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DESIGN OF AN ENERGY-SAVING ENGINE FOR A VEHICLE POWERED BY PNEUMATIC PISTONS

This article presents a comprehensive study on the mechanical design and optimization of a vehicle's power unit, focusing on enhancing performance and maneuverability for competitive scenarios. Preliminary tests have demonstrated the reliability of the power unit under various conditions; however, specific challenges, such as rear wheel lock-up during backward maneuvers, necessitate further investigation. The research aims to develop a clutch or reverse drive mechanism to improve maneuverability and assess the current transmission system for optimal torque delivery while maintaining an effective speed range. Additionally, the study examines the integration of the power unit with the kinematic structure, ensuring a cohesive design that maximizes operational efficiency. Long-term reliability under competitive stress and user feedback from operators will also be considered to refine the vehicle's functionality. By addressing these objectives, the research seeks to contribute innovative solutions that enhance the vehicle's overall performance and competitiveness in future events.

Keywords: *Pneumatic vehicle, pneumatics piston, compressed air, rack and pinion gear*
Fig.: 4. References: 10.

Relevance of the research. The research focuses on the mechanical design and performance of a vehicle's power unit, emphasizing the importance of structural integrity and operational efficiency. On completion of preliminary tests without issues indicates a robust mechanical configuration, highlighting the reliability of the power unit under various conditions.

Additionally, optimizing the transmission range to maximize torque while maintaining an effective speed range is crucial for improving acceleration. This optimization not only enhances performance but also contributes to the overall efficiency and effectiveness of the vehicle in competitive environments. Thus, the research is relevant for advancing vehicle technology and ensuring optimal performance in future competitions.

Problem statement. The current design of the vehicle's power unit has been successfully tested and verified under preliminary conditions; however, specific challenges remain unaddressed that could hinder performance in competitive scenarios. The primary issue is related to the kinematic structure, which results in the rear wheels locking when attempting to push the vehicle backward. This limitation restricts maneuverability and operational flexibility, making it difficult for the vehicle to perform effectively in various situations.

Additionally, there is a need of investigating the optimal range of the transmission system. The existing gearing configuration may not provide the maximum possible torque at the wheels, which is essential for efficient acceleration. Without proper optimization, the vehicle may fail to reach its full potential in terms of speed and responsiveness.

Analysis of recent research and publications. Recent studies in vehicle dynamics and powertrain optimization have highlighted the importance of efficient torque transfer and gear selection for enhancing performance in competitive environments. Furthermore, publications on gear ratio optimization indicate that fine-tuning the transmission system can significantly impact acceleration and overall vehicle efficiency. Studies suggest that a well-designed transmission can maximize torque output while maintaining a manageable speed range, ultimately leading to improved performance metrics. Overall, the analysis of recent literature underscores the need for continuous innovation in vehicle design, particularly in addressing the challenges of power delivery and drivetrain efficiency, to ensure optimal performance in competitive settings [1; 2].

Uninvestigated parts of a common problem. Despite significant advancements in vehicle design and performance optimization, several aspects remain underexplored, particularly regarding the integration of new technologies into existing systems. One such area is the interaction between the power unit and the kinematic structure, specifically how various configurations can influence the vehicle's maneuverability and performance during backward movement. While some studies have addressed individual components, comprehensive research focusing on the entire system's dynamics is limited.

Research objective. The primary objective of this research is to enhance the performance and maneuverability of the vehicle's power unit by addressing specific challenges related to its kinematic structure and transmission system.

The statement of basic materials. Pneumobil is an international competition of single-seat, compressed-air-powered vehicles designed for university students. It is organized by Aventics Hungary Kft., a company involved in the production and sale of compressed air systems. The competition has been held annually since 2008 in the Hungarian city of Eger. In 2008, 16 Hungarian teams participated with relatively primitive vehicles, with the highest speed achieved being 25 km/h and a range of 7 km. By the 11th edition, the maximum speed had exceeded 50 km/h, and the range reached 13 km. For the 12th edition of the competition, apart from our team, 48 other teams from 8 European countries registered [3].

Technical limitations of the propulsion unit and competition disciplines. The motor must be powered by compressed air. The direct use of any other form of energy to drive the vehicle is prohibited; however, it is allowed to use a heat exchanger to warm the compressed air. Recirculation systems that return air back into the system are also permitted, but they must be powered by compressed air.

The conversion of compressed air energy into mechanical work must be done using pneumatic cylinders and valves provided by Aventics. The use of any pneumatic components from other manufacturers is prohibited.

A maximum of four pneumatic cylinders can be used, selected from a catalog specifically assembled for the competition. Only linear double-acting actuators with piston diameters ranging from 32mm to 100mm and strokes from 80mm to 500mm are available.

The maximum allowed working air pressure is 0.1 MPa.

The vehicle may be equipped with only one motor, defined as a mechanism with a single output shaft. The deactivation of any cylinder can only be performed pneumatically or electro-pneumatically.

The motor can extend up to 200mm in front of the silhouette of the front wheels or 200mm behind the silhouette of the rear wheels, but these parts must be protected by a frame welded to the vehicle frame and covered by a protective shield. The highest mechanical part of the motor must not exceed 60% of the vehicle's width in height. The motor must be visible to the jury [6-10].

The competition consists of the following disciplines:

Long-distance race. The goal of this discipline is to cover the longest distance on a single container of compressed air provided by the organizer. The competition takes place on a circuit approximately 580 meters long. During this discipline, drivers must be changed at least three times. The average speed of the vehicle across all completed laps must be at least 15 km/h, and the time required to change drivers is included in the average speed calculation. The time and distance of the last, incomplete lap are not included in the average speed calculation, but the distance of the incomplete lap is included in the total distance covered.

Arcade race. In this discipline, the dynamic driving characteristics of the vehicle and the driver's skills are key. The race takes place on three different tracks. In the first slalom, all teams participate on a 700m long track, and the time to complete two laps is measured. The 18 fastest teams advance to the second round on a different 900m long track. The six fastest teams from the second round move on to the final, third round, where they race on a 1100m long track. In this round, the final ranking of winners is determined.

Acceleration race. The objective is to complete a straight 220m long track in the shortest time possible, focusing on acceleration and maximum vehicle speed. The maximum speed of the vehicle is also measured at the finish line, but this is only for informational purposes in the results list, and the winning team is the one with the shortest time [10].

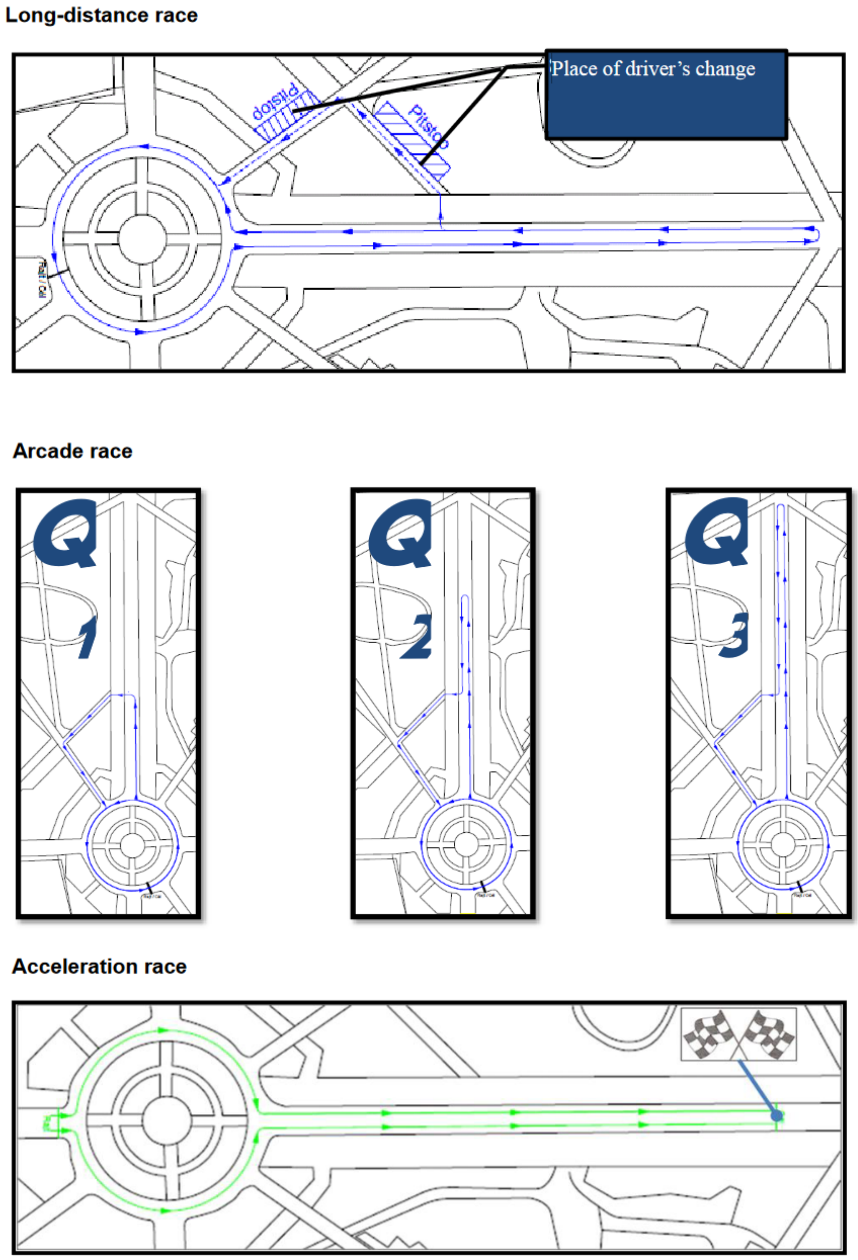

Fig. 1. Layout of competition track

The main criteria for selecting the propulsion unit were a low center of gravity, the possibility of symmetrical placement within the vehicle, and the option for variable cylinder operation. For these purposes, a rack-and-pinion transmission appeared to be ideal. The single-sided rack uses a rack with teeth on one side, along which a gear wheel rolls. The oscillating movement of the gear wheel is converted into unidirectional rotation further in the kinematic structure, as shown in the diagram (Fig. 2) [4].

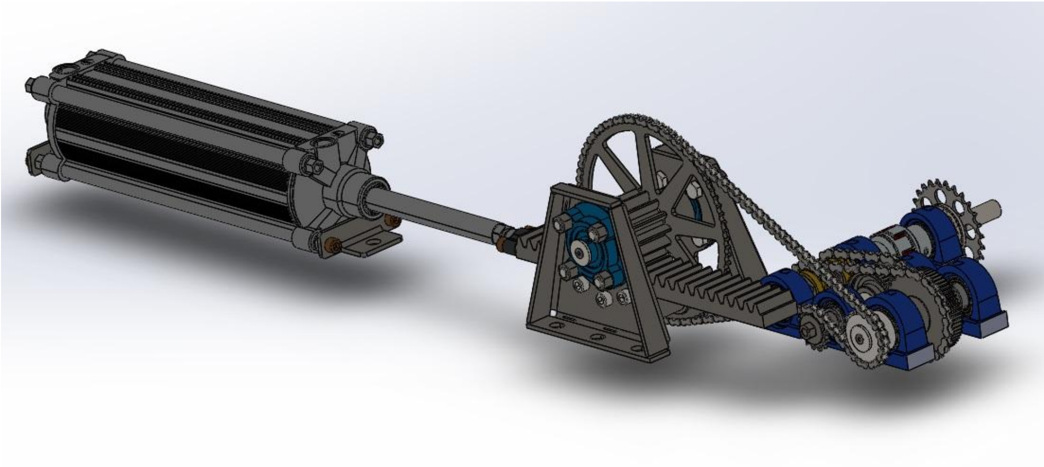

Fig. 2. One-sided rack transmission

The kinematic structure of the single-sided rack transmission is shown in Figure 3. As the piston rod extends, the gear and sprocket wheel (1) rotate in the positive direction, and this rotational motion is transmitted via a chain to the shaft (2). From shaft (2), the motion is transferred via a chain to the freewheel (4), which rotates the output shaft (6). Simultaneously, the rotational motion from shaft (2) is transferred to shaft (3) through a gear train, causing shaft (3) to rotate in the negative direction. The rotational motion from shaft (3) is transmitted via a chain to the freewheel (5), which freely rotates on shaft (6) in the negative direction, not affecting its motion [5].

When the piston rod retracts, the directions of rotation of the shafts and freewheels reverse, and torque is transferred to shaft (6) in the positive direction from freewheel (5), while freewheel (4) rotates freely. By this, we achieve a positive torque direction on the output shaft (6) throughout the entire cycle of the pneumatic cylinder [6].

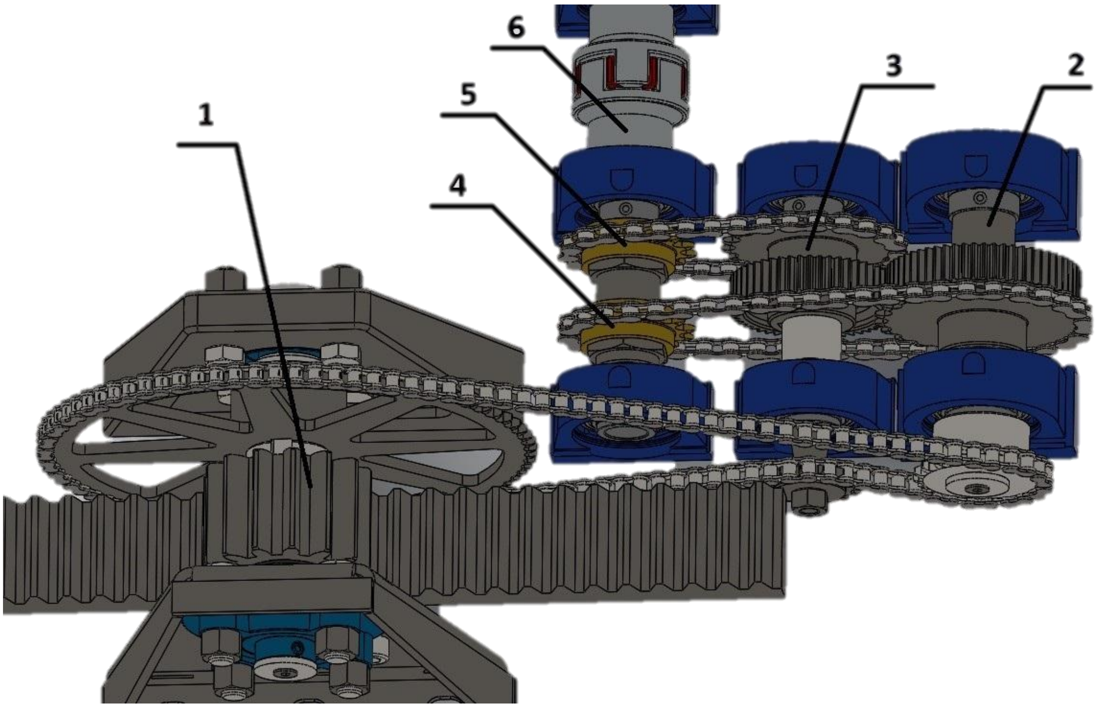

Fig. 3. Kinematics of the single-sided rack drive

Drive sizing: The sizing and design of the propulsion unit will begin with the rack, as we know the maximum force in the piston rod, which is equal to:

$$F_p = \pi . \left(\frac{d_{piest}}{2}\right)^2 . p_{\max} \cong 7854N \tag{1}$$

where:

- F_p ... force in the piston rod
- d_{piest} = 100mm ... diameter of the piston
- p_{max} = 1MPa ... maximum air pressure allowed by the cylinder manufacturer

The gear module influences many design parameters, so it is often selected based on an estimate. From this, other gear parameters are determined, and then a strength check is performed to verify if the chosen module is suitable. Since various qualitative parameters of the specific gear are needed for the calculation, we select a potential gear and rack from the supplier's catalog at this stage. In our case, we chose:

- m_n = 5mm ... gear module in the normal plane
- z_1 = 14 ... number of teeth on the pinion (with allowable undercut)
- b = 50mm ... width of the rack and pinion
- r_1 = 35mm ... pitch radius of the pinion

For gears, the load-bearing capacity of the teeth is checked for bending and contact stresses.

Sizing the first shaft, which will transfer torque from the selected pinion. The torque from this shaft will be transferred to a sprocket placed on the shaft next to the pinion. As will be shown, this shaft is subjected to a combination of bending and torsional moments. The bending moment along the length of the shaft depends on its dimensions and the placement of the pinion and sprocket, so a preliminary model must first be created. It is clear that to minimize the bending moment, it is advantageous to design the shaft as short as possible, with the pinion and sprocket placed as close to the bearings as possible. However, it is also necessary to consider various dimensional constraints of the propulsion unit. After considering these requirements, a preliminary 3D model was created, showing the forces acting on the shaft, as illustrated in Fig. 4 [4; 5].

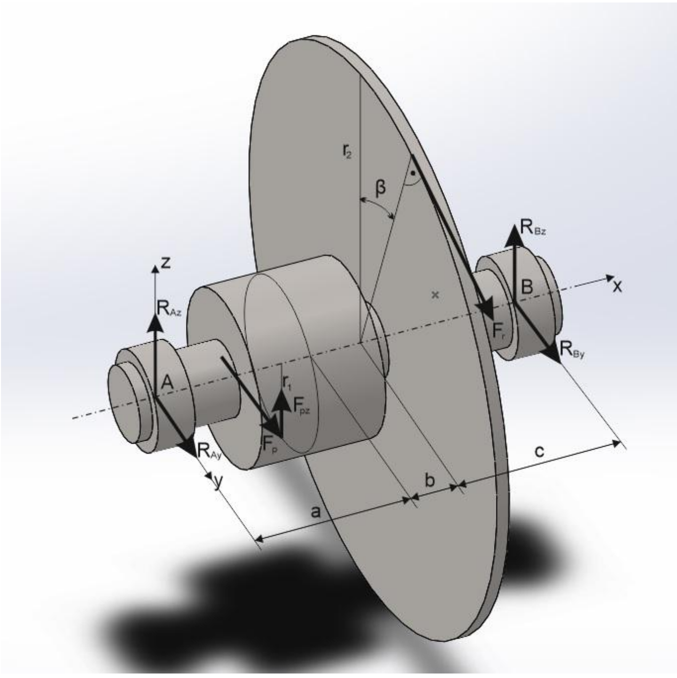

Fig. 4. The computational model of a shaft

a = 53mm ... the distance between the center of bearing A and the center of the pinion
b = 37.5mm ... the distance between the centers of the pinion and the sprocket
c = 55mm ... the distance between the centers of the sprocket and bearing B

r_1 = 35mm ... pitch radius of the pinion
r_2 = 115mm ... the pitch radius of the sprocket
β = 27° ... chain angle relative to the y-axis
F_p = 7854N ... force in the piston rod during extension
F_{pz} ... radial force on the pinion caused by the engagement angle of the involute gearing
F_r ... force in the chain of the sprocket
R_{Ay} ... reaction component in bearing A in the direction of the y-axis, other reactions are analogous according to the indices.

We calculate the force F_{pz} as:

$$F_{pz}= F_p.\mathrm{tg}\alpha \cong 2860N, \tag{2}$$

where α = 20° is the pressure angle of the selected gear. The force in the chain, F_r, is determined from the moment condition around the x-axis. Since the lines of action of forces F_p and F_r are the only ones that do not intersect the x-axis, the moment condition is expressed as:

$$F_p.\ r_1 = F_r.\ r_2. \tag{3}$$

By substituting the known values, we obtain the force $F_r \cong 2390N$. Based on this force, we can now select a suitable chain. It is clear that the torque is transmitted by the section of the shaft between the pinion and the sprocket, and it can be calculated as:

$$M_{k1}=F_p.r_1=F_r.r_2\cong 275Nm. \tag{4}$$

For the calculation of reactions in the bearings and bending moments in the shaft, we can now move the points of application of forces F_p and F_r to the x-axis, as their previously calculated rotational effect is irrelevant for this calculation. It is appropriate to solve the task by projecting into the *xy* and *xz* planes. The force in the chain F_r is the only one that has projections in both planes.

$$\begin{gathered} F_{ry}=F_r.\cos\beta \cong 2130N; \\ F_{rz}=F_r.\sin\beta \cong 1085N. \end{gathered} \tag{5}$$

The components of the reactions are calculated from the moment conditions around points A and B and using the method of an imaginary cut, we determine the course of the bending moment M_{oz}. We obtain the components of the reactions in the bearings:

$$\begin{gathered} R_{Az}\cong -1408N; \\ R_{Bz}\cong -367N. \end{gathered} \tag{6}$$

We calculate the course of the bending moment, and the critical point is at the location of the pinion where *x* = *a*. At this point, the torque M_{k1} = 275Nm acts, and the bending moment is:

$$M_o=\sqrt{M_{oy}{}^2+M_{oz}{}^2}\cong 316Nm. \tag{7}$$

For the reduced stress according to the HMH strength theory, which is suitable for materials in a ductile state:

$$\mathrm{d}=\sqrt[6]{\frac{(32.M_o)^2+3.(16.M_k)^2}{(\pi.\sigma_{red})^2}}. \tag{8}$$

If we substitute the allowable stress σ_d = 140MPa (selected from engineering tables for the chosen material, steel C50) into the given relationship for σred, we obtain a minimum allowable diameter of the component d_{min} = 30mm at this stress level. For the transmission of torque from the pinion to the shaft, we will use a tight fit, so the reduction in cross-section due to the groove must be taken into account during dimensioning. Similarly, we could calculate the minimum allowable diameter of the shaft at each cross-section from the computed bending and torque moments. However, from a design perspective, the shaft must have cylindrical surfaces due to the bearings.

The focus is on determining the minimum cross-section for the bearings, as the bending moments in this part of the shaft are significantly smaller and the torque is completely absent. It is also necessary to check the shaft during the retraction of the piston rod, as the force during retraction is lower than during extension, and the sliding effects of the forces in the chain and piston rod cancel each other out (they were summed during extension). Therefore, we can conclude that the stresses during the retraction of the piston rod will be lower. If the calculated parameters of the shaft significantly differ from the preliminary design, we can adjust the design based on these parameters, which may change the input parameters, and the shaft must be recalculated from scratch.

Conclusions. By the time of completing this article, the vehicle was already mechanically complete. Preliminary tests in the interior on supports and the road at reduced filling pressure without a functioning transmission showed no issues with the power unit, so we can consider the task accomplished.

From the perspective of the power unit, the task for the next competition year is to design a clutch or reverse drive, as the current kinematic structure causes the rear wheels to lock up when attempting to push the vehicle backward. It would also be advisable to examine the usability of the transmission range with the current gearing and optimize it to achieve the maximum possible torque at the wheels for acceleration while maintaining a usable speed range.

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КОНСТРУКЦІЯ ЕНЕРГОЗБЕРІГАЮЧОГО ДВИГУНА ДЛЯ АВТОМОБІЛЯ З ПНЕВМАТИЧНИМ ПОРШНЕВИМ ПРИВОДОМ

Ця стаття являє собою комплексне дослідження механічної конструкції та оптимізації силового агрегату транспортного засобу, зосереджуючись на підвищенні продуктивності та маневреності для конкурентних сценаріїв. Попередні випробування показали надійність силового агрегату в різних умовах; однак специфічні проблеми, такі як блокування заднього колеса під час маневрів назад, вимагають подальшого дослідження. Дослідження спрямоване на розробку механізму зчеплення або приводу заднього ходу для покращення маневровості та оцінки поточної системи трансмісії для оптимальної доставки крутного моменту при збереженні ефективного діапазону швидкості. Крім того, у дослідженні вивчається інтеграція силового агрегату з кінематичною структурою, забезпечуючи цілісну конструкцію, яка максимізує ефективність роботи. Довгострокова надійність в умовах конкурентного стресу та відгуки користувачів від операторів також будуть розглянуті для покращення функціональності автомобіля. Вирішуючи ці цілі, дослідження намагається здійснити внесок в інноваційні рішення, які підвищать загальну продуктивність автомобіля та конкурентоспроможність у майбутніх подіях.

Ключові слова: *пневматичний транспортний засіб; пневматичний поршень; стиснене повітря; рейкова передача.*

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