Strength analysis of the drive wheel hub of a hydrogen-powered prototype hyper-light vehicle

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Abstract. This article presents a strength analysis of the rear wheel hub of the prototype vehicle for the Shell Eco-Marathon competition. Due to only one wheel in the vehicle’s driving axle, all drive components and the wheel are subjected to higher loads. Therefore, the rear hub must withstand the loads generated by the rear part of the vehicle and the powertrain. The aim of the paper was to develop a hub with the minimum possible weight while fulfilling the strength conditions. The low weight of the hub is particularly important because of the efforts to achieve the highest possible efficiency of the drive system, which generates less energy loss and the ability to travel a longer distance on the same amount of fuel. The article presents the developed three-dimensional model of the hub. The suitable materials for the individual elements that are parts of the whole hub were selected. The three-dimensional model was tested to strength analysis using the finite element method with the use of Abaqus® software. The developed calculation mesh as well as the assumptions and boundary conditions of the modelling carried out are presented. The results of the simulation tests, which are the basis for further design work are also described.

1. Introduction
“Hydros” is a hydrogen vehicle created by the students of the Lublin University of Technology’s Air Propulsion Scientific Circle for the European final of the Shell Eco-Marathon competition “https://www.shell.com/make-the-future/shell-ecomarathon/europe.html”. In the competition, student teams from the best universities in Europe compete with each other, whose aim is to drive the vehicle, built by these students, as far as possible using as little energy as possible. Most of the components of the vehicle are designed by the team members.

“Hydros” is a prototype electric vehicle powered by a hydrogen cell, designed to be as energy-efficient as possible. The vehicle’s design, based on carbon fibre composite technology, has been developed to achieve the lowest possible weight and maintain the required rigidity.

In order to achieve the best possible result, it is important to strive to reduce the weight of the entire vehicle while maintaining sufficient structural strength and the regulatory requirements of the Shell Eco-Marathon competition. Many elements have been successfully adapted from mountain biking, but in the course of the research, a point has been reached where these elements have not been performing satisfactorily mainly due to driving forces, the nature of which is slightly different from typical cycling applications. One of the sensitive elements was the rear wheel hub. Compared to bicycle hubs, it is subject to lower loads, especially impact loads. It is therefore possible to reduce the weight of the hub
by maintaining the required strength. Moreover, it was important to reduce energy losses and for this purpose ceramic bearings with low rolling resistance were used.

Simulation tests of element strength are often used in the design of machines, they allow to evaluate mechanical loads in given elements of the structure and on their basis to optimise the structure [7-9]. The results of these tests can be directly used in the design process. The basic approach is to determine the load level with the indication of the places of the highest stress concentration [10-12]. The second approach is to determine places with stresses over the limit values – a method particularly common in composite structures [13, 14]. The most advanced approach is to use automatic optimisation of the shape of the element in order to make the most of the strength of the structure or to distribute the stresses more evenly [15-17]. As emphasised by the authors of the works [13, 15, 17], the proper preparation of the geometric model is essential to optimise the structure. This is primarily related to its parameterisation of dimensions. It is also important to enable the parameterisation of materials [16].

The main aim of the works described in the article was to determine the areas of the highest stress concentration. This will allow determining the strength condition of the developed structure. The second, equally important goal is to develop a method of verification of this element during its structural changes. This will allow shortening the design time. The third goal was to indicate areas of potential structural optimisation in terms of minimising the mass of the system.

2. Research object
The object of research in the presented article is the drive hub of the prototype vehicle. The analysed hub is a part of the rear-drive wheel that rotates on a bearing axle fixed to the rear suspension fork. The spokes that connect it to the rim pass through the hub flanges. In addition, the drive hub is connected to the free-running mechanism, which is driven by a chain transmission from the electric motor. The developed hub unit is built following generally accepted design solutions commonly used in the bicycle industry, although several modifications have been added that have individually adapted the unit to the requirements. Figure 1 shows the rear suspension of the vehicle.

![Figure 1. Vehicle’s rear suspension.](image-url)
3. Wheel hub design

3.1. The project objectives
The design of the new drive hub was based on the modernisation of the design solution of the existing drive system. Analysing this design and the structural solutions used in the bicycle industry, a new design was developed to improve strength and functionality, reducing weight by 5%.

The following design assumptions were adopted:
1) Wheelbase, equal to 135 mm;
2) The need to adapt the hub to the elements that work with it. These include:
   a) brake disc – its position should be compatible with a brake caliper fixed to the vehicle structure, the mounting bolts of the disc being 44 mm apart;
   b) sprocket – its position should ensure that the chain which transmits the drive from the engine is working perfectly;
   c) hub axle – the diameter at the ends of the component corresponds to the diameter of the clamps on the vehicle suspension;
   d) spokes – diameter of through holes in hub flanges adapted to bicycle standards.
3) Weight reduction by using optimally selected materials.
4) Use of ceramic single-row bearings marked N6902P-LL.
5) Placing of a freewheel rotation allowing the wheel and the main body to rotate freely while the drive system remains stationary.
6) Easy replacement of consumables.

3.2. The developed hub model
The 3D model was created using CATIA® V5-6R2019. The design process was based on partial modelling of the general construction of bicycle wheel hubs. This is due to previous assumptions of wheel hub functionality. The final model of the hub to be produced is shown in figure 2.

![Figure 2. The final model of the wheel hub accepted for production.](image-url)
The wheel hub (figure 2) consists of the main axle passing through the whole element (1), on which two pairs of bearings (4) are mounted. The main bearing, which determines the arrangement of rotating elements on the axle, is located in the main body (3). Both its raceways are locked to prevent its movement in relation to the axle. The other bearings are positioned by means of distance bushings (5) and (10), distance washers (6), which block the movement of the internal raceways. The bearings are additionally locked through locknuts (2). This design allows for minimising the stress during hub operating and easy replacement of operating elements such as bearings.

The connection of the body to the freehub body (9) and the transmission of the drive from the electric motor to the wheel is carried out by using two slow-running rings (7), placed one each in the main body and the freehub body respectively through dedicated spline sockets (8). The rings are equipped with mutually cooperating pawls. This design allows minimising the loss of the drive transmission by changing the mutual pressure of the rings using pressure springs, to obtain a reliable drive transmission with minimal rolling resistance when working in bulk, without using a drive motor. The use of bicycle standards such as the spacing and size of the mounting holes for brake discs or the diameter of the spoke holes allows us to minimise production costs and the final cost of the entire drive wheel and ensures easy accessibility and a wide range of wear parts.

An element that required more attention in terms of construction work was the hub body shown in figure 3. It has several holes to which bicycle type spokes and brake disc are attached. Besides, it is a load-bearing element in the rear wheel hub system.

Figure 3. 3D model of the wheel hub body.

The model is made in a parametric way. The geometry was prepared in such a way that it is possible to easily modify its shape by changing selected geometrical parameters. It required dependence of dimensions of particular elements of the structure on selected, key parameters such as:
1) diameter of the spacing of holes for spokes,
2) the diameter of the spacing and the diameter of the brake disc mounting holes,
3) minimum permissible wall thickness in the middle section,
4) minimum radii of transition between planes.

The model developed in this way will allow carrying out tests to optimise the structure after developing further test methodology.
3.3. Selection of the material to be used for the elements

Another important element in the development of the methodology for optimising the construction of this component was to develop a selection of materials. As in the case of geometry, the possibility of changing the materials of individual elements of the structure was introduced. The materials have been selected to reduce the weight of the vehicle and ensure safety while driving. Two materials were used: Aluminium alloy PA9 and AISI 4130 steel alloy.

The material used for the hub axle is a low carbon alloy of AISI 4130 steel. Alloyed steel has higher wear resistance than unalloyed steel. This feature allows for increasing the durability of the element. Low carbon steel has good shock and impact resistance. High strength allows safe movement on potholed roads without fear of damaging the system [1]. The material has high hardness and creeps resistance. It is used for the production of bushings, rims, and light structures of sports and high-performance vehicles [2].

| Table 1. Properties of AISI 4130 steel [3]. |
|------------------------------------------|
| Tensile strength, ultimate               | 560 MPa |
| Tensile strength, yield strength         | 460 MPa |
| Modulus of elasticity                    | 190-210 MPa |
| Bulk modulus (typical for steel)         | 140 GPa |
| Shear modulus (typical for steel)        | 80 GPa  |
| Poisson’s ratio                          | 0.27-0.30 |
| Hardness, Brinell                        | 217     |

The material chosen for the hub body is Aluminium 7075 PN alloy, it has a much lower specific weight than steel and also has good resistance to weathering [4]. Aluminium alloy 7075 PN EN is characterised by high strength properties, very good machinability, and polishing [5].

| Table 2. Properties of aluminium PA9 (AW-7075) – physical properties. |
|-----------------------------------------------------------------------|
| Density                                                                | 2.81 g/cm³ |
| Young’s modulus (E)                                                   | 72 000 MPa |
| Modulus of lateral elasticity G                                       | 27 100 MPa |
| Poisson’s ratio                                                       | 0.33       |
| Coagulation temperature                                               | 475 °C     |
| Melting temperature                                                   | 635 °C     |
| Specific heat capacity                                                | 862 J/kgK  |
| Coefficient of Linear Thermal Expansion                               | 23.5 µm/mK |

This material is used in machine elements and structures operating under heavy loads. It is also used in the aviation industry and the production of sports equipment elements [6]. The use of this material reduces the vehicle weight without reducing vehicle strength and reliability.

4. Simulation tests

4.1. The aim and scope of the tests

The presented simulation tests aim to analyse the strength of the developed hub structure. This was obtained by performing an MES analysis of the developed vehicle hub model using the Abaqus® program.

The scope of the tests includes static load conditions defined from the maximum load of the rear suspension assembly and the maximum driving torque transmitted from the electric motor to the wheel.
4.2. Development of a computational model

The numerical model was prepared using the Abaqus® program. The finite element simulation was performed as a static calculation taking into account geometric nonlinearities – resulting from large deformations. In the study, an elastic-plastic material model with binary characteristics was used (using adequate material models suitable for AISI 4130 steel and PA9 aluminium). The computational model consisted of two main components: a fixed shaft with bearings and a hub body with a freehub body. The boundary and load conditions of the model were adopted based on an analysis of the actual operating conditions of the system, declaring the relevant restraints and loads. Edge conditions of the numerical model included mainly:

- restraint of both ends of the hub shaft;
- definition of the load resulting from the force applied to the wheel \( F = 442.5 \) N;
- determination of the torque on the appropriate hub body diameter \( M_s = 20,000 \) Nmm;
- taking into account the load from the bearing arrangement of the structure – through a declaration of forced displacement, in the direction perpendicular to the axis of the hub shaft at 0.005 mm. The boundary conditions described above are detailed in figure 4.

![Figure 4. Boundary conditions of the discrete model.](image)

Based on the previously developed geometrical model, a model for simulation tests was developed. The discrete model generally consisted of 47,000 finite elements and 89,170 calculation nodes. Every single finite element with a tetragonal structure (within the whole discrete model) had the designation C3D10 – which directly meant using the square shape function in the description of the element and having 3 translational degrees of freedom, in each of the 10 calculation nodes. The discrete model is presented in figure 5.
4.3. Results of strength tests

The results of the simulation studies are presented below. The analysis was based on the distribution of reduced stress determined according to the Huber-Mises-Hencky (H-M-H) strength hypothesis and maximum system displacements.

Figure 6 shows the stress distribution throughout the model. As can be seen, the maximum stress distribution for the whole system of about 344 MPa occurs in the area of the pressure nut.
the end of the cylindrical surface of the freehub body. This is due to the application of a torsional moment. Comparing the maximum stress, which for this system was 86 MPa to the assumed yield point in the range 390–470 MPa, we can certainly say that the system has sufficient strength. Another area of increased stress is the edge at the point of diameter changes on the freehub body. Stress at this point reaches 21 MPa. The other stress areas do not exceed 10 MPa, so these areas can be optimised in terms of stiffness and minimisation of the weight of the assembly.

A similar stress situation was observed within the hub shaft with bearings (figure 8). The maximum stress reached 344 MPa. Comparing this value with the strength limit of 480–540 MPa, we can conclude that the geometry and material of the part have been properly selected and can be further reduced in terms of stress to the mass of the assembly.

![Figure 7](image1.png)

**Figure 7.** Maximum stress level within the hub body.

![Figure 8](image2.png)

**Figure 8.** Maximum stress level within the hub shaft with bearings.
Considering the following arrangement (figure 9) in terms of maximum displacements which are located in the central part of the shaft, it was observed that they reached a value of 0.006 mm. This is due to the forces acting on the wheel, whose value is assumed to be $F = 442.5$ N. These values are regarded as acceptable and do not seriously affect the proper functioning of the hub during use. In addition, the result obtained is such a low value that, concerning the actual operation of the component in question, the deflexion obtained will not adversely affect the load in the long term and no permanent deformation of the system will occur. However, further reduction in dimensions will lead to increased deformation. It must therefore be assumed that the solution adopted is optimal in this aspect.

After the identification of maximum loads in the structure, the analysis of areas that can be used to optimise the weight of the structure was also carried out. In the case of the hub body, it is visible that the area most susceptible to optimisation is in the middle part of the structure. Stress in this area does not exceed 10 MPa and deformations are significantly below the limits. Further research should focus on optimising this area.

As far as the hub shaft is concerned, the greatest optimisation potential is also found in the middle part, but according to the deformation analysis, this area has a small reserve to limit torsional deformation.

![Figure 9. Level of maximum displacement within the hub shaft with bearings.](image)

5. Summary
Analyzing the results obtained, it can be stated that:

1) The maximum stress for the whole system was 344 MPa on the pressure nut. Under the influence of the application of torque acting on one of the ends of the cylindrical surface of the barrel body, the maximum stress was 86 MPa. This is much lower than the yield strength of the material used.

2) On the shaft of the hub axle in the bearing section, the maximum stress is 344 MPa, which is also within the permissible stress range.

3) The greatest displacement occurs in the central part of the shaft, between the bearings located in the hub body. The maximum value in this area is 0.006 mm, which is due to the forces that act on the wheel. However, these values are so small that they will not adversely affect the component being tested.

In conclusion, it can be stated that the designed drive hub of the vehicle will fulfill its role and its strength will be sufficient. The MES analysis allowed for verification of the design and indicated areas of possible structural changes allowing for further reduction of the weight of the element without losing its strength.
The prepared test method allows for further shape optimisation work with the assumption of the deformation and stress limits. This will allow for further research to obtain a lighter structure that still fulfills its function properly.

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