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Mechanical conditions for a prototype of battery electric vehicle in the *Shell-eco* marathon competition

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ABSTRACT

The Shell Eco-Marathon is a competition for students, created by Shell, the British hydrocarbon energy company, and is held in the USA, Brazil, Europe and Asia. The challenge is to build the most energy-efficient vehicle. It seeks to transform the way mobility is viewed by allowing the use of alternative fuels, including electric or hybrid engines [1]. For the team, competition is the opportunity to propose efficient transportation in daily trips in a congested city (home-work-home), where reaching high speed is a utopia, so, low consumption becomes more important in addition to mitigating the impact on the environment and resulting in an economic benefit. Therefore, for the selection of the vehicle's engine, the balance of weight-power-consumption was found in the existing engine options.

1. INTRODUCTION

The Shell Eco-Marathon is a competition for students, created by Shell, the British hydrocarbon energy company, and is held in the USA, Brazil, Europe and Asia. The challenge is to build the most energy-efficient vehicle. It seeks to transform the way mobility is viewed by allowing the use of alternative fuels, including electric or hybrid engines [1].

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2. Methods

2.1 Vehicle physical parameters

The evaluation of mechanical conditions is performed on a prototype category vehicle as shown below, which is driven by an electric motor.

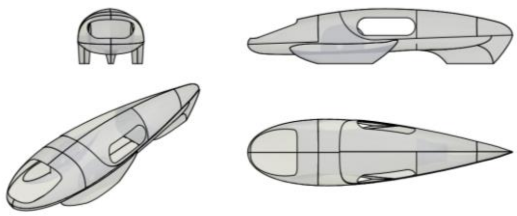

Figure 1. Views of the vehicle body.

In order to obtain more approximate physical parameters of the complete vehicle, two (2) blocks are inserted to the assembly, which allow to simulate how the weight of the driver and the weight of the powertrain affect the moment of inertia and how much the center of mass is displaced.

The weight of the powertrain accounts for 15% of the total weight of the vehicle, since it is composed of the engine, batteries, transmission and axles, among other components that generate the energy that is then converted into motion.

On the other hand, the body is divided into two parts, the main body and a cover, which account for 11.032% and 3.946% respectively of the final weight of the vehicle.

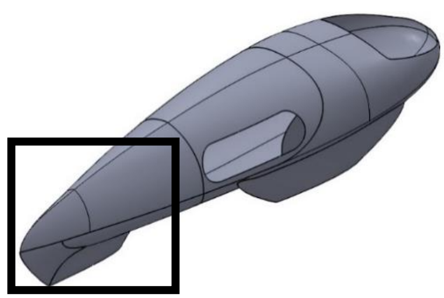

Fig 2. Motor location.
Figure 2. Two parts of the body.

The center of mass of the vehicle is located as shown in the following figure:

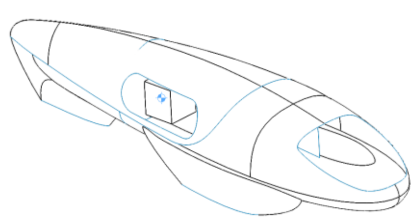

Figure 3. Center of mass of the vehicle

Thanks to the SolidWorks "Physical Properties" tool, the data of the vehicle's mass inertia, and the frontal area of the vehicle are obtained and presented in the table together with the other physical parameters.

Table 1 Physical parameters

Constant	Symbol	Value	Units
Vehicle mass	m_v		Kg

Pilot mass	m_p		Kg
Number of wheels	n_r		-
Wheel radius	r_r	0.660	m
Vehicle moment of inertia	J_v	1.334	kg m^2
Engine torque	K_t	0.110	N m / A
Air density	ρ	1.225	kg/m^3
Front area of the vehicle	a_f	0.355	m^2
Aerodynamic drag coefficient	C_d	0.164	-
Gravitational acceleration	g	9.810	m/s^2
Rolling resistance coefficient	C_r	0.003	-

2.2 Vehicle dynamics

2.2.1 Vehicle longitudinal dynamics

The longitudinal dynamics of the vehicle corresponds to the behavior of the system on a plane, viewed from one side of the vehicle, parallel to the direction of motion.

Such behavior is described by Newton's second law as presented in equation 1 [2].

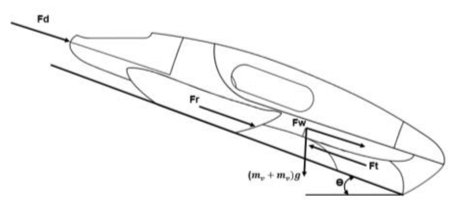

Figure 4. Longitudinal free body

$$Fx - (Fd + Fr + Fw) = (m_v + m_p)a,$$
$$Ft = \frac{Kt\,\eta_g \Omega\, i(t)}{r_r},$$
$$Fd = \frac{1}{2} Cd\, \rho\, a_f\, V^2,$$
$$Fr = C_r(m_v + m_p)gCos\theta,$$
$$Fw = (m_v + m_p)gSen\theta,$$

The figure shows the free body diagram for the vehicle seen laterally, where the forces acting on the vehicle are observed; m_v is the mass of the vehicle, m_p is the mass of the driver,Ω is the gear ratio,η_g is the transmission efficiency, Fd is the aerodynamic force, Fr is the frictional resistance, and Fw is the slope resistance or the weight component on the analysis axis, Ft is the traction force transmitted by the engine, if this force manages to overcome the others, the vehicle moves.

2.2.2 Ackerman Principle

On the other hand, Ackerman's principle allows to analyze the relationship between the steering angle required by the vehicle and the orientation of the wheels, the principle states that "the steering angle at the steering wheel can allow to determine the instantaneous radius of curvature that the vehicle will have" [3].

One aspect to take into account in this project is that the designed vehicle has only three (3) wheels, so a simplification is made in the rear axle, which can be represented as a single wheel, but positioned in the center, as shown in the diagram (Figure 5).

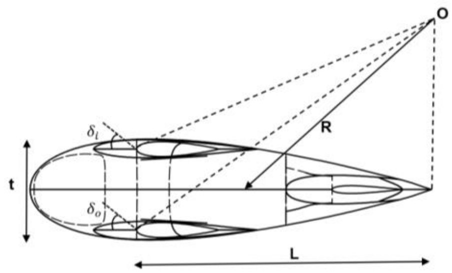

Figure 5. Upper free body

$$cot\delta_o - cot\delta_i = \frac{t}{L},$$
$$\delta_i = \arctan\left(\frac{2L\, tan\delta_a}{2L - d\, tan\delta_a}\right),$$
$$\delta_o = \arctan\left(\frac{2L\, tan\delta_a}{2L + d\, tan\delta_a}\right),$$
$$R = \frac{L}{\tan\delta_a},$$
$$\omega_v = \frac{V_d}{R},$$

The figure shows the free body diagram for the vehicle seen from the top, where the two required angles are observed δ_i y δ_o in function of a radius of curvature R. To know the magnitude of the angles in the wheels, it is necessary to know the distance between the axles L, and with this it is also known the width between wheels, called t.

2.3 Track modeling

The modeling of the track is an extremely important aspect, as it provides information on how the powertrain will behave, specifically on how the energy consumption will change, if the vehicle will require more or less power for its movement. For this project, the Paul Armagnac de Nogaro speed circuit in France, which has an extension of 3,636 km, was taken into account.

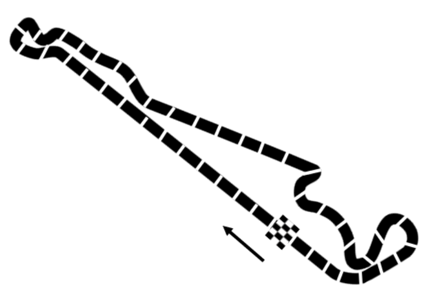

Figure 6. The track modeling: Paul Armagnac circuit

This circuit was divided into key sections, in which the angle (SLOPE) behaves approximately constant, in order to be able to model the track in valleys or slopes and also in flat terrain, the values in radians are those shown in the graph, where the valleys and slopes are simulated according to the track section.

2.4 Powertrain transmission

The transmission of power in the vehicle is an important aspect in the design, because it depends on this that the power in the engine is maximized and not have losses that affect energy consumption increasing unfavorably, for power transmissions have chain/pinion and belt/pulley systems, both effective and with advantages over the other in certain respects; however, the first requirement to be taken into account is avoiding losses and how to maximize the efficiency of the system in the smallest details, the belt/pulley system has a higher probability of power loss with respect to the chain system either by slippage or because it must be in constant tensioning to take advantage of its usefulness [4], this simple reason requires to choose the chain/pinion transmission, which was designed.

2.4.1 Stress transmission

To find the linear speed of the wheel, the primitive diameter Dp and the angular velocity of the wheel ware expressed.

$$Vb = \frac{Dp * \omega}{2},$$

The stress in the chain joints, as well as the stress in the sprocket Gn, are obtained from the maximum chain stress F0, the number of chain joints n, and φ α.

$$Fn = F0 \frac{sen\ \varphi}{sen\ (\alpha + \varphi)}\ n,$$
$$Gn = F0 \frac{sen\ \varphi}{sen\ (\alpha + \varphi)}\ n^{-1},$$

The maximum chain stress is expressed as the sum of the useful stress associated with the transmitted torque Fu, the stress associated with the chain centrifugal force Fc and the stress associated with the chain weight Fp.

$$F0 = Fu + Fc + Fp,$$

The stress associated with the transmitted torque is obtained with the power transmitted by the chain and the average chain speed.

$$Fu = \frac{P}{v},$$

The stress associated with the centrifugal force of the chain is obtained with M, the unit mass of the chain in kg/m and v, the average linear velocity of the chain.

$$Fc = M * v^2,$$

2.4.2 Transmitted power

The power transmitted by the chain can be expressed as the useful stress associated with the transmitted torque Fu, multiplied by the average linear speed of the chain.

$$P = Fu * v,$$

The average speed can be expressed as a function of the angular velocity of the wheel and the primitive diameter.

$$v = \frac{Dp * \omega}{2},$$

However, for the calculation and design of the drive chains, the corrected design power (Pc) will be used.

$$Pc = K1 * K2 * K3 * K4 * K5 * P,$$

Illustrations 1,2,3,4 and 5 are used and Table 2 is used to know the K coefficients required for the power correction [4].

The following illustrations are used to calculate the K1 and K2 coefficients

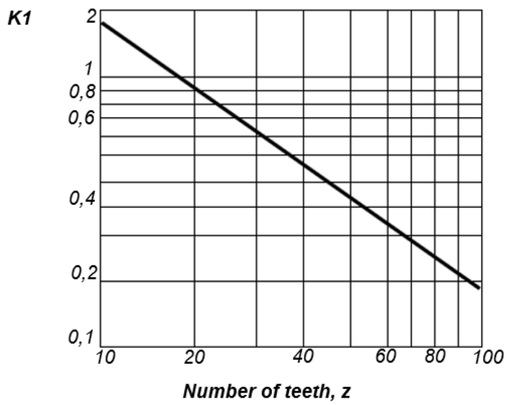

Figure 7. 1 k1

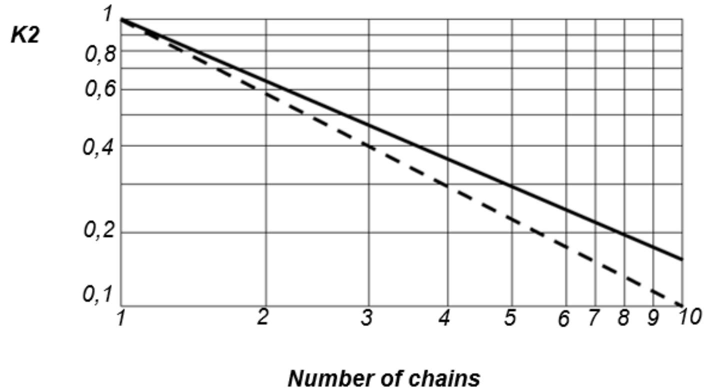

Figure 8. Illustration 2 k2

K2 is the multiplicity coefficient that takes into account the number of chains used in the transmission, whether it is a single, double or triple transmission.

The following tables are used to calculate K3 and K5:

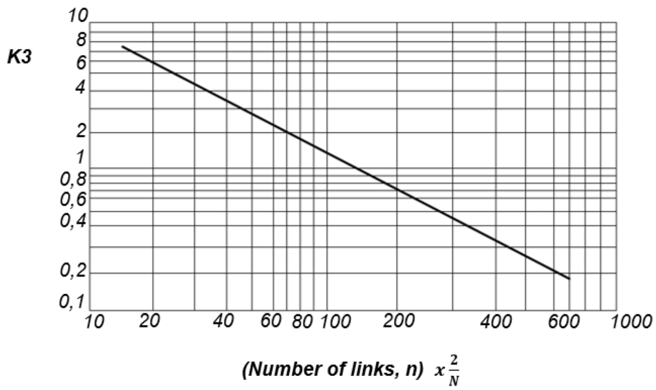

Figure 9. k3

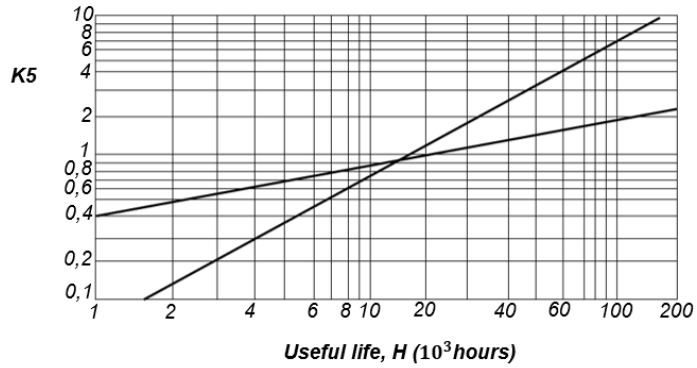

Figure 10. k5

K3 takes into account the number of links in the chain, where n is the number of links and N is the number of sprockets; K5 is the service life coefficient based on the expected service life of the chain.

K4 is the service factor, which takes into account the drive mechanism and transmission application, and is described in the following table:

Table 2. k4

K4 Coefficient, Service Factor			
Receivers	**Electric motor**	**Internal combustion engine with 4 cylinders and more**	**4 cylinders maximum**
-Pumps and compressors -Liquid agitators	1,00	1,10	1,30
-Concrete mixers -Compressors with 3 pistons	1,40	1,50	1,70
-Cylinder hammer mills -Work uses -Oil drilling	1,80	1,90	2,10

With the value of the corrected power Pc and the speed of rotation of the small wheel, the series and type of chain required are selected from Figure 11.

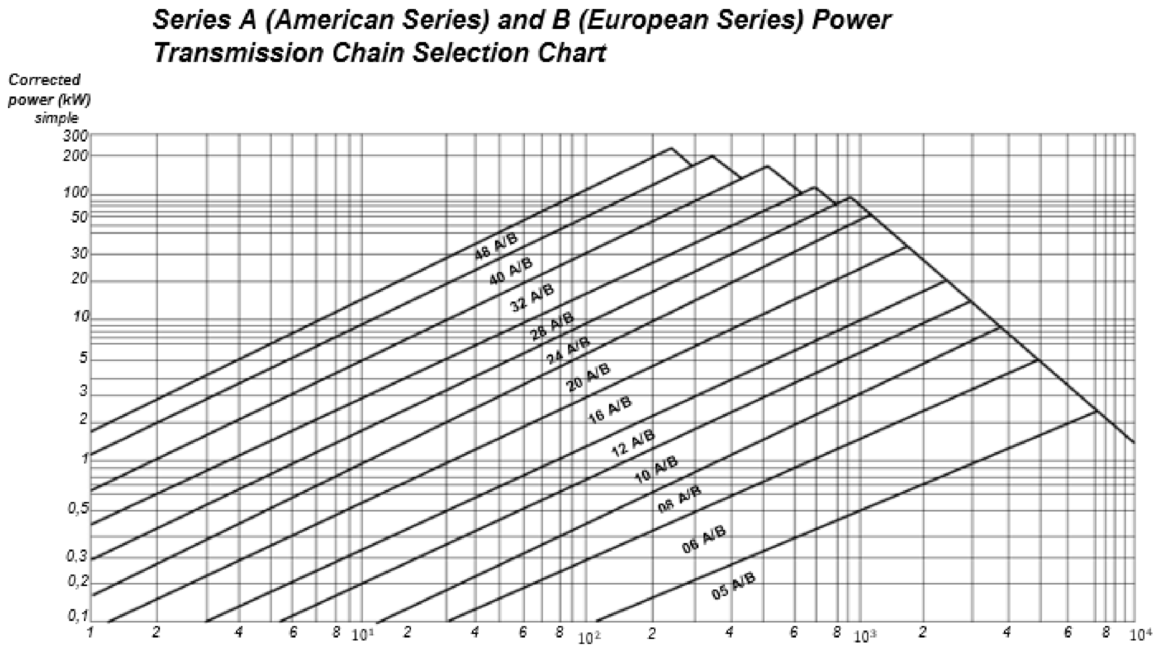

Figure 11. Chain selection

2.4.3 Wheel diameter calculation

Given the type of chain and the pitch, the diameter of the sprockets Dp1 and Dp2 can be calculated,

$$Dp = \frac{p}{sen\,(\frac{\pi}{z})},$$

Where p is the chain pitch in mm, and z is the number of sprocket teeth.

2.4.4 Calculation of chain length

$$\frac{L}{p} = \frac{Z1 + Z2}{2} + (Z2 - Z1)\frac{\beta}{\pi} + O1O2 * \ cos\beta * \frac{2}{p},$$

Where L is the total length of the chain in mm, Z1 is the number of teeth of the sprocket, Z2 is the number of teeth of the large sprocket and O1O2 is the distance between wheel centers in mm.

B is the contact angle in radians, analytically it can be obtained from the following expression:

$$\beta = sen^{-1}\frac{R2 - R1}{O1O2}$$

R2 and R1 being the respective radii of the gears.

2.4.5 Checking the total stress supported by the chain

$$Cs = \frac{R}{F0},$$

The ultimate load or ultimate stress.

2.5 Mathematical model of the engine

The purpose of this section is to mathematically model the direct current motor selected to power the entire propulsion system of the vehicle, through its differential equations, transform them to transfer functions, represent it by state equations and simulate it in Matlab (Simulink).

A direct current dynamo (DC Motor) consists of an element with armature, brushes and field coils in series, parallel or combined, which is intended to be a power converter [5].

The motor is represented with an electrical section, which represents a series resistance R, an inductance L which refers to the inductance of the armature coil, and of course a power supply V which represents the voltage generated by the armature. It should be noted that the current of the armature i flows through this circuit; on the other hand is the mechanical section, from where the motor which is responsible for rotating an axis, this axis represented by W (angular velocity) and Tm (torque generated by this engine), within the engine must consider the load to which it will be required, it is represented as J corresponding to the moment of inertia, when the engine experiences the load on it, the friction coefficient appears, so it is an aspect that can be presented and must be taken into account and is represented as B [6].

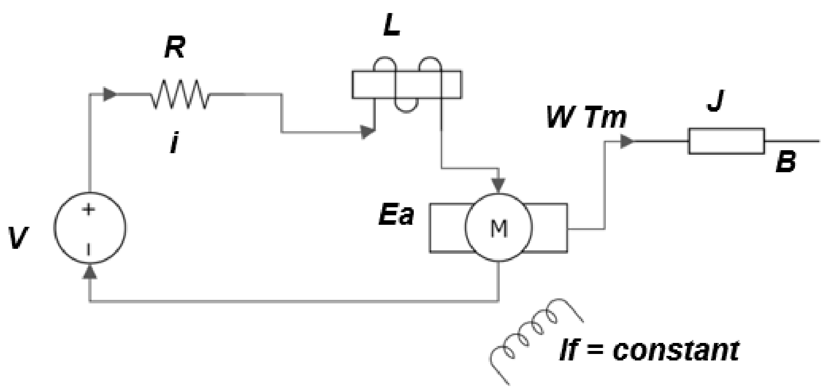

Figure 12. Electromechanical system of the vehicle

As can be seen, the system is electromechanical, so its result can only be obtained if two (2) equations are solved, one for the electrical part and the other for the mechanical part;

$$L\frac{di(t)}{dt} = v(t) - R_i(t) - E_a(t),$$
$$J\frac{dw(t)}{dt} = Tm(t) - Bw(t),$$

Ea(t) is a voltage generated that results when the armature conductors move through the field flux set by the if field current.

It is assumed that there is a proportional relationship, Ka, between the voltage induced in the armature and the angular velocity of the motor shaft, and an electromechanical relationship is assumed that the mechanical torque is proportional, Km, to the electric current.

$$E_a(t) = K_a w(t),$$
$$Tm(t) = K_m i(t),$$

3. Presentation of results

3.1 Vehicle dynamics

The track consists of 3636 m, in which there are different changes in height between slopes and valleys, making a general sum three large sections are obtained: on the one hand we have that the vehicle will face for 200 m a slope of 17º, which although it is not very steep, generates an additional effort in the vehicle to be a great distance, on the other hand we have that the vehicle will face for 1500 m a slope of 6º, and finally it is proposed that for 1300 m the vehicle will go downhill.

The result of the forces that disturb the vehicle when going uphill at 17º of inclination are as follows

Table 3 Result forces at 17

Constant	Symbol	Value	Units
Value for aerodynamic force	Fd	1.2680	N
Value for frictional resistance	Fr	0.0620	N
Value for slope resistance	Fw	66.8735	N
Maximum value for tensile strength	Ft	68.2036	N

The result of the forces that disturb the vehicle when going uphill at a 6º inclination are as follows.

Table 4 Result forces to 6th grade

Constant	Symbol	Value	Units
Value for aerodynamic force	Fd	1.2680	N
Value for frictional resistance	Fr	0.0209	N
Value for slope resistance	Fw	69.6502	N
Maximum value for tensile strength	Ft	70.9392	N

Only the maximum values have been taken into account, since these allow to obtain the maximum capacity that the engine will require to allow the vehicle to satisfactorily overcome the steepest slope, in conjunction with the most aggressive aerodynamic conditions and the maximum weight for which it is designed.

The vehicle takes 9.16 minutes to complete one lap, and it also requires a tractive force of 139.1428 N. Ft of 139.1428 N.

In practice, it is expected to disconnect the motor (prevent current flow) from the power source at intervals when the vehicle is moving by impulse (valleys on the track) as this optimizes power consumption as much as possible.

Once the Ackerman model has been used on the vehicle, the constants necessary for the vehicle's displacement to overcome the lateral forces that disturb it and also to perform optimally in any curve are presented.

Table 5 Ackerman constants

Constant	Symbol	Value	Units
Wheelbase	L	1.4335	m
Width between wheels	t	0.602	m

Turning radius from center of mass	R		m
External angle	δ	10.330	-
Internal angle	δ_i	11.330	-

The average speed that the vehicle would have is obtained thanks to the implemented PID control, which allows the speed to remain constant regardless of the changes of slope in the route, at a speed of 24.47 km/h (15.47 mph).

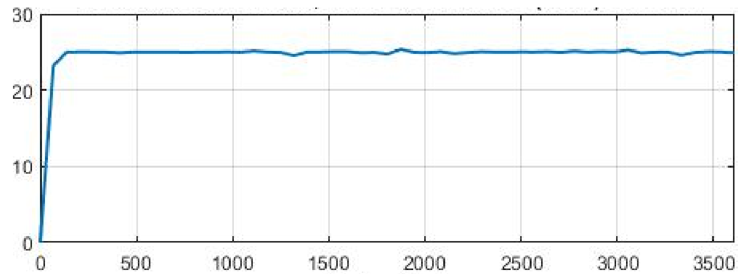

Fig 8 Average vehicle speed

3.2 Powertrain and electrical system design

The linear speed is determined from the following relationship taking into account the output speed (W axle), by the wheel diameter (Ø wheel), by Pi (π), over 60.

$$V = \frac{\pi}{60} * W_{eje} * Ø_{rueda}$$

where;

$$W_{eje} = \frac{V * 60}{\pi * Ø_{rueda}}$$

The vehicle will travel at a speed of 24km/h.

$V = 6{,}666m/s; Ø_{rueda} = 0{,}49m$

$W_{eje} = 259{,}84\ RPM; W_{motor} = 650\ RPM$

The gear ratio is determined from the following ratio, the output speed (W shaft), over the input speed (W motor).

$$R_{Transmision} = \frac{W_{motor}}{W_{eje}} = 2{,}5$$

A suitable number of teeth is selected that meets the following relationship:

$$\frac{Z_2}{Z_1} = 2{,}5$$

Z1 and Z2 are the respective tooth numbers of the ring gear and pinion. According to DIN 8196 the following numbers of teeth for the pinion are preferred: 17,19,21, among others. It is recommended that the number of sprocket teeth be odd, as this ensures that the sprocket teeth mesh with different chain links, thus distributing wear more evenly.

The K coefficients chosen for the system are presented in Table 6.

Table 6 Selected K coefficients

K Factor	Value
K1	*1,1*
K2	*1*
K3	*1*
K4	*1,4*
K5	*1*

Once all the coefficients have been calculated, the corrected power (Pc) can be obtained.

$$P_c \ = 1540\ w$$

β, the contact angle in radians:

$$\beta = \sin^{-1}\frac{R2 - R1}{O1O2}$$
$$O1O2 = 500mm$$
$$\beta = 0{,}07569°$$

The expression L/p (length/step of the chain) must be an integer, so the distance between O1O2 will have to be adjusted for this to be fulfilled. The following table shows the results of applying the above expression in a process that is iterative.

Table 7 Iteration of the number of links

O1O2 mm	B Radians	L/p (no. of links)
	0,07569	134,78
498	*0,7600*	134,36
496,25	*0,7627*	

The total stress supported by the chain was tested and yielded the following results.

Table 8 Transmission mechanical results

Constant	Symbol	Value	Units
Useful effort developed	*Fu*	570	N
Centrifugal force	*Fc*	1,23	N
Chain mass	*M*	0,40	Kg
Breaking load	*Kp*	910	Kp
Total effort	*Fo*	58,2554	Kp
Safety coefficient	*Cs*		-
Ultimate pressure	*Ps*	20,4	Mpa

It is considered good practice to have a safety coefficient of at least Cs>12, so the selected chain complies. The final result of the chain drive is shown in Table 9, 10 and 11, first the specifications for the chain are presented.

Table 9 Chain parameters

Constant	Symbol	Value	Units
Series	-	DIN 06B-1	-
Step	*H*	9.525	mm
Type	-	Single roller chain	-
Development or length (complete turn)	*Lg*	1276,38	mm
Number of links	*E*		-
Distance between wheel centers	*X*	496,25	mm

Next, the specifications for the minor wheel or pinion are presented, this sprocket is located just outside the motor shaft.

Table 10 Sprocket parameters

Constant	Symbol	Value	Units
Number of teeth	Z1		-
Primitive diameter	Dp1	51,84	mm

Finally, the final results are presented for the largest sprocket, which is the one that will be assembled on the wheel, since what the study is looking for is torque and not speed.

Table 11 Crown parameters

Constant	Symbol	Value	Units
Number of teeth	Z2	42	-
Primitive diameter	Dp2	127,43	mm

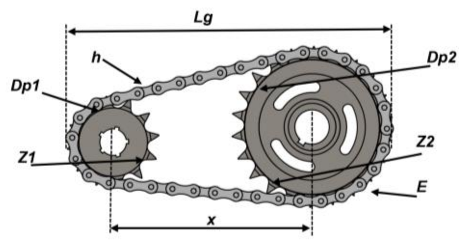

Fig 9 Vehicle transmission

The electrical system of any vehicle comprises all the elements that allow the generation, transport and distribution of electrical energy, which is then transformed into mechanical energy; unlike combustion vehicles, which transform chemical energy into mechanical energy [7].

The initial parameters of the motor are 48 V and 1000 W, however, the current is taken as controllable and time-varying, since it will not be the same if the vehicle is crossing a slope, or if it is on a straight road, among other scenarios that may arise [8]. The electrical system of the vehicle works with a 48 V lithium battery and a 12 V battery, which have the function of storing energy and also allow recharging when the vehicle requires it. The batteries are protected by a battery management system (BMS) in accordance with the 60 V SEM safety standards. The BMS is responsible for protecting the entire system against shorts or surges that may occur due to overheating or any other type of inconvenience. A 3A 30A fuse was placed at the positive output of the BMS according to the estimated maximum current, which should not exceed 30A, and at the negative output the main switch, which should be in the cabin next to the pilot, followed by an emergency switch, which should be on the outside of the vehicle, as established in the SEM competition standards.

A 12V voltage converter is adapted, which allows the connection of accessories such as horn and lights without overvoltage problems. Finally, the controller is connected, which is designed under the PID parameters (proportional, integral and derivative), the establishment time and the over peak of the system were established as requirements to be able to drive the engine correctly and with this the appropriate constants were found, from this the potentiometer (responsible for the acceleration of the vehicle) and a direct switch to the controller which is also established by the rules of the SEM competition, finally the engine is established and followed by this, the transmission which is directly connected to the wheel responsible for the traction of the vehicle.

The diagram shows the circuit assembled to operate the vehicle.

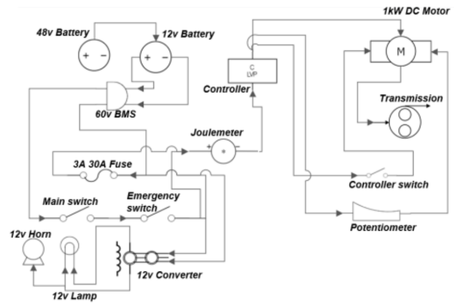

Fig 10 Electrical diagram of the vehicle

3.3 Mathematical model of the electric motor

The study proceeds to obtain the transfer function that allows relating the different outputs of the electromechanical dynamics (torque, counterelectromotive force, armature current, angular velocity and position) with the input (voltage) in the transformed domain of the place.

$$\frac{Tm(s)}{v(s)} = \frac{K_m(Js + B)}{LJs^2 + (RJ + LB)s + RB + K_m\,K_a},$$
$$\frac{Ea(s)}{v(s)} = \frac{K_m K_a}{LJs^2 + (RJ + LB)s + RB + K_m\,K_a},$$
$$\frac{i(s)}{v(s)} = \frac{(Js + B)}{LJs^2 + (RJ + LB)s + RB + K_m\,K_a},$$
$$\frac{w(s)}{v(s)} = \frac{K_m}{LJs^2 + (RJ + LB)s + RB + K_m\,K_a},$$
$$\frac{\theta(s)}{v(s)} = \frac{K_m}{s(LJs^2 + (RJ + LB)s + RB + K_m\,K_a)},$$

For the state space representation, these must be defined and substituted into the differential equations and are represented in a matrix form.
States:

$$x1 = w;\ x2 = i;\ \dot{x}1 = \dot{w};\ \dot{x}2 = \dot{i}$$

Substitution:

$$\dot{x}1 = -\frac{B}{J}x1 + \frac{Km}{J}x2,$$
$$\dot{x}2 = \frac{R}{L}x2 - \frac{Ka}{L}x1 + \frac{1}{L}v,$$

State space representation:

$$\begin{bmatrix}\dot{x}1\\ \dot{x}2\end{bmatrix} = \begin{bmatrix}-\frac{B}{J} & \frac{Km}{J}\\ -\frac{Ka}{L} & \frac{R}{L}\end{bmatrix}\begin{bmatrix}x1\\ x2\end{bmatrix} + \begin{bmatrix}0\\ (\frac{1}{L})\end{bmatrix}v$$

The output equation depends on what you want to see, in this case the two states which are angular velocity and current:

$$\begin{bmatrix}y1\\ y2\end{bmatrix} = \begin{bmatrix}1 & 0\\ 0 & 1\end{bmatrix}\begin{bmatrix}x1\\ x2\end{bmatrix}$$

Simulation in Matlab, Simulink for which the following constants are required:

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Description	Symbol	Value	Units
Rotor moment of inertia	J	1,334	$kg.m^2$
Viscous friction constant of the engine	B	0,100	$N.m$
Electromotive force constant	$Km = Ka$	0,330	$N.m/A$
Motor electrical resistance	R	1,990	Ω
Electrical inductance of the motor	L	0,8779	H

Fig 2 Electric motor constants

The simulation of the current after 10 seconds is as follows:

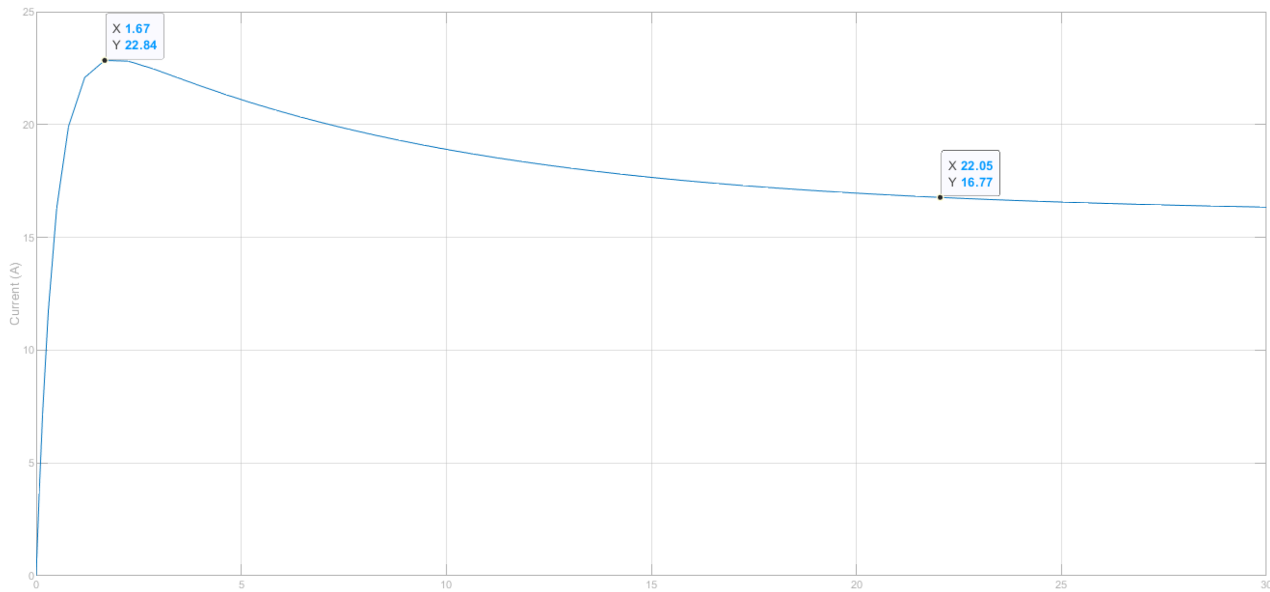

Fig 11 Simulation of the current

As can be seen, the maximum peak is 22.8 Amperes and then stabilizes at 16.7 Amperes, which means that the current consumption is optimized in a very satisfactory way.

The simulation of the angular velocity after 100 seconds is as follows:

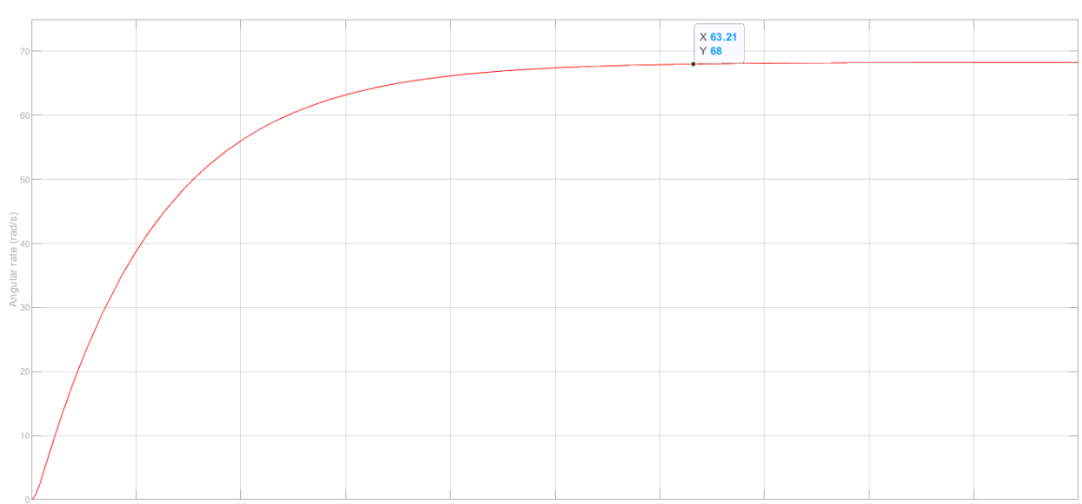

Fig 12 Simulation of angular velocity

As can be seen, the system takes 63 seconds to reach the 650 RPM required in the engine to be able to generate the linear speed at the transmission output, i.e., 24 km/h (15 mph).

3.4 Motor parameters and energy consumption

The energy consumption of the vehicle is the most important aspect, since all the previous work is concluded in that the vehicle is as efficient as possible, taking this into account, the predetermined values that the electric motor has and the speed to be used during the entire route are highlighted.

Table 12 Vehicle parameters

Description	Value	Units
Average speed	24.47	km/h
Engine power	1.00	kW
Minimum current	4.50	A
Maximum current	26.20	A

The vehicle is simulated in three (3) general moments that cover the entire route; however, the most relevant data are the power at the point of maximum effort, the working current with load, the energy consumption of the vehicle for the entire route, and the value of the autonomy it can deliver.

Table 13 Vehicle mechanical results

Description	Value	Units
Maximum power required in one (1) turn	0.9183	kW
Rated current in one (1) turn	22.84	A
Energy consumption in one (1) lap	0.0333	kWh
Electric motor range	89.9627	km/kWh

4. Conclusions

In a brief comparison, it is assumed that a gasoline engine with the same mechanical conditions would have an energy consumption of 2.37 times the consumption of an electric motor, i.e., a consumption of 0.0789 kWh, with a range of 38.0228 km/kWh, well below the range obtained with the chosen engine.

The implementation of computer aided engineering in this project was of utmost importance, because thanks to SolidWorks, Inventor, Matlab, among some other engineering software, it was possible to obtain the final results.

When talking about avoiding losses or maximizing the efficiency of the system, the belt/pulley is more likely to lose power than the chain system, either by slippage or because it must be in constant tension to take advantage of its usefulness, this simple reason leads us to choose the chain/pinion transmission.

The contribution of this project has its focus on clean mobility, in which natural resources are saved on the one hand and economic resources on the other, in addition, although it has purely theoretical data, it allowed to understand how the design and process of a competition vehicle works.

The world is in constant transformation and humanity must be able to adapt to any situation, the purpose of the team was to demonstrate that there are different ways to move in order to obtain the most efficient results, it is expected to motivate many young people to continue with the search and perhaps innovate the way to move between nearby places.

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