549}. \tag{20}$$

Drive cycle simulation for fuel consumption of the vehicle is implemented and the results are shown in Fig. 13 for the Designs 1 and 2 of torque convertors.

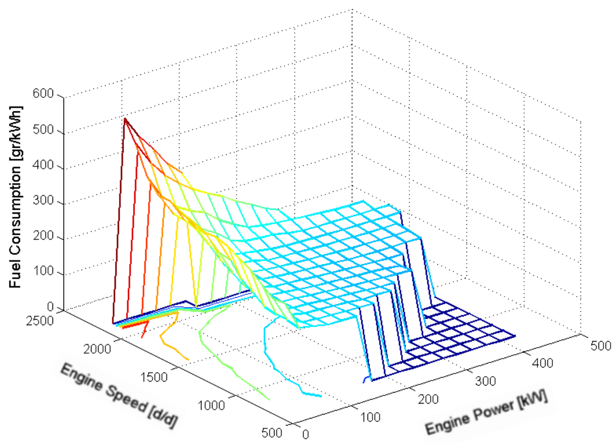


Fig. 12 Fuel consumption data of the engine

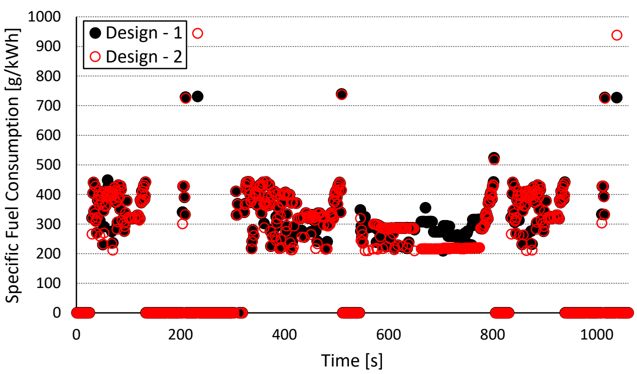


Fig. 13 Fuel consumptions of unloaded vehicles equipped with different designs of torque converters

Spesific fuel consumptions are pointed during HD-UDDS drive cycle for one engine. The simulation is carried out for the unloaded vehicle and the results are obtained from acquirable operational points of torque converter designs. It meants that the comparisations of fuel consumptions with torque converter Designs 1 and 2 consist same number of operation points.

Specific fuel consumption result indicates that although some points are higher in design 2 rather than design 1, 204 g/kWh average fuel consumption are occurred in design 1 which is less than 196 g/kWh consumption in design 2.

4. Conclusion

This paper has introduced a heavy-duty fire truck powertrain, integrating two different types torque converter designs (Design 1 and 2) to provide better performance and less fuel consumption by increasing operational area of engine. Powertrain model of vehicle is presented for the simulation by using HD-UDDS drive cycle.

Performance results consist maximum acceleration and acceleration time from 0 km/h to its maximum speed. Additionally, effects of torque converter designs on fuel consumption of vehicle are investigated. Performance results were given both loaded and unloaded mass of vehicle. However, many points was not achieved to drive cycle requirements for loaded condition that is clearly shown in Fig. 11 (points above the curves in Fig. 11, a and b). Therefore, fuel consumption results were given just unloaded conditions of vehicle to provide more accurate evaluations.

The results and prospects are summarized through the investigations of the effects of design 1 and 2 types torque converters as follow:

- Maximum acceleration capacity can be increased 5% by using Design 2 type torque converter.
- Although, loaded vehicle can reach from 0 to 100 km/h speed in 43.14 s with Design 1 type torque converter, Design 2 type provides 36.97 s. The acceleration time can be reduced 14.5% with the usage of Design 2.
- The acceleration time of unloaded vehicle from 0 to 100 km/h speed is 22.56 s in Design 1. But, 19.64 s accelerated time can be provided with Design 2 which is reduced 12.9% the time for needed.
- 4% fuel consumption savings can be obtained with Design 2 for coworking two engines.
- Power reuirements of HD-UDDS drive cycle can be achived by only Design 2 type in unloaded condition which is clearly shown in Fig 11, d.

This results leads to the conclusion that the powertrain analysis and simulation can use the investigation of the effects of component characteristics. In this paper, different torque converter characteristics in terms of Design 1 and Design 2 are investigated. The study can be extended to evaluate on cost or comfort of vehicle by using this method in future works.

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SIMULATION OF 8x8 HEAVY DUTY TRUCK FOR EVALUATING EFFECTS OF TORQUE CONVERTER CHARACTERISTICS ON VEHICLE PERFORMANCE

S u m m a r y

This paper investigates the powertrain system of an 8x8 heavy duty vehicle through the integration of two couple of diesel engine with automatic transmission as power sources for evaluating the vehicle performance. The primary goal of this paper is to show the effects of torque converter selection on acceleration performance and fuel consumption of the vehicle. To achieve this goal, powertrain system simulation carried out by using two different torque converter individually according to Heavy Duty Urban Dynamometer Driving Schedule (HD-UDDS) drive cycle. Torque converter alternatives (design 1 and design 2) are chosen considering operational area in moment characteristics of the engine. A mathematical model of powertrain system is implemented based on longitudinal dynamics of vehicle motion in HD-UDDS cycle. Required moments on torque converter during the motion of test cycle are compared for converter alternatives and loading condition of vehicle. Fuel consumption results are also compared by using fuel consumption data and required power in the engine. Results demonstrate that the increase in acceleration performance and traction ability of vehicle can be obtained by selection of torque converter correctly.

Keywords: powertrain model, heavy duty vehicle, torque converter, drive cycle.

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