FEM analysis of selected construction bodies of the conceptual bike

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Abstract. The main subject of work is a concept bike featuring a non-standard construction. The classic frame has been replaced with a straight beam. Consequently, there is a much greater stress on some of the components of the bike body. Furthermore, the construction of these body components becomes more complex which calls for the employment of the numerical method of stress calculation. The work involved carrying out a strength analysis for several selected components of the bike body utilizing the finite element method (FEM). The primary object of analysis was the wheel rotation mechanism. The result of the analysis allows to develop an optimized construction, allowing to select the most suitable materials and to determine the appropriate and compact dimensions.

1. Introduction
The purpose of the work is to determine the stresses present in the steering mechanism of the front wheel while riding the conceptual bike (the project is at the initial design stage) and possibly develop an optimal construction solution in terms of materials and dimensions, with utilization of the finite element method. The initial visualization of the vehicle is shown in figure 1. The steering mechanism is mounted in the front wheel rim under a cover (figure 2).

Figure 1. The initial visualization of the conceptual bike.
2. Forces and moments
The key aspect of work is to determine the loads. It was assumed that the bicycle would be made of carbon fiber with density of 2400 kg m$^{-3}$. An additional 20% weight increase was also assumed for individual metal parts that are part of various mechanisms or gears. The vehicle's weight was estimated at 36 kg. Furthermore, an increased weight of the driver was considered—whereas Deutsches Fachbuch [1] proposes to use 80 kg, 100 kg was assumed for this study. Afterwards, two main load cases (without air resistance) were devised. From these cases, the forces and moments present in the analyzed mechanism were determined.

Load case 1: Static, wheel turn
Based on the diagram in figure 3, reaction forces were determined using formulas 1 to 3.
\[
G = G_v + G_h
\]  
(1)

\[
G = m \cdot g = (36 \text{ kg} + 100 \text{ kg}) \cdot 9.81 \frac{m}{s^2} = 1334 \text{ N}
\]  
(2)

\[
\sum M_v = G_h \cdot l_A - G \cdot (l_A - l_s) = 0
\]  
(3)

\[
G_h = \frac{G \cdot (l_A - l_s)}{l_A} = \frac{1334 \text{ N} \cdot (1400 \text{ mm} - 600 \text{ mm})}{1400 \text{ mm}} = 762 \text{ N}
\]

\[
G_v = G - G_h = 1334 \text{ N} - 762 \text{ N} = 572 \text{ N}
\]

Figure 4 presents the visualization of the steering mechanism. It is connected with two parts forming an extension of the bicycle's main frame. There are two such parts: left and right and are actuated by linear electric motors which motors move the gear racks in the desired directions.

![Figure 4. Visualization of the steering mechanism – right turn.]

These gear racks are pushed with certain forces that must overcome the so-called the wheel turning resistance that is highest at zero speed – figure 5.

![Figure 5. Diagram for determining the wheel turning resistance (the cross-section shows the shape of the contact surface of the wheel with the ground); 2a – longer side of the tire imprint on the ground [2–4].]

The gear rack is coupled with a gear pinion mounted on the main shaft of the rotary mechanism. A torque is generated in the mechanism. This moment must overcome the friction that occurs in the cross-section of the tire's contact area with the ground, its value was derived from formula 4.
\[ M_s = \frac{2}{3} a \cdot N \cdot \mu \]  

(4)

The pressure force \( N \) was assumed to be equal to the reaction force of the ground, parameter \( 2a \) value was estimated at 100 mm, and the adopted coefficient of friction was \( \mu = 0.95 \).

\[ M_s = \frac{2}{3} a \cdot G_v \cdot \mu = \frac{2}{3} \cdot 50 \text{ mm} \cdot 572 \text{ N} \cdot 0.95 \approx 18000 \text{ Nmm} \]

Load case 2: Dynamic, sudden braking, maintaining balance by turning the wheel. Based on formula 5, the deceleration (parameter \( a \)) value during braking was calculated – from the initial speed \( v = 30 \text{ km h}^{-1} \) in time \( t = 2 \text{ s} \)

\[ a = \frac{\Delta v}{\Delta t} = \frac{v_k - v_p}{2 \text{ s}} = \frac{0 - 30 \text{ km h}^{-1}}{2 \text{ s}} = -\frac{30}{3600} \cdot \frac{1000 \text{ m}}{2 \text{ s}} = -4.2 \text{ m s}^{-2} \]  

(5)

To determine the dynamic increases during braking, the diagram presented in figure 6 was used.

![Figure 6. Braking; \( \Delta G \) – dynamic increase, \( F_{\text{träg}} \) – inertial force equivalent to the braking force [1].](image)

Calculations based on formulas 6 and 7.

\[ F_{\text{träg}} = m \cdot a = 136 \text{ kg} \cdot 4.2 \frac{\text{m}}{\text{s}^2} = 571 \text{ N} \]  

(6)

\[ \sum M_h = F_{\text{träg}} \cdot h_s - \Delta G \cdot l_A = 0 \]  

(7)

\[ \Delta G = \frac{F_{\text{träg}} \cdot h_s}{l_A} = \frac{571 \text{ N} \cdot 0.5 \text{ m}}{1.4 \text{ m}} = 204 \text{ N} \]

3. FEM

The most important for this analysis [5, 6] is to determine the dimensions of the main shaft on which all the rotating parts that are part of the mechanism are mounted. The baseline design requirement is to
maintain the relatively small dimensions of the rotation mechanism, to maintain the impression that there is no visible steering device of the wheel. Therefore, during the analysis, the stage of shaft geometry simplification (chamfers, roundings, machining undercuts, keyways) was abandoned so that the stress distribution was as approximate as possible. AISI 4130 / 25CrMo4 improved alloy steel (material number: 1.7218) was selected as the material to handle ultra-high loads, with tensile strength limit of \( R_m = 1172 \) MPa and yield point equal to \( R_e \geq 700 \) MPa.

Figure 7 shows the results of the main shaft analysis for the first load case. In the area where the highest stresses were expected, the number of finite elements was increased. Stresses at the level of over 400 MPa are relatively high, even when selecting a material with yield strength at almost double the value of stress.

![Figure 7](image)

**Figure 7.** Results of the shaft analysis for the first load case: a) mesh, b) stress distribution.

As expected, the next stage calls for a dynamic analysis of the machine shaft. See figure 8 for the types of load cycles. It is anticipated that the torque acting on the shaft alternately (depending on the direction of turning the wheel) necessitates to assume at least mutually variable loads.

![Figure 8](image)

**Figure 8.** Types of load cycles and stresses: a) constant, b) unilaterally variable (1 – pulsating, 2 – pulsating unilateral), c) mutually variable (3 – swinging symmetrical, 4 – bilateral asymmetrical), d) transient, \( T \) – period (load and stress change cycle) [7].

The following parameters were analytically calculated for re-analysis: fatigue strength from formula 8, safety factor from formula 9 and allowable stresses from formula 10. The safety factor was adopted at 3.4 – formula 9.
\[ Z_{so} = 0.3R_m = 352 \text{ MPa} \]  \hspace{1cm} (8)

\[ x_z = 3.4 \div 3.7 \]  \hspace{1cm} (9)

\[ x_z = 3.4 \]

\[ k = \frac{Z_{so}}{x_z} = \frac{352}{3.4} = 104 \text{ MPa} \]  \hspace{1cm} (10)

This leads to an optimized shaft geometry – as shown in figure 9. After optimization, a re-analysis was performed as shown in figure 10.

**Figure 9.** Shaft geometry optimization.

**Figure 10.** Results of the shaft re-analysis for the first load case after geometry optimization.

The highest concentrated stresses are two times smaller than the expected allowable stresses. At this stage, it can be said that the dimensions of the shaft are approximately correct. This is all to detail the
other parts in the front of the bike. In the final stage, fatigue analysis of the entire mechanical assembly will be carried out utilizing the connection mode. The final analysis will account for the second load case – the results are shown in figure 11.

![Figure 11. Results of the shaft analysis for the second load case after geometry optimization: a) mesh, b) stress distribution.](image)

Figure 11. Results of the shaft analysis for the second load case after geometry optimization: a) mesh, b) stress distribution.

Figure 12 shows a possible representation of the mechanism cover after re-analysis. Its aesthetics and compactness is significantly reduced.

![Figure 12. View of the mechanism cover: a) before analysis, b) after optimization.](image)

Figure 12. View of the mechanism cover: a) before analysis, b) after optimization.

The purpose of the design was to hide the device controlling the steering angle of the wheel as much as possible. It is possible to reduce the dimensions of the entire mechanism under existing loads by increasing the diameter and maintaining the current shaft length. Consequently, this calls for the use of non-standard prismatic grooves – possibly splined. Another optimization of geometry was carried out and the shaft was subject to another analysis. The results are shown in figure 13.
Figure 13. Results of the shaft analysis for the first load case after geometry re-optimization: a) mesh, b) stress distribution.

As can be seen, the shaft length in this case has no effect on the stress distribution. The only difficulty in the construction process may be the selection of appropriate bearings with a relatively small width. Finally, an analysis was carried out for the two connecting parts which transfer the load directly from the shaft to the final restraint of the part being the extension of the main frame. Figure 14 shows that the fastener is too thin to transfer bending forces. Where the largest stress distribution was anticipated, the number of finite elements was increased.

Figure 14. Results of the connector analysis for the first load case: a) mesh, b) stress distribution, c) the area of highest stresses and concentrated stresses.
It is worth noting that the connector geometry presented in figure 14 already accounts for the change in the main shaft geometry. This means that originally the connector was relatively much smaller and all the more unsuitable for carrying the given load. Figure 15 shows the result of the reanalysis after geometry optimization.

![Figure 15. Results of the connector re-analysis for the first load case after geometry optimization.](image)

As for the part which forms an extension of the main frame, the analysis will be carried out in the subsequent stages of the project, because this part is made of carbon. For such materials, the distribution of stress depends on the direction of composite fibers and the results of the examination will be presented in a separate article.

4. Conclusion
These tests allowed to receive stress approximation in a static analysis. Consequently, dimensions were identified to indirectly account for variable loads. In the further stage of research, the mechanism will be designed in greater detail. Presently, the shaft dimensions will be used to determine the dimensions of the mating parts. Finally, a dynamic analysis of the entire mechanism will be carried out using the connection mode.

5. References
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