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Selection of Transmission and Electrical Motor Parts for ATVs

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Abstract. The article presents the results of calculations to justify the choice of transmission components and traction electric drive. The choice of the parameters of the gearing of the transfer case and differential has been substantiated. The choice of bearings is substantiated from the conditions of ensuring the given resource and strength for the transfer case and main gear.

1 Calculation of the capacity of traction batteries

The developed ATV with an electromechanical transmission and a 6x6 wheel arrangement is designed to transport people and small-sized cargo in off-road conditions, except for virgin snow with a snow depth of more than 250 mm and is characterized by the ability to overcome natural and artificial obstacles, such as slopes, slopes, trenches.

The main factors affecting the capacity and the determined mass of the batteries installed on the ATV are the range, the type of battery cells, the nominal voltage of the DM (drive motor). Table 1 provides a quick comparison of the most common battery cells.

Table 1. Comparison of battery cells.

Cell type	NMC	LCO	LTO	NCM	LFP	NMC
Energy intensity, Ah	15	10	12	30	10	15
Voltage, V	3,6	2,4	2,3	3,7	3,2	3,6
Weight, g	320	300	400	615	312	320
Specific energy consumption, Wh / kg	170	120	85	180	103	170
MAX continuous discharge / charge current	3C/4C	4C	6C	3C	3C	3C/4C
Service life, cycles	5000	10000	20000			5000
Working temperature	-20C +60C			-40C +60C	-20C +60C	-20C +60C

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To assess the operational properties of the ATV, such as the dynamics of acceleration, the climbs to be overcome, as well as to determine the capacity of the energy storage, a mathematical model has been developed [1].

The mathematical model of an ATV with an electromechanical transmission consists of the following subsystems: chassis model, traction electric motor, transmission, and driver model (Figure 1). The chassis model includes models of the translational motion of the vehicle body, the rotational motion of the wheel propeller, and the model of the interaction of the wheel with the supporting surface (Figure 2). The DM is modelled using the Servomotor element of the Simscape / Electronics library (Figure 3). The initial data for the DM model are the dependence of the torque on the rotational speed and the moment of inertia of the motor rotor. The input signal is the required torque on the motor shaft. The Servomotor element is connected to the power supply model using electrical connections (blue lines), and to the transmission model using mechanical connections (green lines). The transmission mathematical model is made up of elements of the Simscape / Driveline library and includes the transfer case and final drive models. The energy store is modelled using the Generic Battery element of the Simscape / Electronics library (Figure 1). The initial data are the main parameters of the drive: nominal voltage, capacity (it is possible to set an infinite capacity, while instead of the battery charge indicator, the energy consumption for movement will be calculated), internal resistance, voltage at a low charge level. The driver model is needed to maintain the target speed and is a proportional controller.

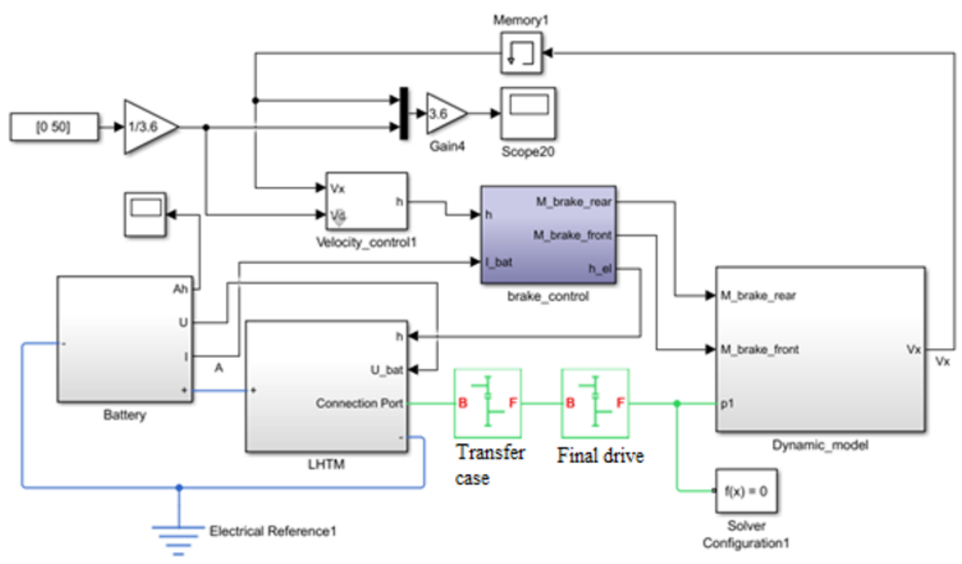

Fig. 1. Block diagram of an ATV model with one DM.

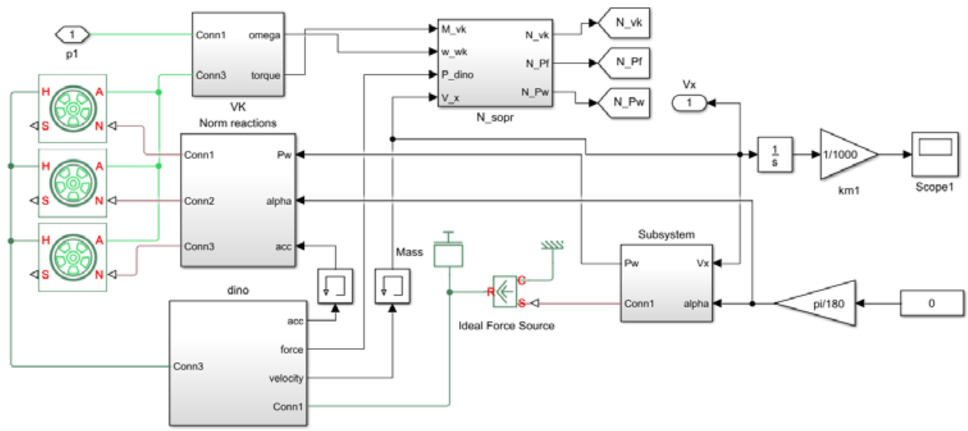

Fig. 2. Block diagram of an ATV chassis model.

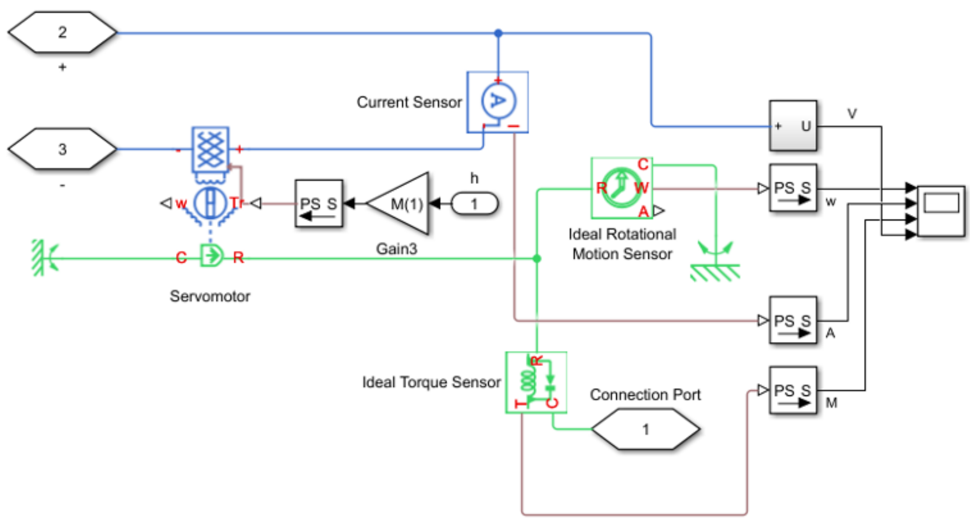

Fig. 3. Block diagram of the DM model.

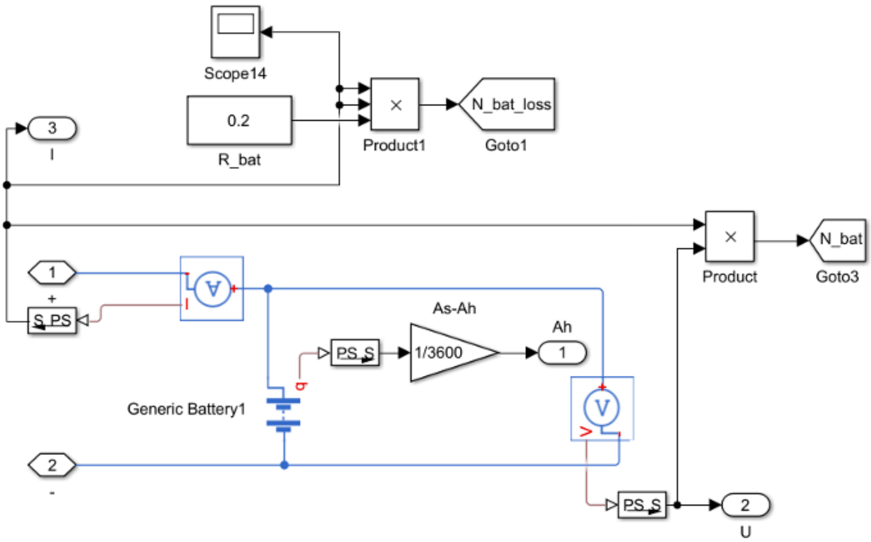

Fig. 4. Block diagram of the battery model.

Figure 5 shows the acceleration performance of an ATV on an asphalt road (rolling resistance coefficient f = 0,018).

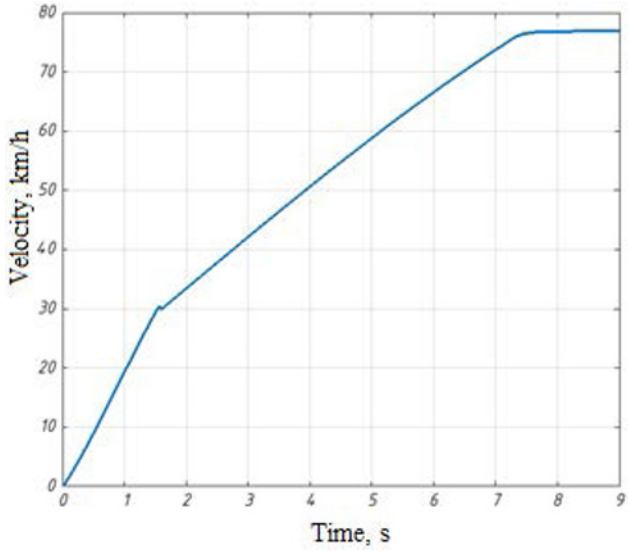

Fig. 5. Acceleration characteristic of the ATV.

The calculation of the capacity of the batteries was carried out based on the provision of a power reserve of 100 km at a constant speed of 60 kph on the support base with a movement resistance coefficient of 0,018 and an adhesion coefficient of 0,8 [2]. The simulation was carried out with a model of an energy storage unit with unlimited capacity, while the resulting negative battery charge is equal to the consumed energy. As a result of the simulation of the movement of the ATV with the ERMAX 228 traction motor, the energy consumption was 12,9 kWh. According to the rules for operating batteries, discharge below 20% is not allowed, respectively, the minimum battery capacity required to provide a power reserve of 100 km under these driving conditions is 15,4 kWh.

2 Design of the transfer case chain drive

The transfer case (TC) is a transmission unit for a wheeled vehicle, designed to separate the power flow and distribute it between the driving axles. The electric ATV uses a two-stage transfer case with a chain drive and a lockable centre differential.

The calculation of the chain transmission was carried out according to the method proposed in [3]. The transfer case uses Renold gear chains, so the selection of the necessary parameters of the chain drive was made considering the recommendations of the manufacturer. The calculation of the chain drive was made from the condition of ensuring the gear ratio u = 2,1. The parameters of the selected chain drive are shown in the Table 2.

Table 2. Parameters of the chain transmission of the transfer case.

Name parameters	**Value**
Chain pitch P, mm	9,525
Number of teeth of the driven sprocket z_1	19
Number of teeth of the driven sprocket z_2	41
Gear ratio u	2,16
Center distance a_w, mm	177,841
Nominal chain width b_a, mm	26,6

3 Calculation of the gearing of the transfer case

To assess the durability of gear wheels, the load spectra are taken, like the 4x4 ATV [4-11]. The calculation was carried out using specialized software. The results of calculating the parameters of cylindrical wheels are shown in Table 3.

Table 3. Parameters of the cylindrical gearing of the transfer case.

Name parameters	**Value**
Normal module m_n, mm	2
Tooth profile angle in normal section α_n, deg	22,5
Number of gear teeth z_1	21
Number of wheel teeth z_2	32
Working width of the tooth when calculating contact stresses in b_{w}, mm	22,0
Working width of a gear tooth when calculating bending stresses b_{f1}, mm	22,0
Working width of a wheel tooth when calculating bending stresses b_{f2}, mm	22,0
Tooth line angle β, deg	24
The degree of accuracy according to the standards of smoothness	6
Wheel hardness, HRC	61
Material, gear processing	20XH3A
Material, wheel processing	20XH3A

The results of the strength calculation for the most loaded second pair of gearing are presented in Table 4.

Table 4. The results of the strength calculation of the gear wheels of the transfer case.

Loading type	Gear	Factor of safety	Factor of safety until yield point
Static force	leading	5,22	3,7
	slave	5,05	3,58
-	-	Safety factor for contact voltages	Safety factor for bending stresses
Tooth profile angle in normal section αn, deg	leading	1,004	1,996
	slave	1,017	1,872

Analyzing the results obtained; we can conclude that the gear wheels of the transfer case meet the strength requirements.

4 Calculation of gears of the transfer case differential

The transfer case uses a differential similar to the differential used in the final drive [12]. The differential of the main gear during operation is loaded with a large moment, therefore, a separate calculation of the differential for the operating conditions of the main gear will not be carried out.

5 Calculation of the transfer case bearings

To assess the durability of the transfer case bearings, the load spectra are taken, like the load spectra of the transmission of a 4x4 ATV. The layout of the transfer case bearings is shown in Figure 6.

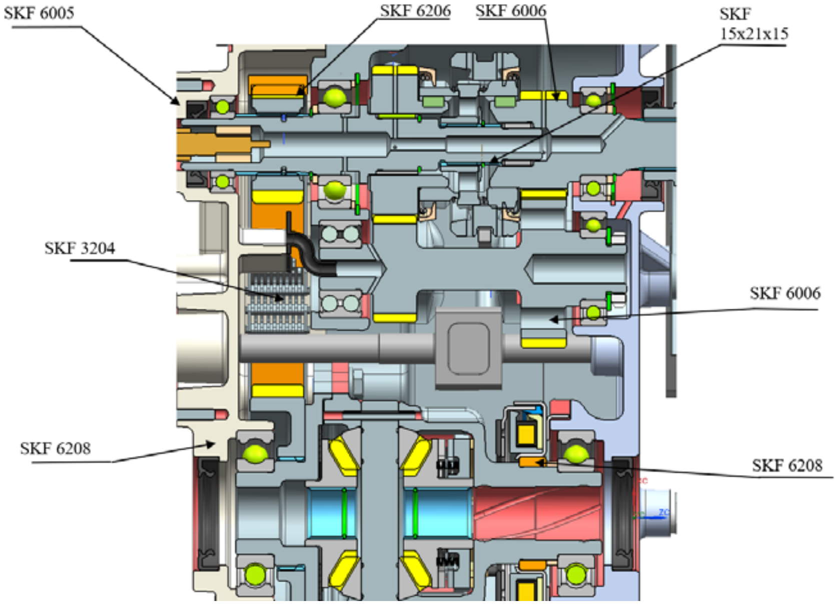

Fig. 6. Location of bearings in the transfer case.

Calculation of bearings for static strength and durability was carried out in a specialized software package. The initial data for the calculation are the spectra of the moments, revolutions and runs of the bearing in each gear in the RK for each type of road. The calculation was carried out only for the lower gear, as the most loaded. The calculation results for the transfer case bearings are presented in Table 5.

Table 5. Calculation results of the transfer case bearings.

Collar	Payload		Result	
	Dynamic, C [kN]	Static, C0 [kN]	Minimum static safety factor	Minimum service life, hours
SKF 6006 (main drive shaft)	13,8	8,3	3,11	460
SKF 6206	20,3	11,2	1,33	428
SKF 6005	11,9	6,55	4,94	22545
SKF 6006 (counter shaft)	13,8	8,3	1,07	40
SKF 3204	20,4	12,9	2,02	817
SKF 6208 (from the rear axle)	32,5	19	3,26	9646
SKF 6208 (from the front axle)	32,5	19	1,39	2535
SKF 15x21x15	13,8	16,3	2,69	440

From Table 5 the selected bearings meet the operating conditions.

6 Conclusion

The choice of the components of the traction electric drive transmission is substantiated by means of various calculations. Thus, when calculating the capacity of the batteries, carried out from the condition of ensuring a range of 100 km at a constant speed of 60 kph on the support base with a movement resistance coefficient of 0,018 and a friction coefficient of 0,8, because of modeling the movement of an ATV with an ERMAX 228 traction motor, the energy consumption was 12,9 kWh, which corresponds to the minimum battery capacity required to provide a range of 100 km under these driving conditions, 15,4 kWh. When analyzing the results obtained when calculating the gearing of the transfer case the following conclusion is obtained: the gear wheels of the transfer case meet the strength requirements. As a result of calculating the bearings of the transfer case, those are selected that meet the resource requirements and working conditions.

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