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LIGHTWEIGHT LHD BEV LOADER WITH AN INDIVIDUAL DRIVE FOR EACH WHEEL

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Abstract:
The paper is devoted to the construction and design solutions of the power supply and drive systems of a prototype, lightweight LHD (Load-Haul-Dump) loader with an innovative "in-wheel drive" transmission system. The loader is a Battery Electric Vehicle (BEV) with replaceable energy storage. It is intended for use in underground mines with no explosion hazard, especially in metal ore mines. The developed model contains four independent, electric drive systems installed in the road wheels. The paper presents the results of simulation tests, which enabled estimating the required driving power and electricity demand in various regimes and working conditions of the loader. The results of the calculations were used to determine the parameters of the drive system components and the battery capacity. Challenges and problems faced by constructors of BEV LHD loaders with an in-wheel drive have been pointed out.

Key words: *self-propelled mining machines, LHD loaders, electric drive, battery power, BEV*

INTRODUCTION

LHD (load, haul, dump) loaders are wheel loaders with front dumping buckets. They are used to transport crushed excavated material in mines. Vehicles of this type transport the spoil from the face to the transfer point, moving back and forth along the mine tunnel, which is usually several hundred meters long. Compared to typical front loaders, they are distinguished by their low height and high efficiency, as well as high manoeuvrability. The low height of the loader enables reaching places in the mine with a very low roof. LHD loaders belong to the most common mining machines that are used in the room and pillar system in underground mines, in particular when access is difficult or dangerous.

One of the major elements of the LHD loader is a rear body *1*, connected with a platform *2* by means of a joint *4* with a vertical axis. The articulated connection of the rear body and the platform with hydraulic cylinders is responsible for the turning of the machine. A bucket *3* is attached to the platform *2*. Between the rear body *l* and the oscillation axle 5 there is a joint 6 with a horizontal axis (Figure 1b). The articulated oscillation axle is responsible for the ability to ride on an uneven surface. The drive units *8* are mounted in the immediate vicinity of the road wheels, while the battery *7* is located in the central part of the loader.

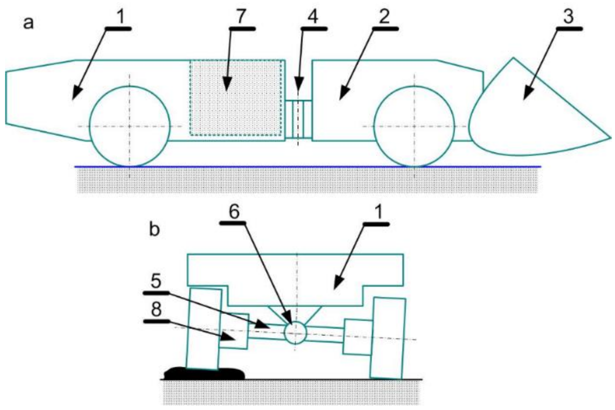


Fig. 1 Construction diagram of the articulated LHD loader: a. general view of the machine, b. view of the rear body oscillation axle

Due to the specificity of LHD loader working conditions, it is desirable to reduce the emission performance of this type of vehicles. Therefore, one of the directions of development in the field of LHD loader construction is to replace conventional combustion drive systems with systems based on electric motors. An additional benefit resulting from the use of electric drives is the simplification of the machine design and the resulting increase in the durability and reliability of the loader, also due to the simpler diagnostics of systems with electric drives.
The article is devoted to a lightweight battery-operated LHD loader with the designation EV-LKP1, which is the subject of the project carried out in cooperation between the Institute of Innovative Technologies EMAG and Bumech S.A.
The EV-LKP1 loader is a BEV (battery electric vehicle) with a removable energy storage (battery swap) located in the middle part of the working machine (item 7, Figure 1a). The use of battery swap is an important aspect of the loader operation. The possibility to quickly replace the battery in the event of its discharge is associated with a short technical break, after which the loader is ready for further operation. The construction of the loader's drive system enables energy recovery during braking and downhill drive, which should translate into an extended working time between successive battery replacements.
To build the energy storage, a set of lithium-ion batteries with a relatively large capacity was used, which ensures satisfactory long-term operation of the machine. The current capacity of the energy storage was selected so that the drive system of the machine could ensure its competitive performance. In the design process, great emphasis was placed on the safety of the lithium-ion cells and the functional safety of the energy storage. The safety in question was achieved, among others, through the use of a hierarchical system for monitoring and managing the work of the team, along with a set of sensors and an active temperature management system.
Against the background of existing solutions with a similar load capacity, the loader in question is distinguished by the use of an individual electric drive in each wheel (in-wheel-drive) (item 8, Figure 1b). Each of the four drives (motors with inverters and gears) makes up a separate drive unit that ensures achieving the assumed driving and braking conditions of the loader. It is equipped with an individual control system and a system to monitor working conditions, as well as a cooling system.

LITERATURE REVIEW

In recent years, an accelerated development in the construction of mining machines, including LHD loaders, has be observed, especially in the field of battery power, support systems, remote control and work autonomy [1]. At the same time, the process of designing machines for underground mines requires the use of modern methods that allow meeting the user requirements, while taking into account extremely difficult working conditions [2, 3, 4].
In the case of machines operating in closed spaces with limited air supply, electric motors can provide an effective alternative to internal combustion engines. In such conditions, the fact that the drive unit does not use oxygen and does not release emissions is a particularly expected property. Another benefit resulting from the use of electric drives is high efficiency (close to 90%), which translates into reduced heat emitted by the machine to the environment.
Challenges in the design of modern machines intended for underground work also include occupational health and safety issues, among others the frequently discussed problem of excessive noise [7]. A significant advantage of electric drives is clearly visible also in this aspect. In addition, the electric drive unit is less complicated [1, 8] in relation to combustion systems, which translates into the machine's increased reliability and durability.
The income of a mine is directly dependent on the rate of extraction, which is determined primarily by parameters such as: loading, haulage and dumping, as well as: drilling, blasting, roof support etc. [8]. The more often this cycle is repeated, the higher the transport efficiency is. For this reason, as well as for safety considerations, it is desirable to automate LHD vehicles [9, 10]. This trend is consistent with the direction of LHD loader development that involves the use of electric drives, which, due to their specificity, make it possible to develop complex systems for operating parameters control and monitoring.
Battery power of machines is a design and economic challenge [11]. In battery-powered electric vehicles, the range (or working time) depends on the electrical capacity of the storage. A higher battery capacity entails a larger vehicle weight, which is counterproductive. It is important to properly balance these aspects in LHD machines, primarily by carefully planning the machine use and the user's work organization. The economic aspect is expressed in both the efficiency of transport (transport of empty energy storage capacity) and the cost of the batteries themselves. In terms of construction, removable energy storages require an appropriate mechanical structure, adjusted to the rest of the vehicle with regard to dimensions, while taking into account the method of installation and replacement (potentially also the development of dedicated systems on the machine itself). Another challenge

could also be posed by the method and components used to electrically connect the energy storage circuits.

Battery-electric vehicles (BEVs) with a four-wheel drive, powered by multiple motors located on different axles, are gaining popularity, especially commercial vehicles, offering excellent dynamics and safety. They create new possibilities in terms of traction control and the use of anti-lock braking systems (ABS) [12, 13]. The key issue in such a solution is appropriate distribution of the energy flow between the power source and individual drive units. This applies to drive as well as braking solutions [14, 15].

Among the construction solutions available on the market of battery-powered LHD loaders with a load capacity similar to the case discussed in this study, there is the Artisan A4 haul truck. It has a load capacity of 4 tons and is equipped with batteries having a capacity of 88 to 133 kWh and a total drive power of 250 kW [16]. Another solution is Sandvik's LH518B loader, with a load capacity of 18 tons, equipped with three electric motors (2x180 kW on the front axle and one 180 kW motor on the rear axle), powered by a 353kWh battery [17]. A similar model, but with a slightly smaller working capacity, is the Scooptram ST14 loader (Epiroc). It is characterized by a load capacity of 14 tons, equipped with one 200kW drive motor and a 160kW hydraulic system. This loader is powered by a 300kWh battery [18]. A detailed discussion of the global trends in the development of BEV mining machines has been presented in [1]. It should be noted that none of these solutions is fully consistent with the concept of "in wheel-drive" construction.

RESEARCH METHODOLOGY

The aim of the work was to develop a computational model for selecting the required parameters of a drive system for the new BEV LHD loader. A number of assumptions were specified and, next, the physical model was written in the form of a mathematical one. The subject of research is a unique LHD loader, which is currently in the design phase, hence empirical studies cannot be used. Calculations were carried out to select the key components of the drive nodes, and the boundary conditions for the loader operation were determined. In order to facilitate the interpretation of the results and due to the typically practical nature of the model, all calculations were performed using masses expressed in kilograms.

PHYSICAL AND MATHEMATICAL MODEL OF THE LHD LOADER

Concept of the loader drive system solution

The prototype LHD loader in question belongs to the class of loaders with a bucket capacity of 4000-5000 kg, equipped with an electric drive.

The schematic diagram of the loader's power supply and drive system has been shown in Fig. 2.

The use of four independent motors, each of which is installed in one drive wheel, resulted in a significant simplification of the loader construction in mechanical terms.

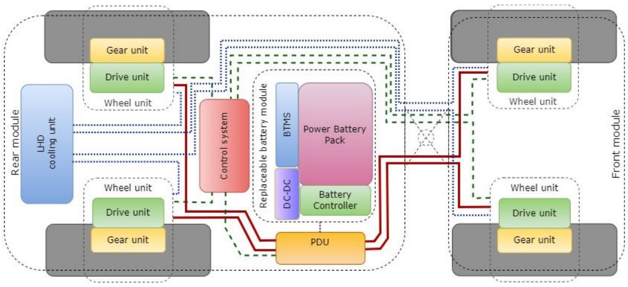


Fig. 2 Power and driving systems of the analysed LHD

The drive transmission takes place in a relatively short kinematic chain. Such a solution may be beneficial due to increased reliability of the machine. Each of the drive units is a separate module, including: a BLDC electric motor with an inverter and a reduction gear equipped with a mechanical brake. The autonomy of each of the drive units enables operating the loader also in the event of a failure of one or more systems.

Loader parameters adopted for calculations

The design calculations have been based on the following operational parameters of the loader:

- Q_{LHD} = 1400 kg (loader's mass)
- Q_{bat} = 2000 kg (mass of battery pack)
- Q_{exc} = 4000 kg (mass of excavated material)
- D_{tire} = 1270 mm (tyre external diameter)
- v_{max} = 32 km/h (expected maximum speed of driving)
- v_{nom} = 5 km/h (expected nominal driving speed (possible under full load in most operating conditions).

Therefore, the total maximum mass of the loader for which the calculations of the drive system parameters have been made equals (eq. 1) Q_{tot}=20 000 kg.

$$Q_{tot} = Q_{LHD} + Q_{bat} + Q_{exc} \tag{1}$$

Based on separate analyses, the moment of inertia of each road wheel was determined as J_w = 90 kg·m^2, and the wheel mass as m_w = 200 kg.

It has been assumed that the drive torque of the loader should allow the vehicle to enter the road with an inclination of α = 17° (i.e. 30%), assuming that in this case the vehicle can drive without acceleration. Additionally, it has been assumed that the loader's acceleration should reach a = 2 m/s^2.

Loader operating parameters

Figure 3 shows a diagram of the loader with marked values characteristic of the vehicle's motion up the hill.

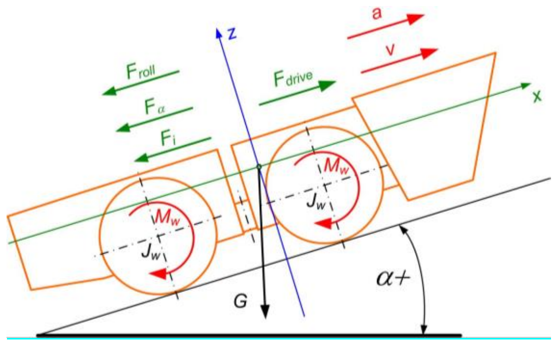


Fig. 3 Diagram of the loader with marked physical quantities characterizing the vehicle movement

Appropriate values of rolling resistance that must be overcome by the drive system have been assumed as in the case of a typical wheeled vehicle. This provided a basis for determining the required total driving force (F_{drive}):

$$F_{drive} = F_f + F_{roll} + F_\alpha \quad (2)$$

where:

Rolling resistance force:

$$F_{roll} = Q_{tot} \cdot g \cdot \cos(\alpha) \cdot f_f \quad (3)$$

The rolling resistance coefficient (f_f) was adopted on the basis of [19, 20], which allowed adopting three ranges of this parameter values for analysis: f_f = 0.1/0.35/0.5 – they express, respectively: 0.1 – ride on a hardened dirt road; 0.35 – difficult driving conditions, muddy road; 0.5 – extremely difficult road conditions, very muddy unpaved road. Resistance force during driving with acceleration (inertia resistance):

$$F_i = \left(Q_{tot} + 16\frac{J_w}{D_{tire}^2} + 4m_w\right) \cdot a \quad (4)$$

Resistance force when driving uphill (uphill resistance):

$$F_\alpha = Q_{tot} \cdot g \cdot \sin(\alpha) \quad (5)$$

Table 1 lists the required values of the driving force and the corresponding values of the torques to be generated on each wheel of the loader. It has been assumed in the calculations that the load will be distributed evenly on each wheel. This approach is a significant simplification of the issue, but the purpose of the calculations was to estimate the required parameters of the drive units. It should be emphasized that the results presented in the table refer to the case of the loader driving with a full load up a hill with a significant slope (α = 17°). To refer the results to the case of driving on a horizontal surface, the values of the required drive torque per one wheel ($M_{w\_\alpha 0}$) for the case (α = 0°) have been given in the last row of Table 1.

Table 1

Values of the required force and torques of the loader

f_f [-]	0.1		0.35		0.5	
a [m/s²]	0	2	0	2	0	2
F_{drive} [kN]	**76.1**	**119.7**	**123.0**	**166.6**	**151.1**	**194.7**
M_{tot} [kNm]	48.3	76.0	78.1	105.8	96.0	123.6
$M_w = M_{tot}/4$ [kNm]	**12.1**	**19.0**	**19.5**	**26.4**	**24.0**	**30.9**
$M_{w\_\alpha 0}$ [kNm]	**3.1**	**10.0**	**10.9**	**17.8**	**15.6**	**22.5**

Based on the obtained results, it has been estimated that the required torque per one wheel is in the order of 20-25 kNm. This value will allow driving the loader with a full load even in difficult terrain conditions.

Another aspect related to the driving ability of the loader is the expected adhesion of the wheels to the ground and the associated risk of skidding. For this purpose, estimate calculations were carried out, assuming that the friction coefficient (tyre-road adhesion) in the tyre-ground contact was μ = 0.6. The assumed value corresponds to the conditions of the contact between the tyre and a wet surface, such as soil, clay (also wet), assuming that the tyre has a high tread, or that tyres with a chain are used in difficult terrain conditions [19, 20].

The analysis was based on two factors: the inclination angle (α = 0 or 17°) and load distribution on the front and rear axles. For a loader driving under full load, a higher load on the front axle was assumed due to the presence of excavated material in the bucket. In this case, the distribution of loads on the front and rear axles was assumed as 80/20, while for the loader without excavated material (Q_{exc} = 0 kg) the distribution of loads reached: 40/60. The calculation results are presented in Table 2.

Table 2

Results of the analysis of wheel slip torque

(M_{ftot} – total torque of loader wheel adhesion

M_{ffw}/M_{frw} – friction moment per single front/rear wheel)

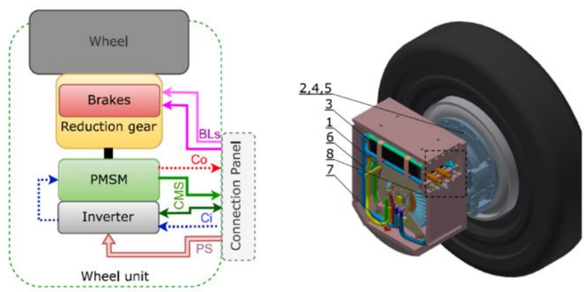
α [°]	0		17	
Q_{tot} [kg]	0	4000	0	4000
M_{ftot} [kNm]	59.7	74.7	57.2	71.5
M_{ffw} [kNm]	**12**	**29.9**	**11.4**	**28.6**
M_{frw} [kNm]	**17.9**	**7.5**	**17.2**	**7.1**

The results of the calculations indicate that for the assumed total weight of the loader, the maximum torque which can be transferred from the wheels to the ground without slipping is approximately 55-75 kNm. The analysis of the distribution of individual wheels' adhesion torques shows that selection of drive subassembly elements that enable generating a torque the value of which approaches or exceeds 30 kNm is pointless, as this torque will not be utilised due to the lack of sufficient adhesion of the wheels to the ground.

The data listed in Tables 1 and 2 indicates that in difficult and extremely difficult off-road conditions, the drive system may not be able to keep the vehicle moving. Importantly, this is not due to the torque deficit, but to the achievable adhesion of the wheels to the ground and a relatively small weight of the loader in relation to the parameters of the drive nodes.

DESIGN AND PARAMETERS OF DRIVE NODES

The construction of the traction system of the designed LHD loader is based on the use of four independent drive units integrated with road wheels (Figure 4).



a) b)

Fig. 4 a) Diagram, and b) Model of the drive unit components' arrangement

1-motor, 2,4,5-transmission with a parking and manoeuvring brake, 3-inverter, 6-cooling system, 7-sensors of conditions in the motor compartment, 8-connection panel.

It was assumed that the design would be based on the COTS (Commercial off-the-shelf) principles; therefore, all the components of the designed system were selected as commercial solutions.

Each of the drive units is an autonomous structure equipped with a brushless motor with permanent magnets, characterized by a nominal torque M_{nom} = 450 Nm, which, combined with a gearbox having a ratio of i = 25, allows obtaining a nominal drive torque on each wheel reaching M_{w_nom} = 11 kNm. The maximum torque of 22 kNm results from the permissible gear load. A comparison of these values with the results of calculations presented in Tables 1 and 2 allows concluding that the obtained range of drive torques is well suited to the drive demand and allows utilizing the loader's tyre adhesion to the ground. The designed drive unit enables achieving the maximum driving speed of the loader V_{LHD_max} = 32.7 km/h.

The applied reduction gear is equipped with two brakes: a multi-disc parking brake (normally closed) and a double-clamp manoeuvring disc brake (normally open). Each of them provides a separate braking torque of 16 and 20 kNm, respectively.

An individual inverter powering the motor has been integrated in each of the drive nodes. Both the motor and the inverter require cooling as they are installed in a closed space. A serial system of cooling liquid circulation has been applied. The refrigerant (a mixture of glycol and water) first passes through the inverter and, then, through the motor. Next, the liquid leaves the drive node through the connection panel and is transferred to the external heat exchanger.

As the designed system is a prototype, each unit will be equipped with sensors of operating conditions, i.e. temperature, humidity and presence of liquid in the drive unit chamber.

An important component of each of the drive nodes is the connection panel, which enables independent disconnection of each drive node. The connection interface includes: a DC power supply connection, connections for two brake lines, control and measuring signal connectors, and a connection for the cooling liquid circuit.

In Fig. 4a, each of the circuits is marked as:

PS – power supply,

Ci/Co – cooling in/out,

CMS – control, monitor and steer signals,

BLs – brake lines.

Figure 5 shows the overall dimensions of the drive unit.

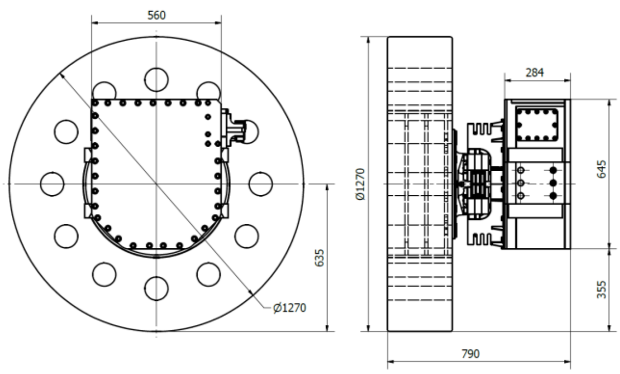


Fig. 5 Overall dimensions of the drive unit

Major design constraints include the need to maintain adequate clearance under the axles of the loader and a limited space in the axial and longitudinal direction of the machine.

For this reason, it is advantageous to use the "axial-flux" motor, which allows compact installation of the drives. Placing the inverter in the upper part of the drive unit enabled obtaining a clearance of 355 mm under the axles of the loader.

DESIGN AND PARAMETERS OF THE ENERGY STORAGE

The energy storage capacity required for powering the drive system has been estimated on the basis of an arbitrarily adopted work cycle. It has been assumed that a single work cycle includes 4 stages:

I. Driving at a constant speed, without excavated material, over a distance of 250 m,
II. Driving with constant acceleration (gaining speed), with excavated material
III. Driving at a constant speed, with excavated material, over a distance of 250 m,
IV. Driving with constant acceleration (gaining speed), without excavated material

Analysis was carried out for driving on a flat terrain and a terrain inclined at an angle of α = 2°, with two sets of speeds v_{low} = 10 and 3 km/h, v_{high} = 15 and 5 km/h. The higher speed corresponded to driving without excavated material, and the lower – to driving with full load. In addition, each scenario was tested with 4 values of the rolling resistance coefficient f = 0.1/0.2/0.35/0.5. The f = 0.1 value corresponded to the rolling of wheels on a straight, paved surface, 0.35 corresponded to difficult terrain conditions, and 0.5 to very high driving resistance (muddy terrain). The simulation scenarios are schematically presented in Fig. 6.

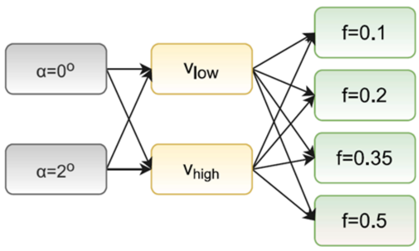


Fig. 6 Variables adopted for the simulation

For each set of data (α, v, f,), the values of the required total driving power were determined in a spreadsheet, taking into account the basic relationships (Eqs. 2-5) and with consideration of the gear ratio and loader parameters.

$$E_{tot} = F_{tot} \cdot v_{tmp} \tag{6}$$

The value of temporary driving power was obtained, for each stage of each work cycle. For the v_{low} scenario, the machine will complete 8 repetitions of duty cycles in an hour, while for the higher driving speed (v_{high}) there will be 13. In this way, the hourly demand for propulsive power was obtained, results were shown in Table 3. A fivefold increase in driving resistance (expressed in terms of the coefficient of rolling friction) results in a 3 to 3.6-fold increase in propulsion power requirements. As can be seen, an increased driving speed results in a significant increase of the drive system energy consumption. Changing the operating speed scheme from v_{low} to v_{high} results in an increase in drive power demand of about 1.6 times.

Table 3

Power demand per one hour of the loader work [kWh/h]

	v_{low}		v_{high}	
	α = 0°	α = 2°	α = 0°	α = 2°
	E_h [kWh/h]			
f = 0.1	29.6	37.1	47.0	59.7
f = 0.2	35.3	50.7	70.3	82.6
f = 0.35	75.4	83.0	122.9	135.4
f = 0.5	107.9	115.9	175.6	188.1

However, it should be noted that in the case of driving with a set of lower speeds (v_{low}), the loader will perform 8 full cycles during one hour of operation, whereas for (v_{high}) it will be 13 cycles, which allows obtaining a capacity of 32 and 52 t/h, respectively, when the assumed bucket capacity is fully used. As can be seen from equations 2-6, the total work efficiency does not depend on the choice of speed profile, but only on whether the loader is moving on an inclined or flat terrain. Taking into account the number of duty cycles performed, it was determined how much drive power was required under various operating conditions. The values of drive power demand required to transport 1 tonne of excavated material (E_t), determined for various operating conditions, have been given in Table 4.

Table 4

Power demand per one tonne of transported spoil (kWh/t)

	α = 0°	α = 2°
	E_t [kWh/t]	
f = 0.1	0.84	1.06
f = 0.2	1.00	1.44
f = 0.35	2.14	2.36
f = 0.5	3.07	3.29

Assuming further that the expected working time of the loader should be at least 8 hours, basing on Table 3 and assuming the most favourable operating conditions (driving at a low speed on a flat terrain), it was estimated that the minimum energy storage capacity for the loader's drive system should reach E = 240 kWh.

A given energy capacity is obtainable with the use of commercial off-the-shelf (COTS) solutions, such as battery modules for electric buses and trucks. More challenging is development of this class machine with comparable dimensions and stability [21].

Due to the parallel usage of four electric motors in the drivetrain, a battery swap nominal voltage was selected in the range of 600-800 VDC, due to two factors:

– Increased supply voltage allows for reduced motor currents (at same power) and reduced wire cross-sections, leading to weight and space reductions;
– Various COTS subsystems/components are limited to supply voltages of 800-850 VDC – higher voltages force use of custom-designed solutions; also voltages above 1kV usually require additional electric permissions from maintenance personnel.

Rechargeable lithium modules (NMC – lithium nickel manganese cobalt oxides, LFP – lithium iron phosphate, LTO – lithium titanate) were mainly considered as a power source for the loader [1]. A closer look was also given to molten salt (SNC, sodium nickel chloride) batteries due to their increased safety [2,3], which is potentially advantageous in underground mining applications. Sodium nickel batteries have lower current capabilities compared to lithium-based batteries of comparable voltage and energy capacity [2] – a disadvantage when supplying a drivetrain of four independent motors.

The following factors were taken into consideration when selecting the type (chemistry) of the battery pack:

– total mass and total volume of the battery cells;
– long-term permissible and peak charging and discharging current;
– durability and safety,
– operational requirements of the cells.

The analyses took into account the data of selected commercially available cells. It was assumed that in the case of energy storage total weight reaching 2 t, the weight of the cells alone should not exceed 1.5 t. In addition, it was assumed that the volume of the cells should not exceed 1 m^3, which results from the adopted maximum size of energy storage and the need to allocate adequate space for electrical equipment and a battery temperature management system. The total number of cells for each considered type was estimated as the product of the number of cells connected in parallel and in series, where:

– the number of cells connected in parallel was selected as the maximum of the values calculated for the following criteria: required energy storage capacity in Ah, required continuous discharge current (determined on the basis of the average drive system power demand), required peak current (estimated on the basis of peak power demand);
– the number of cells connected in series was selected on the basis of the required energy storage voltage. SNC cells were rejected primarily due to insufficient charging and discharging current parameters as well as rather cumbersome operational requirements. Estimated weights and volumes of the cells alone in the case of LFP, NMC, SNC, LTO types of cells are: 2.7 t/1.7 m^3, 1.5 t/0.8 m^3, 1.7 t/0.85 m^3, 3.5 t/1.65 m^3, respectively. This comparison shows that the criteria of mass and volume simultaneously meet only the NMC cells. They also guarantee an acceptable level of safety and service life.

Operational requirements for NMC cells come down to providing them with a BMS (Battery Management System) and a BTMS (Battery Thermal Management System). The NMC cells are widely applied in electromobility (buses, heavy machinery). On the market there are modular battery pack solutions, equipped with BMS systems and liquid heat exchangers, as well as active BTMS systems, which cool or heat the battery pack depending on environmental and operating conditions. For safety reasons, the available battery packs are approved according to, among others, UN ECE R100. Due to the relatively large capacity of the energy storage and the need to ensure effective balancing of the cell, it was assumed that the storage would be built of three chains (packages) of NMC cell, each of them working in parallel, under the control of its own BMS system.

Apart from the selection of the type of lithium-ion cells and the most advantageous commercially available solutions (battery modules, BTMS system), an additional design challenge was to properly apply these solutions and functionally arrange them in the space intended for energy storage in the machine. Major implementation aspects to be solved included: selection of the master controller, selection of a set of sensors to increase the security of the energy storage in the machine operating under-conditions of high environmental exposure, selection of elements ensuring power distribution, selection of the electrical connection between the energy storage and the machine, selection of the charging standard and charging controller of the energy storage, selection of elements ensuring the functional safety of the energy storage and selection of the communication and telemetry system. Another challenge was to design a functional housing with a locking system in the machine.

A separate issue is ensuring the fire safety of the vehicle. This concerns both the battery vehicle itself [25] and the operating conditions [26, 27]. At the current stage of the project, the focus is on the selection of energy storage components, in terms of whether they are homologated (UN ECE R100.02) and approved for use in off-highway, mining vehicles [28].

The above problems were solved mainly by using solutions available on the market. Among others, the following were selected: a PLC dedicated to mobile machines, dedicated battery safety sensors, a safety relay supervising the E-stop function and an electrical connector with a total long-term current carrying capacity of HV power contacts reaching 600 A, which also contained low-voltage, communication, control and HVIL (Hazardous/High Voltage Interlock) circuit contacts. The energy storage will be charged in the CCS Type 2 standard. It has been assumed that the energy storage will be fully charged within 2 hours, which requires a 120 kW charging station. The assumed charging time of the energy storage has been correlated with the operating regime of the machine and seems to be sufficient. A relatively small charging current of the energy storage – 0.5C will enable correct balancing of the cells in the energy storage.

Table 5 lists the achievable working times of the loader, assuming that it is equipped with a battery of 240 kWh capacity. The power requirements of the loader's hydraulic and auxiliary systems have not been taken into consideration in the calculations.

Table 5

Expected operating time (t_{op}) of the loader equipped with a 240 kWh battery

	V_{low}		V_{high}	
	α = 0°	α = 2°	α = 0°	α = 2°
	t_{op} [h]			
f = 0.1	**8.0**	6.4	5.1	4.0
f = 0.2	6.7	4.7	3.4	2.9
f = 0.35	3.2	2.9	1.9	1.8
f = 0.5	2.2	2.1	1.4	1.3

RESEARCH RESULTS

The designed drive system meets the adopted design assumptions. The required parameters of the system have been estimated and the components selected on the basis of the COTS assumptions.

In order to ensure the machine's ability to work even in difficult terrain conditions, it must be provided with a drive system the parameters of which are unparalleled in this class of machines (i.e. lightweight mining loaders). The developed design of the drive system is characterized by a very high total drive torque, available from the lowest rotational speed values.

The calculated demand for drive power and torque depend to a large extent on the adopted operating conditions. It is worth mentioning that the model description of field conditions is based on a number of simplifications. The actual operating conditions will have a key impact on the loader's performance. The results of the analyses presented in this study will be verified during field tests of the machine prototype.

A comparison of the obtained simulation results with the parameters of competitive solutions (combustion and electric) indicates that the demand for drive power and energy storage capacity is lower than the calculated one.

The estimated capacity of the energy storage is sufficient to ensure the required working time. The loader is equipped with a quick battery replacement system, which in the case of operation under conditions of increased demand for driving power will allow maximizing the efficiency of the machine. In addition, it should be noted that the drive system balance did not take into account the potential energy yields due to the use of regenerative braking in the system.

CONCLUSIONS

The paper presents an analysis of the possibility of designing a drive system for a lightweight LHD BEV loader equipped with an "in-wheel drive" system. The use of four independent drive nodes is an innovative solution compared to available loaders with similar functional properties.

Due to the lack of available benchmark data and publicly available, representative models of the operation of drive systems in LHD vehicles, enabling accurate calculation of power demand for powering this type of vehicle, estimate calculations were carried out in accordance with the principle of staying on the safe side of the model, i.e. the assumed operating conditions and simplifications used in the computational models were selected so as to enable the loader's driving in each case.

Implementation of the design of a loader equipped with four independent drive systems, installed in the immediate vicinity of the road wheels, poses a significant design challenge.

This is, among others, related to: a significant limitation of the available space for installing the systems, the need to ensure the required thermal conditions for the drive systems and the distribution of electric power on the vehicle – difficulties associated with the arrangement of power and cooling cables, resulting from their high rigidity and the need to run them in the immediate vicinity of each wheel.

Due to the prototype nature of the designed system, it is planned to build a test stand (for testing drive systems), and carry out operational tests of the loader prototype.

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REFERENCES

[1] Bołoz Ł. Global Trends in the Development of Battery-Powered Underground Mining Machines. *Multidiscip Asp Prod Eng,* 2021, 4, 178-89. https://doi.org/10.2478/mape-2021-0016.

[2] Bołoz Ł, Castañeda LF. Computer-Aided Support for the Rapid Creation of Parametric Models of Milling Units for Longwall Shearers. *Manag Syst Prod Eng,* 2018, 26, 193-9. https://doi.org/10.1515/mspe-2018-0031.

[3] Eger T, Stevenson J, Callaghan JP, Grenier S, VibRG. Predictions of health risks associated with the operation of load-haul-dump mining vehicles: Part 2 – Evaluation of operator driving postures and associated postural loading. *Int J Ind Ergon,* 2008, 38, 801-15. https://doi.org/10.1016/j.ergon.2007.09.003.

[4] Eger T, Stevenson J, Boileau P-É, Salmoni A. Predictions of health risks associated with the operation of load-haul-dump mining vehicles: Part 1 – Analysis of whole-body vibration exposure using ISO 2631-1 and ISO-2631-5 standards. *Int J Ind Ergon,* 2008, 38, 726-38. https://doi.org/10.1016/j.ergon.2007.08.012.

[5] Fugiel A, Burchart-Korol D, Czaplicka-Kolarz K, Smoliński A. Environmental impact and damage categories caused by air pollution emissions from mining and quarrying sectors of European countries. *J Clean Prod,* 2017, 143, 159-68. https://doi.org/10.1016/j.jclepro.2016.12.136.

[6] Hartman HL, Novak T, Gregg AJ. Health hazards of diesel and electric vehicles in an underground coal mine. *Min Sci Technol,* 1987, 5, 131-51. https://doi.org/10.1016/S0167-9031(87)90365-3.

[7] Biały W, Bołoz Ł, Sitko J. Mechanical Processing of Hard Coal as a Source of Noise Pollution. Case Study in Poland. *Energies* 2021, 14, 1332. https://doi.org/10.3390/en14051332.

[8] Priyadarshini Nayak N. LHD-LPDT cycle time optimization of underground metal mine. Mater Today Proc, 2023, 72, 2817-21. https://doi.org/10.1016/j.matpr.2022.07.091.

[9] Dragt BJ, Camisani-Calzolari FR, Craig IK. Modelling the dynamics of a load-haul-dump vehicle. IFAC Proc, 2005, 38, 49-54. https://doi.org/10.3182/20050703-6-CZ-1902.01390.

[10] Ye W, Shen W, Qian Z, Zheng J. Robust longitudinal motion control of underground mining electric vehicles based on fuzzy parameter tuning sliding mode controller. *Comput Electr Eng,* 2022, 98, 107683. https://doi.org/10.1016/j.compeleceng.2022.107683.

[11] Burd JTJ, Moore EA, Ezzat H, Kirchain R, Roth R. Improvements in electric vehicle battery technology influence vehicle lightweighting and material substitution decisions. *Appl Energy,* 2021, 283, 116269. https://doi.org/10.1016/j.apenergy.2020.116269.

[12] Ivanov V, Savitski D, Shyrokau B. A Survey of Traction Control and Antilock Braking Systems of Full Electric Vehicles With Individually Controlled Electric Motors. *IEEE Trans Veh Technol,* 2015, 64, 3878-96. https://doi.org/10.1109/TVT.2014.2361860.

[13] Nasri A, Gasbaoui B, Fayssal BM. Novel Four Wheel Drive Propulsion System Control Using Backstepping Strategy. *Procedia Technol,* 2016, 22, 509-17. https://doi.org/10.1016/j.protcy.2016.01.110.

[14] Cui H, Ruan J, Wu C, Zhang K, Li T. Advanced deep deterministic policy gradient based energy management strategy design for dual-motor four-wheel-drive electric vehicle. *Mech Mach Theory,* 2023, 179, 105119. https://doi.org/10.1016/j.mechmachtheory.2022.105119.

[15] Xu W, Zhao H, Ren B, Chen H. A regenerative braking control strategy for electric vehicle with four in-wheel motors. 2016 35th Chin. Control Conf. CCC, 2016, 8671-6. https://doi.org/10.1109/ChiCC.2016.7554741.

[16] Artisan Vehicle Systems | The World's Most Powerful Industrial Electric Vehicles. n.d. https://artisanvehicles.com/ (accessed March 24, 2023).

[17] Mining Equipment, Parts & Services. Sandvik Min Rock Technol n.d. https://www.rocktechnology.sandvik (accessed March 24, 2023).

[18] Epiroc. Epiroc in your region. Epiroc n.d. https://www.epiroc.com/sys/splash (accessed March 24, 2023).

[19] Blundell M, Harty D. The Multibody Systems Approach to Vehicle Dynamics. Elsevier; 2015.

[20] Schramm D, Hiller M, Bardini R. Vehicle Dynamics. *Modeling and Simulation*. Berlin: Springer Berlin, Heidelberg; 2014.

[21] Bołoz Ł, Kozłowski A, Horak W. Assessment of the Stability of BEV LHD Loader. Manag Syst Prod Eng, 2022, 30, 377-87. https://doi.org/10.2478/mspe-2022-0048.

[22] Nazir MH, Rahil A, Partenie E, Bowkett M, Khan ZA, Hussain MM, et al. Comparison of lithium-ion battery cell technologies applied in the regenerative braking system. Battery Energy, 2022, 1, 20220022. https://doi.org/10.1002/bte2.20220022.

[23] The Salt Battery – Mining n.d. https://www.fzsonick.com/applications/mining (accessed March 28, 2023).

[24] Salt Batteries: Opportunities and applications of storage systems based on sodium nickel chloride batteries | Think Tank | European Parliament n.d. https://www.europarl.europa.eu/thinktank/en/document/IPOL_IDA(2023)740064 (accessed March 28, 2023).

[25] Dorsz A, Lewandowski M. Analysis of Fire Hazards Associated with the Operation of Electric Vehicles in Enclosed Structures. *Energies*, 2022, 15, 11. https://doi.org/10.3390/en15010011.

[26] Halim A, Lööw J, Johansson J, Gustafsson J, van Wageningen A, Kocsis K. Improvement of Working Conditions and Opinions of Mine Workers When Battery Electric Vehicles (BEVs) Are Used Instead of Diesel Machines – Results of Field Trial at the Kittilä Mine, Finland. *Min Metall Explor*, 2022, 39, 203-19. https://doi.org/10.1007/s42461-021-00506-8.
[27] Li YZ, Ingason H. Overview of research on fire safety in underground road and railway tunnels. *Tunn Undergr Space Technol,* 2018, 81, 568-89. https://doi.org/10.1016/j.tust.2018.08.013.
[28] Global Mining Guidelines Group (GMG), Recommended Practices for Battery Electric Vehicles in Underground Mining, n.d. https://gmggroup.org/ (accessed May 10, 2023).

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