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Article

Design of a three-speed automatic transmission system with reverse for all-terrain vehicles (ATV)

Diseño de un sistema de transmisión automática de tres velocidades con reversa para vehículos todo terreno (ATV)

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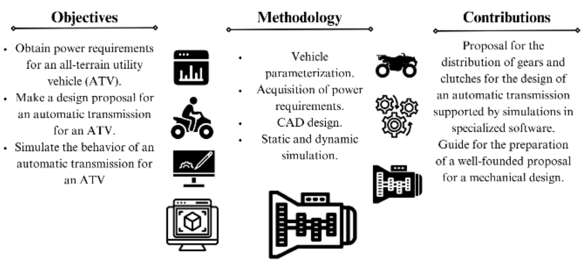
Abstract

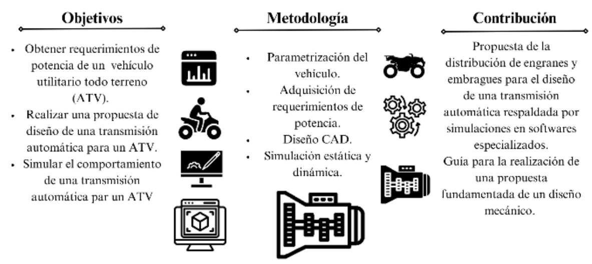
This work proposes the design and simulation of a conventional three-speed automatic transmission, including reverse gear, for an all-terrain utility vehicle (ATV). The proposed design was generated considering the power requirements needed for the selected vehicle to operate on a specific driving cycle over uneven terrain with variable slopes ranging from -0.5° to 2.5°. A vehicle with well-defined mass, dimensions, and engine displacement was selected as the design criteria. Using the numerical software MATLAB, the power requirements were calculated, establishing that the transmission ratios range from 0.18 to 1. Using commercial Multibody Dynamics (MBD) software, ADAMS, the reliability of the proper functioning of the gear trains was determined based on boundary conditions. Similarly, to verify that the selected materials and components can withstand critical loads, structural static simulations were carried out using the finite element method with the commercial software ANSYS.



Finite element analysis, Simulation CAE, Automatic Transmission

Resumen

Este trabajo propone el diseño y la simulación de una transmisión automática convencional de tres velocidades, incluyendo reversa, para un vehículo utilitario todoterreno (ATV). El diseño propuesto se generó tomando en cuenta los requerimientos de potencia necesarios para que el vehículo seleccionado sea capaz de circular, siguiendo un ciclo de manejo específico, sobre un terreno irregular con pendientes variables oscilantes entre -0.5° y 2.5°. Utilizando el software numérico MATLAB©, se calcularon los requerimientos de potencia, que van desde 0.18 hasta 1. Empleando un software de Multibody Dynamics (MBD) comercial, ADAMS©, se corroboró el adecuado funcionamiento de los trenes de engranajes con base en las condiciones de frontera, de forma similar, para corroborar que los materiales y componentes seleccionados soporten las cargas críticas, se realizaron simulaciones estáticas estructurales mediante el software comercial ANSYS©, basado en la teoría de los elementos finitos.



Análisis por elemento finito, Simulación CAE, Transmisión automática

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Introduction

The design of vehicle transmissions is a fundamental process in automotive development and innovation, as it allows vehicles to operate effectively in various terrains and under different driving conditions. To achieve this, it is essential to carry out a thorough analysis of the power and torque requirements necessary for the vehicle to perform its functions properly. Several research studies on this subject have been documented in the literature. For example, CENIDET (Centro Nacional de Investigación y Desarrollo) mathematically modelled the mechanical part of an electric vehicle, determining the differential equation that describes the forces exerted by the road and the equations that govern its transmission (M. Durán, 2009). For its part, the University of Chile developed a power system for a single-seater vehicle, using simulations of the SolidWorks© 2015 Motion Analysis module to determine the power requirements based on the Theo Jansen mechanism (Andrade, M. A. L., 2016).

The power requirements are directly related to the exhaust emissions as a function of the driving dynamics, which in turn reflects the engine load. Thus, the driving dynamics calculation must consider the five main resistances that affect engine load: wheel force (Fw), drag resistance (Rd), rolling resistance (Rr), gradient resistance (Rg) and acceleration resistance (Ra) (Song, J., & Cha, J., 2021).

Once the power requirements are established, the design criteria for the drive system can be defined. For example, Zarate, W. E. H. (2020) evaluated the design criteria for an electronic continuously variable transmission (E-CVT), considering as initial criteria the maximum torques and the rated power of the three motors that make up the powertrain, and then defining the characteristics of the planetary gears, including dimensions, module, and number of teeth. Similarly, Pórtilla, L. V., et al. (2013) emphasised the importance of the type of material from which the components of the transmission system are manufactured, highlighting the relevance of strength and reliability analysis, as well as the appropriate selection of materials based on the normal tensile stresses and shear stresses to which the system is subjected.

The development of transmissions for vehicles is not only limited to internal combustion vehicles, as can be seen in the work carried out at the Escuela Tecnológica Instituto Técnico Central in Bogotá, where a transmission was adapted to improve the efficiency and performance of a human traction vehicle (VTH), Castro Escobar, W. A., & Ramírez Moreno, L. E. (2024). The University of the Armed Forces also developed a transmission that allows coupling an internal combustion engine with an electric motor in very extreme conditions, with slopes of 15° and an engine of 486 kg, which makes the development and selection of materials and components a complex and interesting work, the difference to what is proposed in this article is that this work opts for the analysis and selection of a manual transmission. Gavilanes, A., Arturo, C., et al. (2024).

Taking into consideration the above, this article focuses on the design of an automatic transmission for an all-terrain vehicle (ATV), starting from the power curve generated by the engine and the power requirements of the road. The main objective is to design the gear and clutch layout of a conventional three-speed automatic transmission with reverse, so that it meets the specific power requirements of the vehicle. In addition, the correct operation of the proposed design will be validated through a series of numerical experiments consisting of static structural and dynamic multi-physical simulations of the gears that make up the transmission ratios of the planetary systems using software based on finite element theory

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Development of headings and subheadings of the article with subsequent numbers

Design criteria

We start with the parameterisation of the vehicle to be analysed, taking as a basis the chassis of an Italika© 150 ATV. Figure 1 shows the dimensions of the vehicle and the wheelbase.

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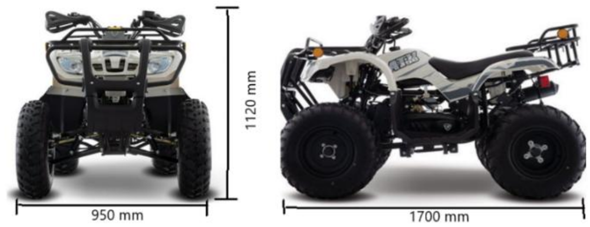
Box 1



a) b)

Figure 1
a) Height and width of the vehicle b) Wheelbase of the vehicle

Source: Own elaboration

Subsequently, Table 1 details the system design criteria that underpin the transmission box proposal developed in this work.

Box 2

Table 1
Numerical vehicle design criteria.

Experiment conditions	Value
Road slope	Variable according to graph
Wind speed	1 km/h
Rolling resistance coefficient	0.092
Drag coefficient	0.62
Air density	1.18 kg/m^3
Dynamic radius	0.2667 m
Vehicle parameters	Valor
Rim diameter	0.254 m
Tyre size	21\7 - 10
ATV total mass (with rider and transmission)	245 kg
Total frontal area of ATV (with rider)	1.031 m^2
Centre of mass in X	1.34 m
Centre of mass in Y	1.21 m
Centre of mass in Z	0.60 m
Engine size	177.3 cc
Gear ratios	Value
M_{v1}	0.266
M_{v2}	0.53
M_{v3}	1
Mv reversa	0.2

Source: Own elaboration

Once the chassis model has been selected, a rider with an average height of 1900 mm is chosen. Using SolidWorks© software, a virtual model of the ATV including the rider is developed, as shown in Figure 2. In this process, the total frontal area of the vehicle is calculated together with the rider.

Box 3

Figure 2
CAD model of the rider with the ATV

Source: Own elaboration

In addition to the criteria presented in Table 1, specific objectives for the possible implementation of the design are set out below:

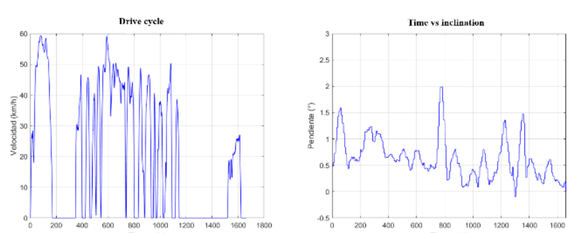
- **Light weight:** The design should be as light as possible to optimise vehicle performance.

- **Number of drivetrains:** The vehicle should have three forward gears and one reverse gear.

- **Drive cycle:** The vehicle shall be able to move according to the driving cycle on slopes, as illustrated in Figure 3.

- **Engine power and torque curve:** The engine selected to power the vehicle develops the power and torque curve shown in Figure 4.

Box 4



a) b)

Figure 3
a) Management cycle and b) Terrain slope.

Source: Own elaboration

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Box 5

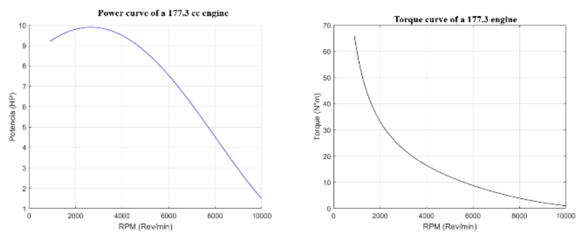

Figure 4

a) Management cycle and b) Slope of the terrain

Source: Own elaboration

By considering weight criteria, transmission functionality, hill performance and engine characteristics, it is sought to ensure that the vehicle not only meets the technical requirements, but also adapts to the actual operating conditions. This holistic approach ensures that the final design is efficient, effective and suitable for implementation in the intended context of use.

Power requirements

To meet the above-mentioned criteria and the specific parameters of the ATC, it is essential to consider the power requirements of the ATC. These requirements are directly related to the driving resistances faced by vehicles in general. As illustrated in Figure 5, there are several forces that affect vehicle dynamics, which must be overcome and taken into account when designing any type of automotive transmission.

Box 6

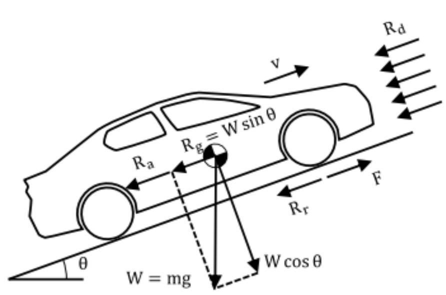

Figure 5

Forces affecting vehicle dynamics

Source: (Song, J., & Cha, J., 2021)

The total resistance to conduction can be calculated using the following expression.

$$F_w = R_r + R_d + R_g + R_a \quad (1)$$

where, R_r represents the rolling resistance force, R_d represents the drag resistance force of the air, R_g represents the resistance of the road slope, and R_a represents the acceleration resistance (Song, J., & Cha, J., 2021).

The air drag force models the opposing force that a wind volume presents when it is displaced in order for the vehicle to take its place. It is calculated by:

$$R_d = 0.5\, Cd\, A\rho v^2 \quad (2)$$

In equation (2), the coefficient Cd is the dimensional drag coefficient, while A contains the value of the frontal area of the vehicle in metres, ρ represents the value of the air density in kg/m^3, and v represents the ratio of vehicle speed to wind speed at m/s.

Rolling resistance force, R_r, contains the magnitude of the opposing force that the road exerts on the wheel, thus preventing its movement, and is determined by the following expression:

$$R_r = \left(Csr + Cdr\left(\frac{v}{100}\right)^{2.5}\right) * Wcos(\theta) \quad (3)$$

Where;

$$Csr = -514.7\left(\frac{Ptire}{100}\right)^3 + 53.72\left(\frac{Ptire}{100}\right)^2 - 1.877\left(\frac{Ptire}{100}\right) + 0.03051 \quad (4)$$

$$Cdr = -793.1\left(\frac{Ptire}{100}\right)^3 + 83.98\left(\frac{Ptire}{100}\right)^2 - 2.977\left(\frac{Ptire}{100}\right) + 0.0375 \quad (5)$$

while$v$, represents the speed of the vehicle in m/s, W is the weight of the vehicle in N, Θ son los grados de inclinación del camino y *Ptire* representa a la presión de la rueda en unidades bar.

On the other hand, the slope resistance, R_g, indicates the magnitude of the force required to keep the vehicle in a steady state when moving on a road with a gradient θ. The slope resistance is determined using:

$$R_g = Wsin(\theta) \quad (6)$$

Where, W, represents the proportional weight of the vehicle.

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In addition to the acceleration resistance, R_a, determines the amount of force required to increase the vehicle speed from an initial speed to a final speed in a given time.

$$R_a = ma \quad (7)$$

where, m represents the mass of the vehicle and a represents the acceleration of the vehicle. (m/s^2).

Based on the elements collected so far, MALAB© is used to analyse the behaviour of the engine power and torque distribution as a function of time during the driving cycle. This analysis allows to start the selection of commercial components for the final design of the transmission system.

Transmission design

For the design of this transmission, planetary gear systems have been used, which are composed of a sun gear (central gear), planetary gears (intermediate gears) and a ring gear (external gear with internal teeth), as shown in Figure 6.

Box 7

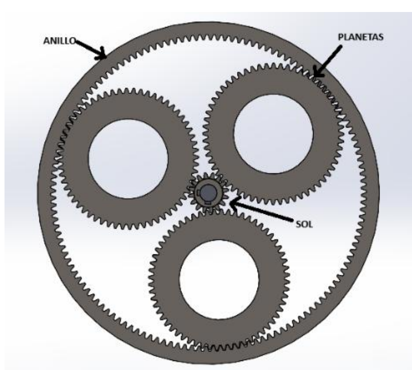

Figure 6
Planetary gear system
Source: Own elaboration

To establish the gear ratios presented in Table 1, it is necessary to select the module, pressure angle and number of teeth of the gears as shown in Table 2. In this case, two planetary systems with a single ring gear arrangement have been used, which are connected by an arm and a central shaft, as shown in Figure 7.

Box 8

Table 2
Gear selection

Gear design			
Position	No teeth	Module	Ang. Pressure °
Solar 1	16	2	20
Planets 1	60	2	20
Ring	136	2	20
Solar 2	16	2	20
Planets 2	68	2	20

Source: Own elaboration

Box 9

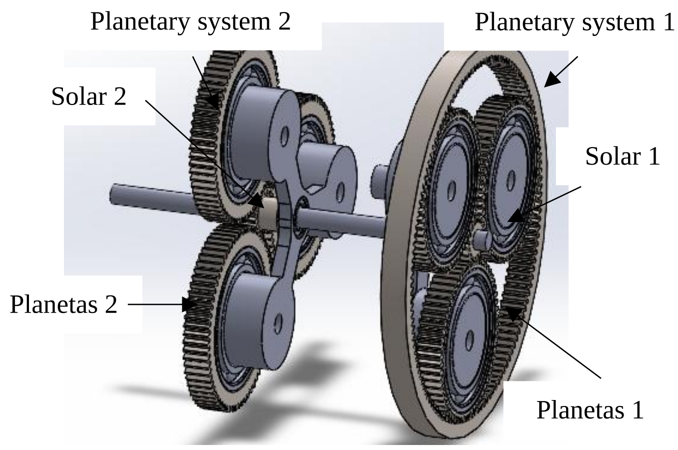

Figure 7
Planetary gear system
Source: Own elaboration

With this arrangement, it is intended that, with the help of clutches, the drive trains can be configured to transmit the engine power to the wheels. In first gear, the ring gear is required to remain completely static by actuating the clutches, thus allowing the sun gear of the first planetary system to rotate with the angular velocity of the engine. This motion is transmitted to the planets, of which the connected output shaft generates the first gear ratio.

For the second gear, the solar gear of the second system rotates with the power input of the engine, transmitting the power to the planets of that system. By disengaging the clutch system that holds the ring stationary, the planets can rotate it, thus transmitting motion to the planets of the first system, and again, the shaft connected to these planets allows the output of the second ratio.

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For the third gear, the rotational motion of both the planets and the ring is prevented in both systems, so that the whole assembly rotates at the same speed as the solar gear, which coincides with the input speed of the motor, thus achieving a 1:1 ratio.

For reverse gearing, a third planetary system is commonly used. However, due to sizing limitations of the test vehicle, a configuration is proposed that allows the reverse motion to be performed using only two planetary systems. This configuration requires decoupling the output axis of the planets and coupling it to the ring, which will generate the motion of the output axis in the opposite direction. It is necessary to decouple the planetary systems, which are connected by means of two crown gears that are kept in contact through clutches as required.

Finally, rotational movement around the solar gear of the planets is prevented, allowing the planets to move on their own axis. In this way, the ring gear is made to rotate in the opposite direction, transmitting the speed ratio of the reverse gear. The final distribution can be seen in Figure 8.

Box 10

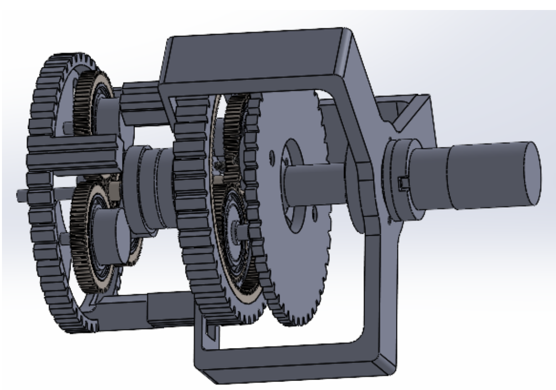
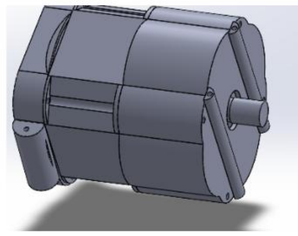
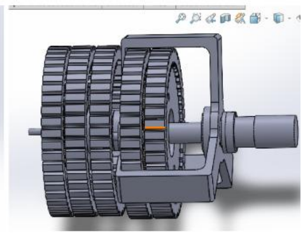


Figure 8
Main transmission distribution
Source: Own elaboration

As a final part of the transmission design, a proposal is made for the housing that will contain the whole mechanism as shown in figure 9 together with the clutch layout.

Box 11





a) b)

Figure 9
a) Proposed housing b) Clutch distribution
Source: Own elaboration

Structural static analysis of critical components

With the design established, it is essential to verify that the selected components can withstand the critical loads that the transmission will demand. For this purpose, a structural static analysis based on the finite element theory is carried out using the commercial software ANSYS Workbench©. The maximum torque expected in the case is taken as a reference, and the analysis will be carried out on the critical elements of the assembly. In particular, a torque of 330 Nm is used for the analysis on the central shaft, as well as on the sun gears and the 60-tooth gear, which is the component with the smallest amount of material.

In addition, a structural analysis of the *housing* supports will be carried out, considering a total mass of 50 kg. In this case, the housing supports must be able to withstand at least 490.5 N, which means that they must resist 2,662 kPa. Next, figure 10 shows the analysis of the central axis where the maximum torque is applied to the solar gears, figure 11 shows the same torque, but in the 60-tooth gear, which is the one that contains the least material, and finally, figure 12 shows the analysis of the *housing*.

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Box 12

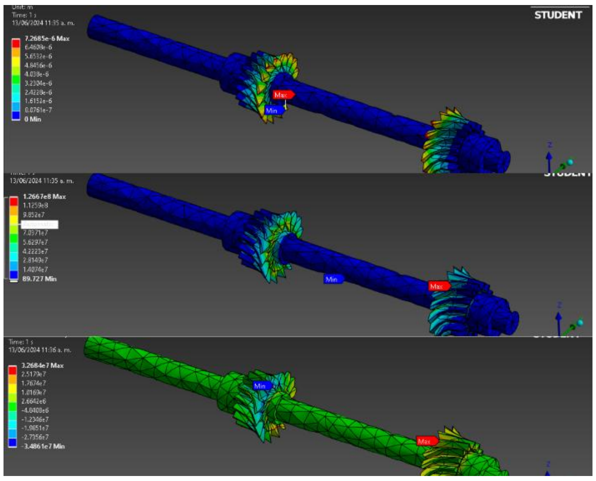

Figure 10

ANSYS© analysis of deformation, Von-Mises stress and shear stress on axis and suns

Source: Own elaboration

Box 13

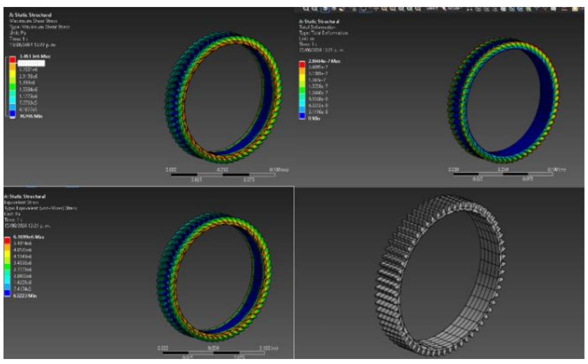

Figure 11

ANSYS© analysis of deformation, Von-Mises stress and shear forces on the 60-tooth planet

Source: Own elaboration

Box 14

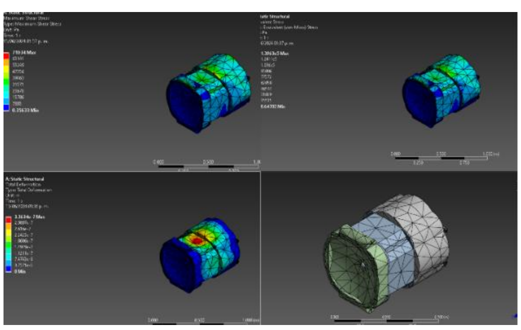

Figure 12

ANSYS© analysis of deformation, Von-Mises stress and shear forces on the *housing*

Source: Own elaboration

Dynamic gear simulation

With the structure analysis developed in ANSYS© it is possible to determine whether or not the material will support the loads to which the mechanism is subjected. However, these studies do not ensure the correct movement of the proposed design, so it is necessary to carry out a dynamic study of the mechanism to verify the proper functioning of the transmission system. For this purpose, the ADAMS© commercial software has been used, which allows the power curve of the motor to be introduced in order to verify the transformation of the power at the output of the system. In addition, the software allows the presence of possible interferences in the components, derived from a deficient design, to be determined. Within the software, only the distribution of the planetary systems has been used for simulation, as can be seen in Figure 13.

Box 15

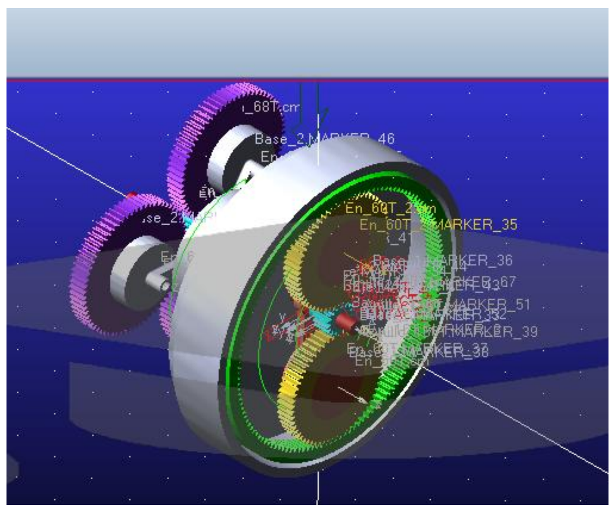

Figure 13

Dynamic simulation in ADAMS©

Source: Own elaboration

Results

Based on the elements collected so far, MATLAB© is used to analyse the behaviour of the engine power and torque distribution as a function of time during the driving cycle. Figure 14 can be consulted for a detailed visualisation, where the maximum and minimum torque and power that the vehicle delivers throughout the driving cycle to pass over the established terrain can be observed.

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Box 16

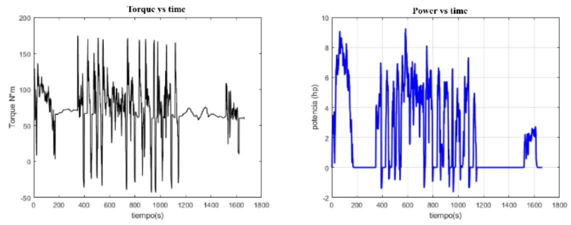


Figure 14

a) Torque vs. time and b) Power vs. time.

Source: Own elaboration

Figure 15 shows how the slope changes over the course of the driving cycle and the power required by the vehicle to pass through the terrain described by the slopes.

Box 17

a) b)

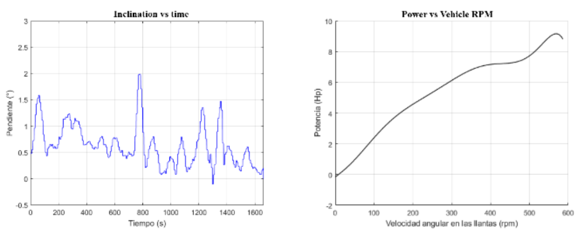


Figure 15

a) a) Slope vs. time and b) Power requirement.

Source: Own elaboration

On carrying out the test using simulink and obtaining the graph of torque vs. time, power vs. time and finally the graph of speed vs. time. It is observed that there is similarity in the graphs obtained. When testing with Simulink and obtaining the graphs of torque vs. time, power vs. time, and speed vs. time, there is a notable similarity in the graphs obtained, as shown in Figure 16.

Box 18

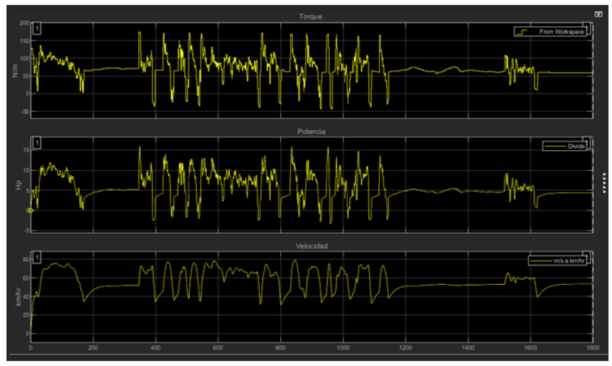


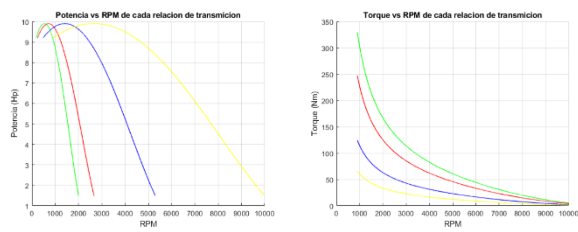
Figure 16

Testing graphs with Simulink.

Source: Own elaboration

Figure 17a shows the transmission response with the gear ratios, in red colour the Mv1, in blue colour the Mv2, in yellow colour the Mv3 and in green colour the Mvr, in the same order for the torque response, it can be seen in Figure 17b.

Box 19



a) b)

Figure 17

a) Power vs transmission rpm and b) Torque vs transmission rpm.

Source: Own elaboration

Figure 18 shows the comparison between the power output and the power required to travel the set route, which includes both the road gradients and the speed variations captured in the driving cycle. The black graph represents the power required to complete this route. As can be seen, the graphs of the power supplied by the transmission confirm that the proposed transmission system adequately transforms the power coming from the engine in such a way that it allows the vehicle to drive along the selected road following the driving cycle described in Figure 3a.

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Box 20

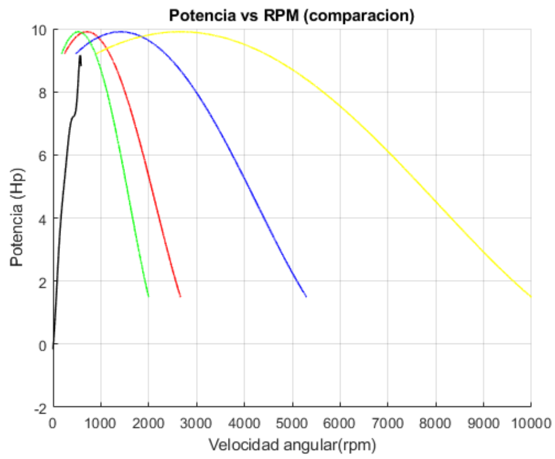


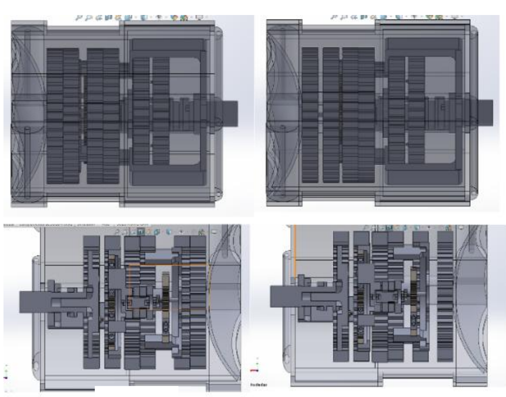
Figure 18

Comparison between the power required by the road and the power delivered by the transmission.

Source: Own elaboration

With these results the design can be concluded, leaving the configurations described above as follows and can be seen in figures 19 and 20, where the different configurations of the clutches can be seen, in figure 19a it can be seen how the crowns are fixed together with the ring to a clutch fixed to the housing, in 19b the clutch is released from the housing, allowing the crowns and ring to rotate; in figure 20b we can see how the arms are locked together, joining the movement of the suns to the planets and the ring and finally in figure 20b we can see how the crowns are separated and the movement of the planetary system 2 is locked and how the output shaft is moved to connect with the ring and the arms are locked together so that the planets only rotate on their own axis.

Box 21



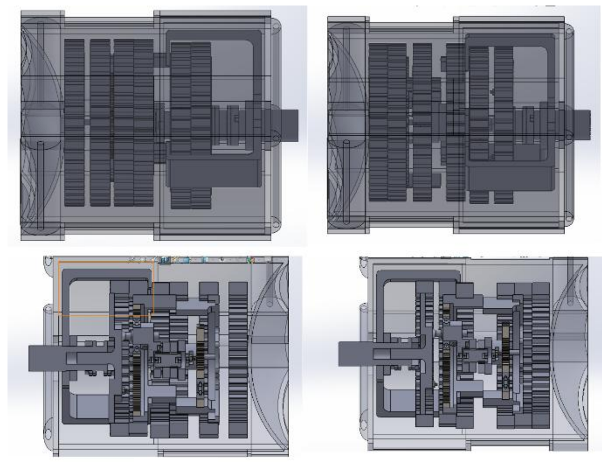
a) b)

Figure 19

a) march 1 and b) march 2.

Source: Own elaboration

Box 22



a) b)

Figure 20

b) March 3 and b) Reverse.

Source: Own elaboration

As mentioned above, it is necessary to verify that the elements support the expected loads. Structural steel has been selected as the main material for the internal components, its main characteristics can be seen in table 3, while for the housing aluminium Series 5000 has been selected, its properties are presented in table 4. The mechanical properties of steel and aluminium are presented in figures 11 and 12, respectively. It is important to mention that these data are standardised and are provided by ANSYS Workbench© as part of its material libraries.

Box 23

Table 3

Mechanical properties of structural steel obtained from ANSYS *Workbench* ©

Steel properties	
Mechanical property	**Value**
Young's modulus	2e11 Pa
Bulk modulus	1.667e11 Pa
Shear modulus	7.6923e10 Pa
Creep modulus	2.5e8 Pa

Source: Own elaboration

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Box 24

Table 4

Mechanical properties of aluminium 5000 obtained from *ANSYS Workbench©*

Properties of aluminium	
Mechanical property	**Value**
Young's modulus	7.1e10 Pa
Bulk modulus	6.9608e10 Pa
Shear modulus	2.6692e10 Pa
Creep modulus	2.8e8 Pa

Source: Own elaboration

The results obtained in the structural static analysis carried out in ANSYS Workbench© are summarised in Table 5.

Box 25

Table 5

Results of ANSYS©

Análisis estructural				
Element	**Study**	**Minimo**	**Maximum**	**Average**
Shaft with suns	Deformation	0 m	7.2685e-6 m	7.9278 e-7 m
Shaft with suns	Von-Mises Stress	89.727 Pa	1.2667e8 Pa	1.3434e7 Pa
Shaft with suns	Shear Stress	3.4861e7 Pa	3.2684e7 Pa	25287 Pa
Planet 60 teeth	Deformation	0 m	2.8e-7 m	4.6558e-8 m
Planet 60 teeth	Von-Mises Stress	63223 Pa	6.1699e6 Pa	2.0598 e6Pa
Planet 60 teeth	Shear Stress	36206 Pa	3.4613e6 Pa	1.147e6 Pa
Housing	Deformation	0 m	3.3634e-7 m	2.3255e-8 m
Housing	Von-Mises Stress	0.64702 Pa	1.3963e5 Pa	13569 Pa
Housing	Shear Stress	0.35633 Pa	71034 Pa	7543.6 Pa

Source: Own elaboration

As can be seen in Table 5, the deformations are in the order of10^{-7} Pa and the greatest effort in the components is $1.3434 \cdot 10^{7}$ Pa. For the steel used, its yield stress is of the order of$2 \cdot 10^{11}$ Pa, so that the maximum stress in the transmission system is the $1.3434 \cdot 10^{7}$ de the creep resistance of the material. Therefore, it can be said that the transmission system supports the workloads adequately.

To finalise the validation of the proper functioning of the proposed transmission, a dynamic simulation is carried out with the help of the ADAMS© commercial software, as shown in the following figures.

Figure 21 shows the representation of the reverse, where the blue colour shows the input speed of the motor and the red one is the output which, with the change of sign, shows that it is turning towards the opposite side, achieving the reverse. Figure 22 shows the transition of the input and output between first and second speed with an increase in speed as time increases.

Box 26

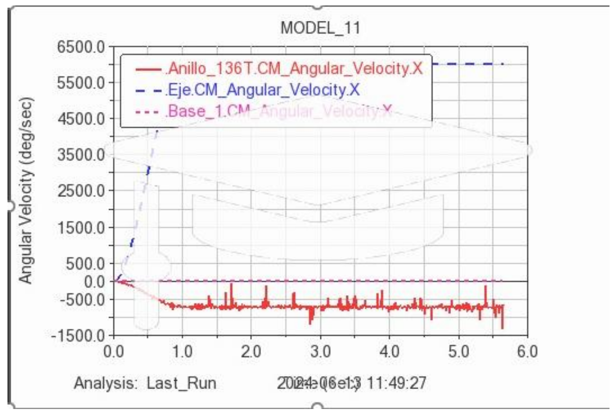

Figure 21

Angular exit speeds from first to third gear.

Source: Own elaboration

Box 27

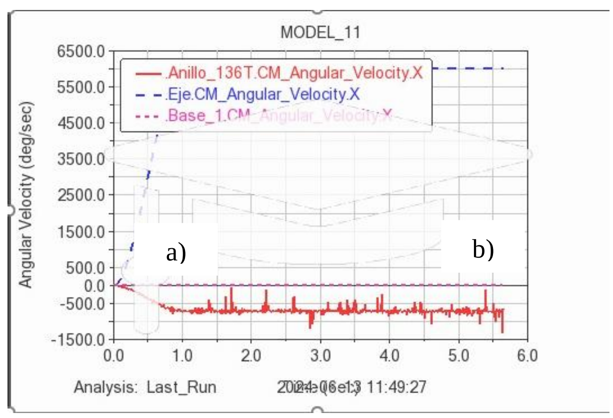

Figure 22

Reverse output angular speeds.

Source: Own elaboration

To appreciate the movement of the transmission, a link to videos of the transmission configurations and the dynamic behaviour of the transmission is included, https://drive.google.com/drive/folders/1aRXZoZLg_rRX8rjhOMRphyDVrRsDpLXf?usp=drive_link.

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Conclusions

At the end of the previous work, the proposal for the distribution of gears and clutches for the design of a conventional automatic transmission for an ATV vehicle has been completed, where it can be observed that in order to carry out different transmission ratios it is not necessary to use an excessive amount of gears, since with the help of a single planetary system the Mv1, Mv3 and reverse ratios are carried out, which allows saving space and components at the time of making the box, which is essential due to the limited space for the transmission that these vehicles have.

Furthermore, this work can serve as a small guide for the validation of any mechanism, showing the key points necessary for its validation and reliability, which would be: obtaining design criteria, numerical analysis, design sketch, validation of the resistance of the materials, numerical validation and dynamic simulation for the validation of the movement. It also serves as a background for future research such as the control system for the movement of the clutches or for a possible implementation of a prototype with this design proposal.

Declarations

Conflict of interest

The authors declare no conflict of interest. They have no known competing financial interests or personal relationships that could have appeared to influence the article reported in this article.

Author contribution

Ramírez-Ceja, Axel Ivan: Contributed to obtaining power requirements, vehicle parameterisation, CAD proposal and realisation, static structure simulation, dynamic CAE simulation and development of the written article.

Manríquez-Padilla, Carlos Gustavo: Contributed to the assessment and development of a Matlab code to determine the power requirements that allowed the development of the vehicle.

Pérez-Cruz, Angel: Contributed to the advice on the proposal of the vehicle dynamics to obtain the forces affecting the vehicle.

Torrez-Rico, Luis Armando: Contributed to the advice on the use of ADAMS© software.

Availability of data and materials

Contact the author of reference for any information or clarification related to the materials or data seen in the article.

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Abbreviations

List abbreviations in alphabetical order.

ATV	All-Terrain Vehicle
CAD	Computer-Aided Design
CAE	Computer-Aided Engineering
CENIDET	National Research and Development Centre
E-CVT	Electronic Continuously Variable Transmission
MBD	Multi Body Dynamics

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