

DESIGN AND PERFORMANCE ANALYSIS OF A 2-SPEED TRANSMISSION FOR ELECTRIC HEAVY-DUTY MINING VEHICLES

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Electrification of heavy-duty vehicles draws increasing attention, which could lower fuel consumption and emission. A novel concept design of a 2-speed transmission for electric heavy-duty mining vehicles is proposed in this paper. Its working principle and shift process are introduced. A shift strategy considering vehicle speed, torque ratio and payload coefficient is developed for such vehicles, which takes account of the influence of payload variation. A mining vehicle (payload: 50 metric tons) is modeled including 5 subsystems, namely drive cycle, driver, transmission, battery and motor, and vehicle dynamics. Simulations are conducted under a typical operation cycle defined with payload, speed and grade profiles of such mining vehicles to investigate economic and power performance of the proposed transmission. The economic efficiency of the vehicle model with a 1-speed gear box and the proposed transmission are evaluated in terms of state of charge (SOC) and energy consumption under one working cycle. The results suggest the vehicle implementing a 2-speed transmission reveals lower energy consumption and much higher accelerating and grade ability compared with a 1-speed gearbox. It is demonstrated that the implementation of the 2-speed transmission improves economic and power performance of such vehicles, which could be employed in the future electric heavy-duty mining vehicles.

Keywords: 2-speed transmission, Electric Heavy-Duty Mining Vehicle, Shift Strategy, Planetary Gear Train

1. Introduction

Development of hybrid electric vehicles (HEV) and electric vehicles (EV) is a major contribution on the growing demands on attenuation of fuel consumption and emission. Heavy-duty mining vehicles widely employed in open pit and underground mines with huge amounts of fuel consumption, however, are lack of attention on sustainability investigation in past decades. The electrification of such vehicles could contribute a lot on solving energy and environmental problems. In recent years, an advanced hybrid propulsion and energy management system for mining trucks is developed to make use of the braking energy normally

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dissipated in grid resistors [1]. The energy management strategy of such hybrid mining truck is discussed by considering road geometry data [2]. Although the hybrid propulsion system could improve fuel efficiency, the fuel consumption of such vehicles is still considerable. The electrification of heavy-duty vehicles increasingly draws attentions from manufacturers and researchers [3].

A single-speed gearbox rather than a transmission is mainly deployed to the power train of electric vehicle due to a better torque performance of a motor than an internal combustion engine especially at low speed. A 2-speed transmission, however, could still obtain a wider range and an improved efficiency of motor. The investigation of the impact of transmission on EV performance could be found in several literatures [4, 5]. The impact of different transmission ratios on EV's dynamic and economic performance was studied [6]. The possibility of using a transmission with more than 2 speeds in EV is investigated, and three EV cases respectively with 2-, 3-, and 4-speed transmission are compared in terms of efficiency and performance [7]. The reported studies are generally focused on electric passenger cars. The improvement of dynamic and economic performance of commercial vehicles especially heavy-duty mining vehicle by implementing a 2-speed transmission is lack of studies.

The other investigations on 2-speed transmission mainly focused on concept design, gear ratio determination and shift strategy development. Shin et al. [8] proposed a 2-speed transmission design employing a Simpson type planetary gear train for electric commercial vehicles. A new invention method "Inventogram" was used to develop a 2-speed transmission structure and the optimized shift control process was investigated [9]. A shift control algorithm was proposed for a 2-speed DCT to improve the shift quality [10]. A shift logic based on the optimal efficiency of a motor-transmission system is developed with 2 parameters for plug-in buses [11]. An online shift schedule optimization strategy for a 2-speed automated manual transmission (AMT) is designed to improve EV's energy efficiency [12]. An optimal coordinated control problem of the motor and the friction clutch with complex constraints on both the state and control variables during the overlapping shifts is investigated [13]. An optimal controller for the gear shift of a 2-speed AMT of a EV is proposed, and the controller is designed as a time-varying feedback [14]. The heavy-duty mining vehicles work nearly 24 hours per day under huge payload and adverse operation condition, which requires a compact and reliable design of transmission.

In this paper, a novel 2-speed transmission design will be proposed for heavy-duty mining vehicles. And the improvement of economic and power performance for electric heavy-duty mining vehicles by implementing the proposed transmission will be investigated. Firstly, the working principle and shift process of the proposed transmission are described. Moreover, a shift strategy is

developed by considering vehicle speed, torque ratio of motor and payload coefficient. A 50 tons mining vehicle is modeled to investigate the economic and power performance of the proposed transmission. Furthermore, simulations under a typical operation cycle of such vehicles are conducted, and the performances of the vehicle with the proposed 2-speed transmission and a 1-speed gearbox are analyzed.

2. Design of the transmission

The powertrain layout of an electric mining truck is illustrated in Figure 1. It consists of a battery pack, a motor, an automatic 2-speed transmission or a 1-speed gearbox, a final drive, a differential, two half shafts, and two hub reduction gears. The hub reduction gears are necessary for heavy-duty mining vehicles to obtain required driving torque. The transmission is placed between the motor and final drive, which could provide 2 different ratios. The design purpose of transmission for electric mining vehicle is to improve ranging, dynamic and efficiency performance of motor.

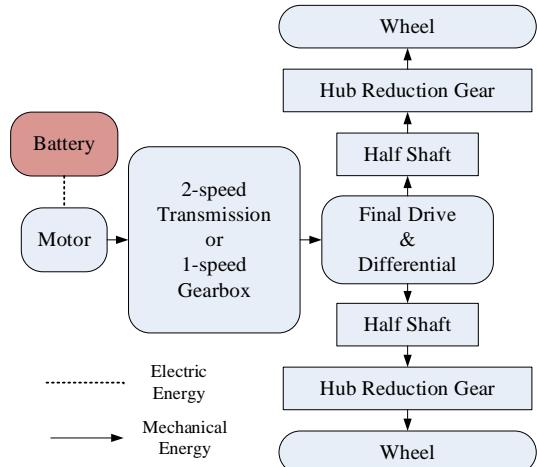


Fig. 1. Powertrain layout of an electric mining vehicle

In this paper, a 2-speed transmission employing a planetary gear train is proposed for electric heavy-duty mining vehicles, which is shown in Figure 2. Planetary gear train is widely implemented in Automatic transmission (AT) which is used in both road and off-road vehicles, and the multi-disc wet clutches are also widely employed to realize gear shift for such transmission. Therefore, this paper proposes a novel 2-speed transmission with single planetary gear, whose gear shift is actuated by hydraulic pressure. A hydraulic system involves pump, magnetic valve and pipeline will be designed to provide hydraulic pressure and control to the proposed transmission. The planetary gear train is composed of a sun gear, 3

planetary gears, a planetary carrier and a ring gear. Its kinematic equation could be formulated as

$$\omega_b + \frac{1}{p} \omega_a - \frac{(1+p)}{p} \omega_x = 0 \quad (1)$$

where ω_a , ω_b , and ω_x are rotational speed of the sun gear, ring gear and planetary carrier in the planetary gear train, respectively. And p is the teeth ratio of ring gear to sun gear.

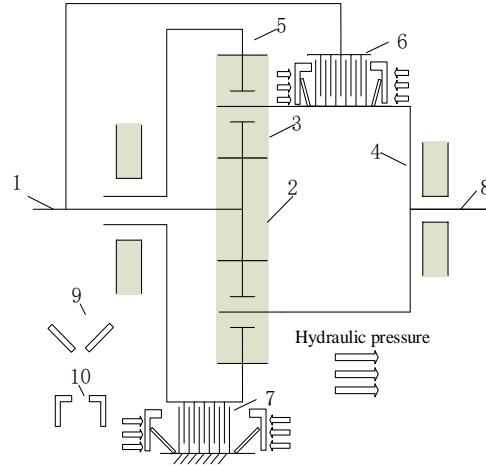


Fig. 2. Configuration of the 2-speed transmission (1-input axle, 2-sun gear, 3-planetary gear, 4-planetary carrier, 5-ring gear, 6-clutch 1, 7-clutch 2, 8-output axle, 9-return spring, 10-actuating component)

Two different gear ratios could be realized by on/off combinations of two clutches, which is listed in Table 1. When clutch 1 is off and clutch 2 is on, the ring gear is fixed to transmission box, which means ω_b is equal to 0. Planetary carrier is the output component of the planetary gear train, and its rotation speed could be calculated as

$$\omega_x = \frac{\omega_a}{1+p} \quad (2)$$

The ratio of 1st gear is

$$i_{ax}^b = \frac{\omega_a}{\omega_x} = 1+p \quad (3)$$

When clutch 1 is on and 2 is off, ω_a is equal to ω_b . Therefore the ratio of 2nd gear is 1. And a neutral gear could be realized by making both clutch 1 and 2 disengaged.

Table 1

On and off for clutches during shift process

	Neutral	1st	2rd
Clutch 1	Off	Off	On
Clutch 2	Off	On	Off

The ratio of 1st gear of the proposed 2-speed transmission is dependent on parameters of the planetary gear train, while that of 2rd gear is 1 despite of the teeth number of each gear. The proposed design is a compact structure with only 1 planetary gear train and 2 clutches. “On” of two clutches could be realized by actuating components powered by hydraulic pressure. Return springs are employed to obtain “Off” state of both clutches.

3. Shift control strategy

Mass parameter plays a very important role in vehicle dynamics [15]. Mass of the heavy-duty mining vehicles significantly varies in different payload conditions, which should be emphasized in shift strategy development and performance evaluations. In this paper, a shift control strategy taking payload variation into account is developed for the proposed 2-speed transmission, which is determined by dynamic performance shift logic.

The acceleration of a vehicle could be calculated by

$$a = \frac{du}{dt} = \frac{1}{\delta m} [F_t - (F_f + F_i + F_w)] \quad (4)$$

where u is vehicle speed, m is vehicle mass, δ is transition coefficient of rotary mass of vehicle, F_t is driving force, F_f is rolling resistance, F_w is aerodynamic resistance, and F_i is grade resistance, respectively. Equation 4 could be extended by the following formulas:

$$F_t = \frac{T_{tq} i_g i_t \eta_t}{r}; \quad F_f = mgf \cos \alpha; \quad F_i = mg \sin \alpha; \quad F_w = \frac{C_D A u^2}{21.15} \quad (5)$$

where T_{tq} is output torque of electric motor, i_g is total ratio of final drive and hub reduction gear, i_t is gear ratio of transmission, η_t is drive efficiency of the powertrain system, r is rolling radius of wheel, f is rolling resistance coefficient, g is gravity acceleration, α is road slope, C_D is aerodynamic resistance coefficient, A is frontal area of the vehicle, respectively.

T_{tq} is related to maximum output torque of the electric motor at different vehicle speed and driver input, which is formulated as

$$T_{tq} = \xi T_m(u) \quad (6)$$

where T_m is maximum output torque of the electric motor, and ξ is torque ratio defined as the ratio of required torque to maximum torque to describe driver input. i_t is different respectively at 1st gear and 2rd gear. Mass parameter, playing a

significant role in vehicle dynamics, is taken into account in shift strategy development for such vehicles. Therefore, a load coefficient is defined as

$$\lambda = \frac{m_{real} - m_{curb}}{m_{total} - m_{curb}} \quad (7)$$

where m_{real} , m_{total} , and m_{curb} are actual mass, laden mass and unladen mass of vehicle, respectively. λ clearly indicates payload state of vehicle. The vehicle takes no load when $\lambda=0$, while it is full-loaded when $\lambda=1$. Equation 4 could thus be rewritten by taking λ into account

$$a = \frac{1}{\delta m(\lambda)} [F_t(\xi, u, i_t) - (F_f(\lambda) + F_i(\lambda) + F_w(u))] \quad (8)$$

where driving force is a function of torque ratio, vehicle speed and gear ratio of transmission, rolling resistance and air resistance are related to load coefficient.

The upshift speed is determined when the 1st and 2nd gears yield the same acceleration under the same torque ratio and vehicle speed.

$$a_1 = a_2 = \frac{1}{\delta m(\lambda)} [F_t(\xi, u, i_t) - (F_f(\lambda) + F_i(\lambda) + F_w(u))] \quad (9)$$

The downshift speed is thus obtained by taking a negative value to upshift speed to avoid unnecessary high-frequency gearshifts. A shift strategy determined by vehicle speed, torque ratio and load coefficient based on dynamic performance shift logic could be obtained using Equation 9. The demonstration of the shift control strategy is conducted by simulations, and its influence on clutch wear and energy consumption will be investigated in the future study.

4. Vehicle Model

An electric mining vehicle whose payload is 50 metric ton is modeled employing a bottom-up approach in Matlab/Simulink software to analyze the economic and power performance improvement of the proposed transmission. Matlab/Simulink is widely employed to model a system according to the characteristics of its physical plant and to validate a design before prototyping. The software could combine textual and graphical programming to design the system in a simulation environment. Figure 3 illustrates the schematic of the vehicle model, which is consisted of 5 subsystems, namely drive cycle, driver, battery and motor, transmission, and vehicle dynamics.

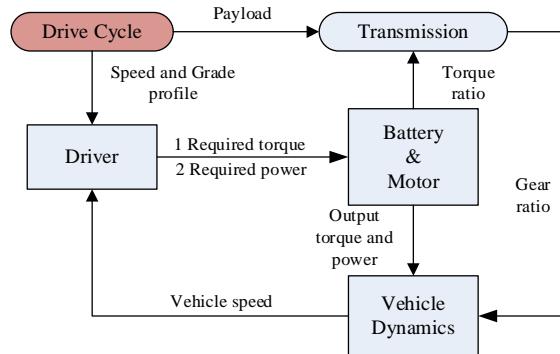


Fig. 3. Model of an electric heavy-duty mining vehicle

Driver cycle is defined by payload, speed and grade profiles, which is established based on the operational data of the modeled vehicle. Transmission subsystem is established based on shift strategy and performs a shift decision during simulation.

Driver model. A driver model is established to obtain the required driving torque of electric motor. It firstly calculates the required driving force of the vehicle along its forward and backward direction:

$$F_t = F_f + F_w + F_i + F_j \quad (10)$$

where F_j is acceleration resistance. It is calculated by

$$F_j = \delta m \frac{du}{dt} \quad (11)$$

The required driving force of the vehicle could be obtained by Equations 5, 10 and 11. The required driving torque of motor could be subsequently calculated by

$$T_{req} = F_t \frac{r}{i_g i_t \eta_t} \quad (12)$$

Parameters of the modeled mining vehicle for calculation of required driving torque and shift strategy are listed in Table 2.

Table 2

Vehicle parameters

Parameters	Symbol	Value	Unit
Unladen mass of vehicle	m_{curb}	30000	kg
Laden mass of vehicle	m_{total}	80000	kg
Total ratio of final drive and hub reduction gear	i_g	26	
Gear ratio of transmission	i_t	3 (1st); 1 (2nd)	
Frontal area of the vehicle	A	13.12	m^2
Rolling radius of wheel	r	0.97	m
Rolling resistance coefficient	f	0.025	
Gravity acceleration	g	9.8	m/s^2
Aerodynamic resistance coefficient	C_D	0.8	
Transition coefficient of rotary mass of vehicle	δ	1.05	
Drive efficiency of the powertrain system	η_t	0.9	

Battery and motor. The battery system is modeled to calculate the state of charge (SOC) of battery during simulation.

$$SOC = \frac{C - \int I}{C} \quad (13)$$

where C and I are respectively capacity and real-time current of the battery. I could be obtained by

$$I = \frac{Vter(t) - \sqrt{Vter(t)^2 - 4R(t)P(t)}}{2R(t)} \quad (14)$$

where $Vter(t)$ and $R(t)$ are terminal voltage and internal resistance of the battery, and $P(t)$ is input power of motor, respectively. $Vter(t)$ could be calculated by

$$Vter(t) = Voc - IR(t) \quad (15)$$

where Voc is open circuit voltage of the battery. The capacity and open circuit voltage of the battery used in the mining vehicle are 400 Ah and 720 V, respectively.

The output power, output torque and efficiency of electric motor could be calculated from motor system by using the following equations:

$$P_{req} = \frac{T_{req}n}{9550}; \quad P_{out} = \min\{P_{req}, P_{max}\} \quad (16)$$

where P_{req} , P_{out} , and P_{max} are the required, output and maximum power of electric motor, respectively. n is the output motor speed which is related to vehicle speed. The power of electric motor could be calculated by

$$P = \frac{P_{out}}{\eta_M} \quad (17)$$

where η_M the efficiency of motor.

Figure 4 shows the torque limit and efficiency of a motor which is employed in the vehicle model. It indicates η_M is more than 0.9 when motor speed is from 750 rpm to 3000 rpm, while η_M is less than 0.7 especially when motor operates in lower speed (0 to 500 rpm). Lower efficiency of motor may lead to higher energy waste, which should be avoided. The power limit of the modeled motor could thus be obtained from Figure 4. These characteristics of the motor are defined as look-up functions in Simulink software and could be determined by the output torque and motor speed.

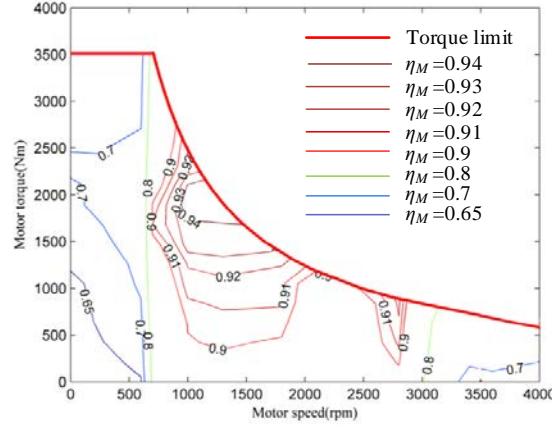


Fig. 4. Torque limit and efficiency of the motor

Vehicle Dynamics. The vehicle dynamics module is formulated to obtain location, speed and acceleration of the vehicle. The forward acceleration of the vehicle could be calculated by Equation 4. The vehicle speed could be subsequently obtained by integrating the forward acceleration.

A traction constraint is also considered, which could be expressed as

$$F_x = \min \{F_t, F_\varphi\} \quad (18)$$

where F_x is traction force of the vehicle, and F_φ is traction force limit, respectively.

$$F_\varphi = F_z \varphi \quad (19)$$

where F_z is vertical force of the vehicle, and φ is adhesion coefficient, respectively. The vertical force of the modeled vehicle could be calculated by

$$F_z = mg \left(\frac{a}{L} \cos \alpha + \frac{h_g}{L} \sin \alpha \right) - \frac{1}{2} C_{Lr} A \rho u^2 + m \frac{h_g}{L} \frac{du}{dt} + mg \frac{rf}{L} \cos \alpha \quad (20)$$

where L is wheelbase, h_g is height of mass center, a is distance between mass center and front wheel, C_{Lr} is rear aerodynamic lift coefficient, and ρ is air density, respectively. The values of afore-mentioned parameters are listed in Table 3.

Table 3
Vehicle parameters for vertical force calculation

Parameters	Symbol	Value	Unit
Height of mass center	h_g	1.8	m
Distance between mass center and front wheel	a	2.68(laden); 2.01(unladen)	m
Wheelbase	L	4.02	m
Rear aerodynamic lift coefficient	C_{Lr}	0.4	
Air density	ρ	1.29	kg/m ³

5. Result and discussion

Shift strategy. A shift strategy determined by torque ratio, vehicle speed and load coefficient is developed for the proposed 2-speed transmission based on the vehicle parameters listed in Table 2. Figure 5 illustrates the strategy respectively when load coefficient is from 0 to 1. The shift speed varies significantly between laden and unladen states, which indicated the payload has a significant impact on shift strategy. The shift strategy will be subsequently employed in the “transmission” subsystem of the vehicle model to investigate the economic and power performance of an electric mining vehicle with the proposed transmission.

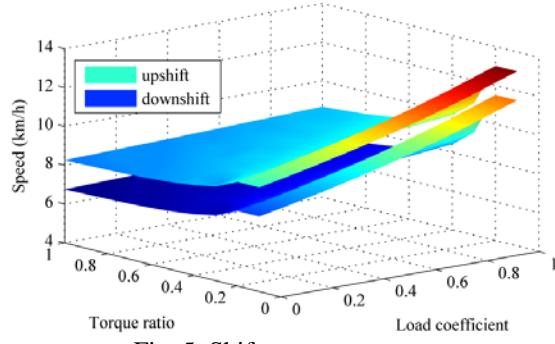


Fig. 5. Shift strategy

Economic and power performance. The economic performance of the modeled heavy duty mining vehicle implementing the proposed 2-speed transmission is analyzed by using a typical mining driving cycle according to speed and grade profiles in one operation cycle, which is shown in Figure 6. The cycle time is 1730s. The maximum speed and grade are 30 km/h and 13.65%, respectively. The mean speed and grade are 16.66 km/h and 1.38%, respectively. From 0 to 940 s, the vehicle model with full load goes uphill to dumping site, and then it goes downhill with no load from 967 to 1730 s.

A 1-speed gearbox whose gear ratio is 1.5 is also employed to investigate the relative economic and power performance of the proposed transmission. The actual speed of the vehicle model respectively with 1-speed gearbox and 2-speed transmission during the simulations are illustrated in Figure 7(a). The actual vehicle speeds follow the target speed adequately with only a little error during the process of driving uphill. The result shows that both of the 1-speed gearbox and 2-speed transmission could satisfy the operation requirements of mining transportation. Figure 7(b) shows the gearshift during the operation cycle. It suggests that 1st gear is primarily used in the process of going uphill with full load, while 2nd gear is employed when the vehicle requires a higher speed.

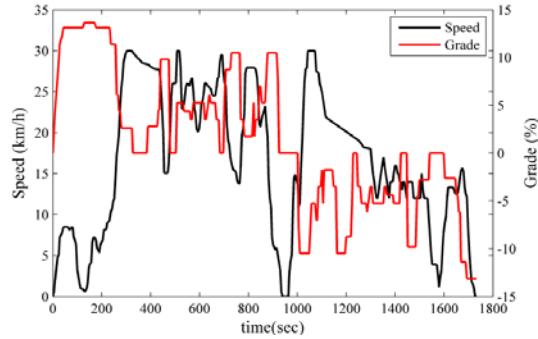


Fig. 6. Operation cycle

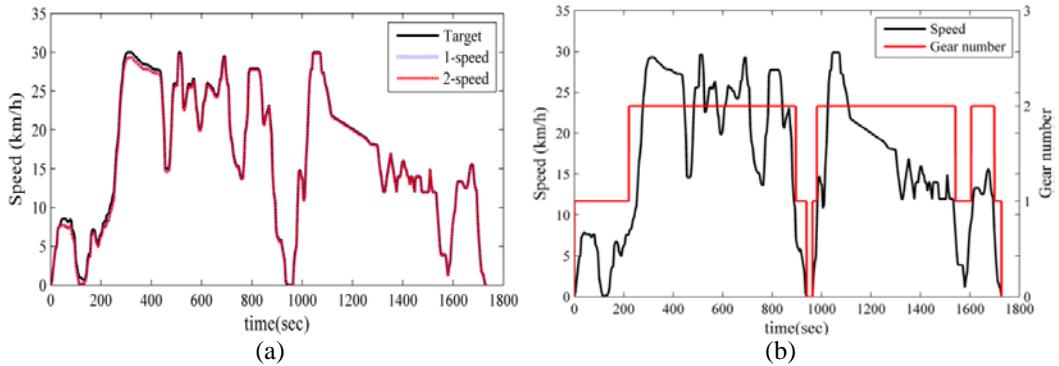


Fig. 7. Speed following and Gear selection of the model: (a) speed following and (b) Gear selection

Figures 8(a) and 8(b) illustrate the operating points of the electric motor with 1-speed gearbox and 2-speed transmission during the simulation, respectively. The motor with 2-speed transmission mainly operated within a higher efficient zone during the entire drive cycle. Moreover, its outputs would not exceed the rated torque limit in the constant torque range. While the motor with the 1-speed gearbox operated in much higher torque points when the motor speed is less than 750 rpm for climbing and accelerating.

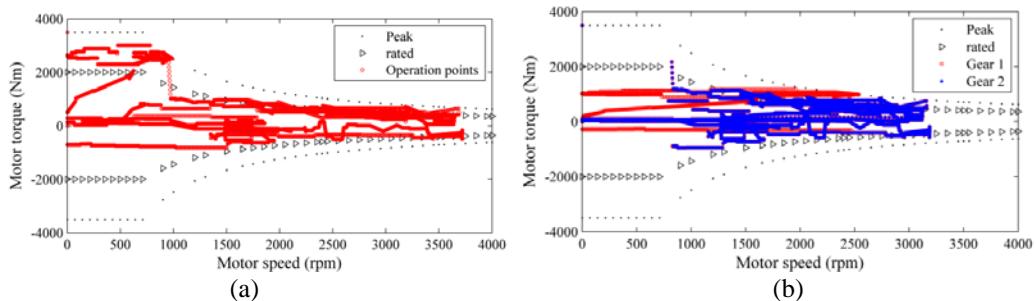


Fig. 8. Operating points of electric motor with: (a) 1-speed gearbox and (b) 2-speed transmission.

The energy consumption and SOC reduction under one operation cycle are listed in Table 4. The vehicle model with a 2-speed transmission reveals much lower energy consumption than that with a 1-speed gearbox. There is 2.76% deduction of energy consumption by taking the proposed transmission. Figure 9 compares the SOC reduction processes of the vehicle respectively with gearbox and transmission. SOC reduced rapidly from 0 to 940 s due to going uphill with full load, while it went down slowly after the vehicle unloaded and went downhill. Figure 9 also demonstrates that the 2-speed transmission could improve the economic performance of the mining vehicle compared with a 1-speed gearbox.

Table 4

Energy Consumption

Parameters	1-speed gearbox	2-speed transmission
SOC reduction (%)	23.56	20.80
Energy consumption (kWh)	67.85	59.90

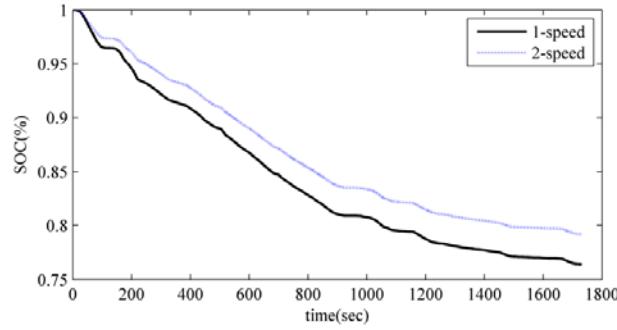


Fig. 9. SOC reduction during one driving cycle

Power performance of the mining vehicle with 1-speed gearbox and 2-speed transmission are compared by calculating three indicators, namely maximum acceleration, maximum speed and grade ability. Table 5 shows that the vehicle with the proposed 2-speed transmission reveals higher values of maximum acceleration and grade ability and much wider speed range. The proposed 2-speed transmission yields lower energy consumption and higher power performance compared with a 1-speed gearbox, which may be desirable for future electric heavy-duty mining vehicles.

Table 5

Results of power performance

Parameters	1-speed gearbox	2-speed transmission
Maximum acceleration (m/s^2)	1.27	2.78
Maximum speed (km/h)	37.50	56.26
Grade ability (%)	14	30

6. Conclusion

This paper presented a novel design of a 2-speed transmission for electric heavy-duty mining vehicles by employing a planetary gear train and two clutches. The vehicle with the proposed 2-speed transmission yields 2.76% deduction of energy consumption than that with a 1-speed gearbox in one operation cycle. It is noted that the economic performance analysis does not take account of the cost of the transmission and the energy consumed by the hydraulic clutch actuators. A comprehensive economic performance analysis will be completed in the future study. The electric motor with the 2-speed transmission could primarily operate in high efficiency region. The vehicle with the transmission also yields much higher accelerating and grade ability as well as a wider speed range. The vehicle exhibits lower energy consumption and greater power performance by implementing a 2-speed transmission. The proposed transmission improves energy efficiency and power performance of the vehicle, which may thus be employed for the future design of electric heavy-duty mining vehicles. An experimental model of the proposed transmission will be fabricated, and performance tests will be conducted to verify the feasibility of the concept design.

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R E F E R E N C E S

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