



SHAFT DESIGN FOR ELECTRIC GO-KART

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Abstract

This article presents the results of work related to the design, analysis, and manufacturing of the shaft for an electric go-kart. Works considered the stiffness of the shaft for various conditions affecting the vehicle while driving. In the previous stage of the project, the electric motor and gear transmission were selected. The main goal of this case study was to design the shaft for 10 kW electric go-kart. The rear driving axle of the go-kart is not equipped with a differential. The equal rotational speed of two rear wheels causes that occurs skidding and greater forces acting on the vehicle and driver when cornering. We were considering two types of the shaft – full and drilled. The first one provides greater stiffness, the second one is “softer”. The analysis allowed for the selection of a more appropriate shaft, and then for its manufacture and assembly in the vehicle.

Introduction

Electric motorsport is becoming more popular and displacing traditional internal combustion engines. Formula 1, go-karts, Dakar rally, all these contests have their category tailored to electric vehicles. Electric motors used in drive systems need a different design of some components. The electric drive generates greater torque, higher efficiency, and weight is less than an internal combustion

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engine. Designed components have to take into account all these restrictions and often they are different in shape, dimensions, material.

Shafts transfer the torque from the motor to the wheels. Go-karts do not have a differential which causes that wheels have this same rotational speed. This kind of power transmission contributes to faster wear of tires and components of the driveline. A steering system equipped with adjustable steering tie rods allows controlling go-kart without differential (*The axle: theory, practice and quick tips* 2016, *What is a Live Axle on a Go-Kart?* 2020, *Why Go-Karts Don't Have a Differential* 2021). The steering knuckle of the go-kart lifts the outer wheel slightly upwards during a turn. Force of inertia tilts the go-kart so that it passes the turn on three wheels – two front and one rear (*The Axle* 2016).

The shaft is exposed to torque, bending moments, and forces which affect depending on the drive transmission method. The appropriate design of the shaft is to prevent not only the possibility of twisting the axle but also to provide the required stiffness, which translates into vehicle steerability (MIRONE 2010).

Go-karts shaft has one diameter on the whole length. There is no place to bearing placing. To solve the problem of bearing mounting go-kart shaft is supported by radial insert ball bearings. The outer race ring is convex, which allows for the coordination of shaft misalignment. To set the position of bearing we use a set screw (*SKF spherical roller thrust bearings. For long lasting performance* 2010, STEENEKAMP, SWART 2020). The use of a radial insert ball bearing allows for quick and easy assembly.

Currently, there are two methods of designing shafts for karts (*Axle evolution* 2016). One is to use a shaft with a rod cross-section which is stiffer, the other with a pipe cross-section which is less stiff (Fig. 1).

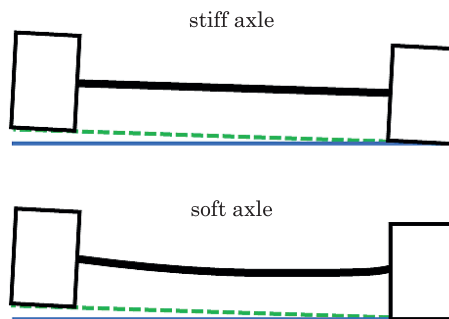


Fig. 1. Comparison of stiff and soft axle during cornering

The goal of work was the choosing of the appropriate shaft design. The work included the design and analysis of shafts under the influence of forces acting on the shaft during operation. The selected shaft will be manufactured and used in the prototype of the electric go-kart.

Design

The stiffness of the driving axle depends on several factors – shaft diameters (outer diameter and in case of pipe cross-section also inner diameter), length, material (FIEREK et al. 2020). A rod cross-section shaft has usually 25 or 30 mm of diameter, a pipe cross-section shaft 30, 40, or 50 mm of diameter. Drilled shafts have different wall thicknesses. It is respectively 5 mm for ϕ 30 mm, 4 mm or 3 mm for ϕ 40 mm, and 2 mm for ϕ 50 mm. The choice of the type of shaft depends on the power of the vehicle, but also on its intended use – rental kart or racing kart. It was decided to check one material type – 42CrMo4 and two types of shafts – 30 mm full and 50 mm drilled with a 2 mm wall thickness. Parameters of the 42CrMo4:

- tensile strength R_m – 1,030 MPa;
- yield point R_e – 880 MPa;
- density – 7.83 g/cm³.

Parallel to the work related to the design of the go-kart structure, work was carried out on the electrical system and the power supply system. Loads were taken from the selected electric motor and gear transmission which was chosen in the previous stage of work. The parameters of the drivetrain are presented below:

- nominal power – 10 kW;
- torque – 30 Nm;
- RPM – 3,200 1/min – (max. power);
- ratio transmission between electric motor and go-kart shaft – 2.5:1;
- transmission type – tooth belt.

Most karts have 3 radial insert ball bearings (Fig. 2), but some manufactures use also only two. It was decided to check the use of three bearings in the analysis. It is a much more common solution in go-kart constructions.

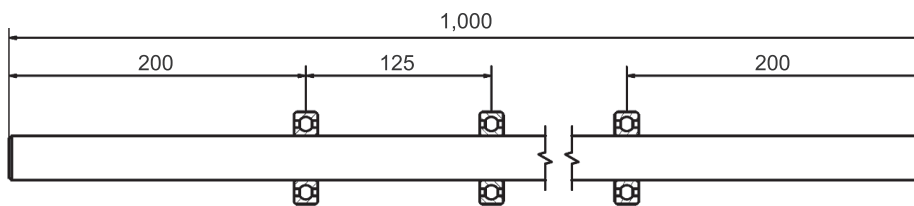


Fig. 2. Typical go-kart shaft construction diagram with bearings

Shaft analysis

The static analysis assumes examining the shaft at a specific point in time. Static analysis advantage is that there is no need to simulate the increase and decrease in the value of individual forces in the time domain. This assumption allows for precise determination of the critical forces and safety parameters that the designed part should meet. The adopted shafts loading conditions will allow us to know the stiffness of the shafts and to verify whether the analyzed shafts meet the assumed criteria (SINGH 2020, SATHISHKUMAR, UGESH 2016, NANDA, PARHI 2013). The location of the bearings, transmission, and the brake disc is shown in Figure 3.

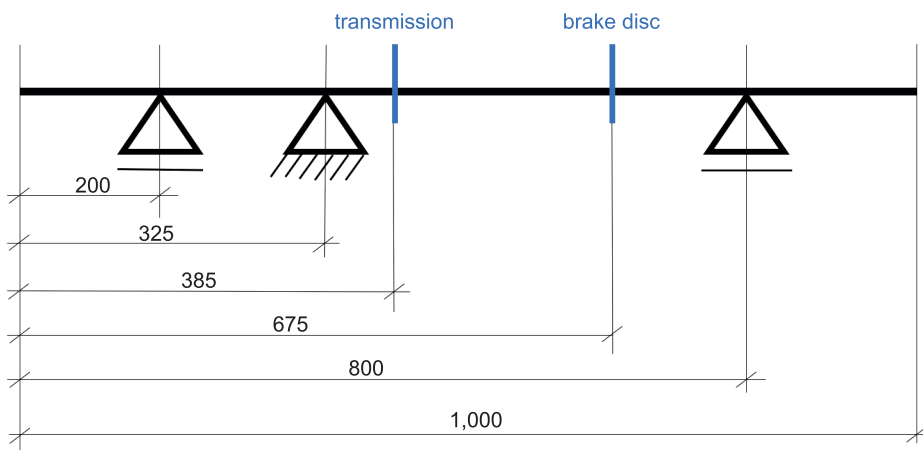


Fig. 3. Shaft diagram included in the analysis

We prepared simulation for shafts supported in three places and also for two situations on the tracks, which can be most critical during the race:

- driving at top speed on track and braking before cornering. The moment of application of the maximum forces just before the tires breaking traction is considered;

- drive through the chicanery, raise the model, and redirect the axial forces so that they act only on one wheel. The moment of leaving the turn is considered, therefore it is necessary to apply the maximum torque generated by the drive system.

First case

Data:

- the driving torque acting on the shaft is the moment of inertia of the wheels and the moment of inertia of the motor;
- the braking torque is the force generated by the caliper rubbing against the brake disc mounted on the drive shaft.

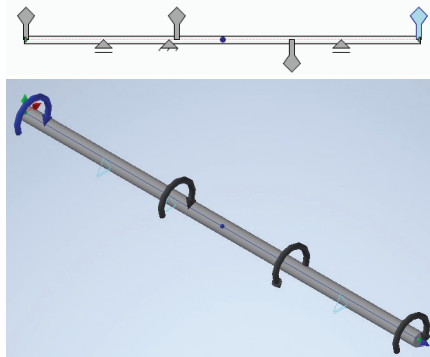
Acting forces:

- torque generated by one wheel and the car inertia – 22.5 Nm;
- moment of inertia of the motor – 5 Nm;
- braking torque – 50 Nm.

Table 1

Results for the first case study

The stress distribution along the axis of the tested shaft for the first load case



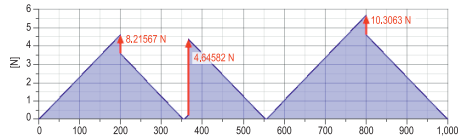
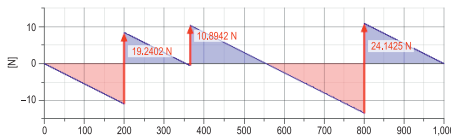
Full shaft

drilled shaft

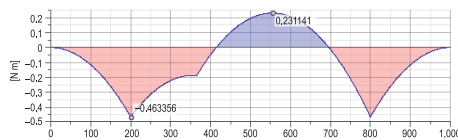
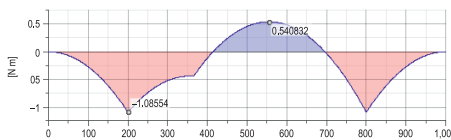
Outer diameter: ϕ 30 mm
 Material: 42CrMo4
 Weight: 5.54 kg

outer diameter: ϕ 50 mm
 wall thickness: 2 mm
 material: 42CrMo4
 weight: 2.36 kg

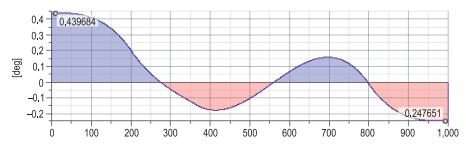
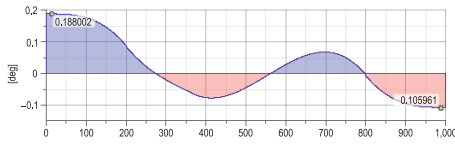
Shear strength



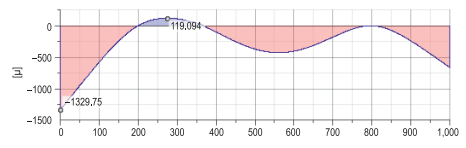
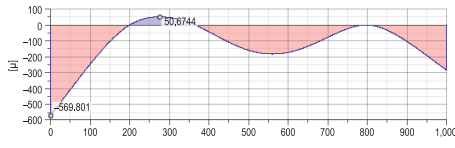
Bending moment



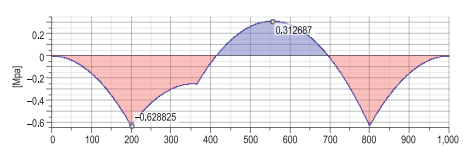
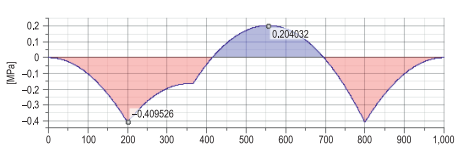
Deflection Angle



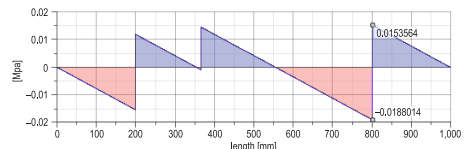
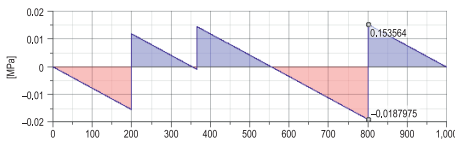
Deflection



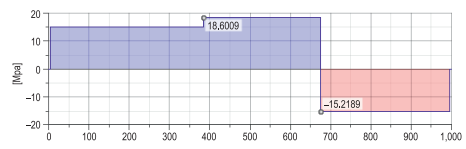
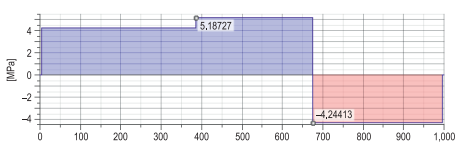
Bending stresses



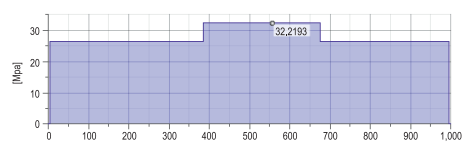
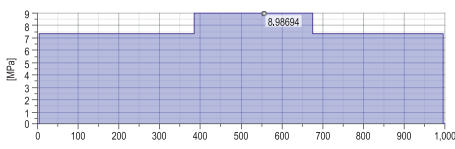
Shear stresses



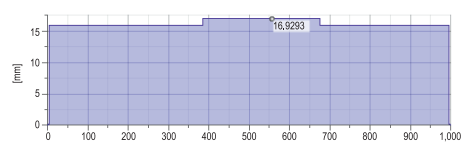
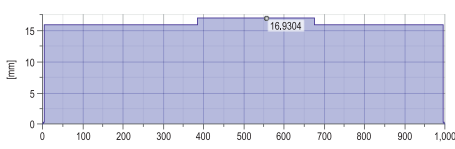
Torsional stresses



Reduced stresses



Ideal diameter chart



Second case

Data:

– the axial force is the centripetal force generated by the go-kart wheel as it turns. The force was taken as the weight of the kart multiplied by the maximum overload and an appropriate factor of safety;

– the torque acting on the axle is the maximum torque generated by the electric motor.

Acting forces:

– torque generated by the electric motor – 75 Nm;

– moment of resistance of the wheel touching the ground – 75 Nm;

– bending moment acting in the central part of the shaft due to the weight of the kart – 100 Nm;

– axial force acting on the tire during a turn – 3,000 N.

Due to the inability to model a pipe as an analyzed element in Autodesk Inventor, we decided to determine the cross-sectional area of the analyzed drilled shaft and model it as a full cross-section. For this we used formula of equivalent radius:

$$P = \pi R^2 - \pi r^2 \quad (1)$$

$$R_s = \sqrt{\frac{P}{\pi}} \quad (2)$$

where:

R – outer diameter of the drilled shaft,

r – inner diameter of the drilled shaft,

P – drilled shaft cross-sectional area,

R_s – equivalent radius of a drilled shaft.

Calculations have shown that the diameter of the replacement shaft should be approximately 19.6 mm.

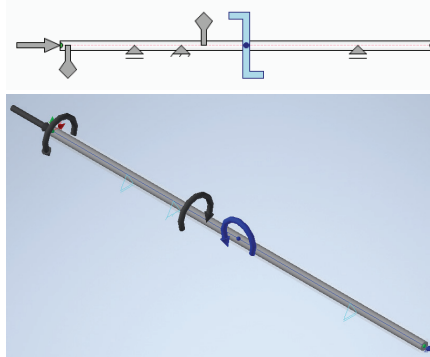
The ideal diameter chart was the information that allow us to verify if the chosen shaft design was able to meet the load criteria. The charts showed that for the first case study both shafts are designed properly. A second case study showed that parameters of the full shaft are sufficient but drilled shaft will not meet the criteria.

The next step after shafts analysis was related to the manufacturing of one type of shaft for prototype kart. After the results of the static analysis, we decided to choose the option of a full 30 mm diameter shaft. This kind of shaft provides the factor of safety requirements specified in the assumptions. Additionally, manufacturing a full shaft is a few times cheaper than the drilled shaft. The technological process of the full shaft is also less complicated.

Table 2

Results for the second case study

The stress distribution along the axis of the tested shaft for the second load case



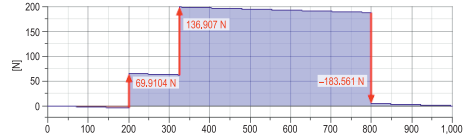
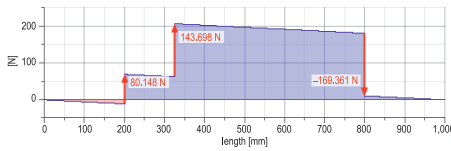
Full shaft

drilled shaft

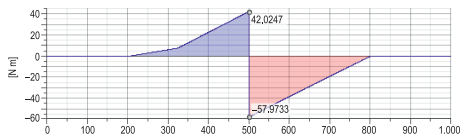
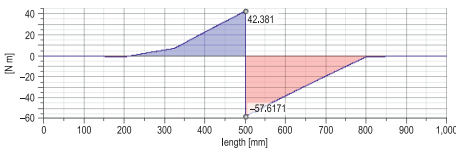
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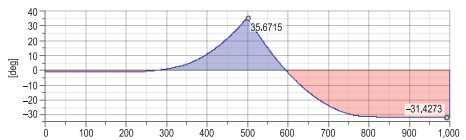
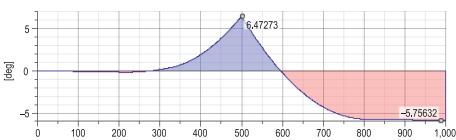
Shear strength



Bending moment

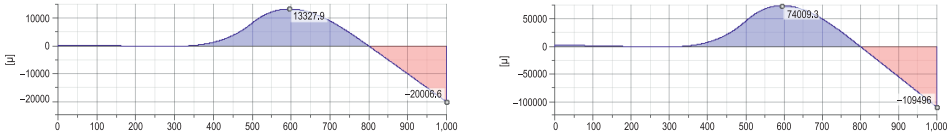


Deflection Angle

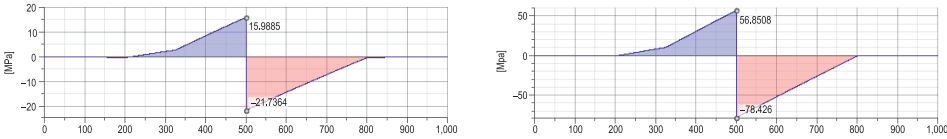


cont. Table 2

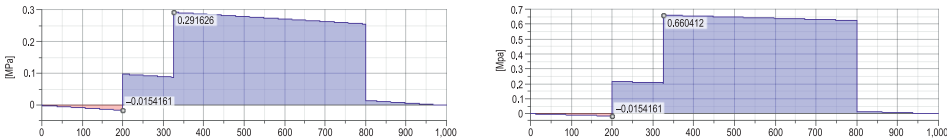
Deflection



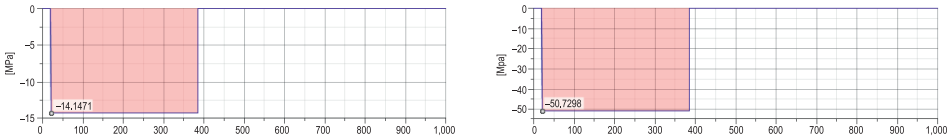
Bending stresses



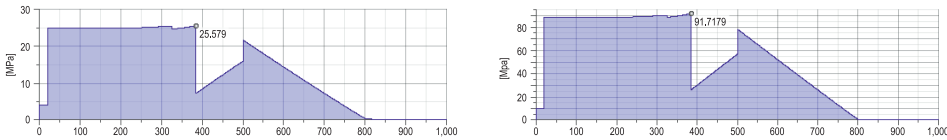
Shear stresses



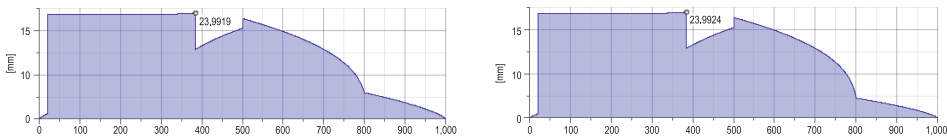
Torsional stresses



Reduced stresses



Ideal diameter chart



The produced shaft was used as the drive shaft for the prototype of an electric go-kart (Fig. 4). On the shaft was mounted brake disk and HTD 56-8M-30 TL2012 gear wheel using Taper Lock fastening sleeve. To place the bearings to the go-kart frame bearing carriers were attached (Fig. 5).



Fig. 4. Shaft mounted in the go-kart chassis

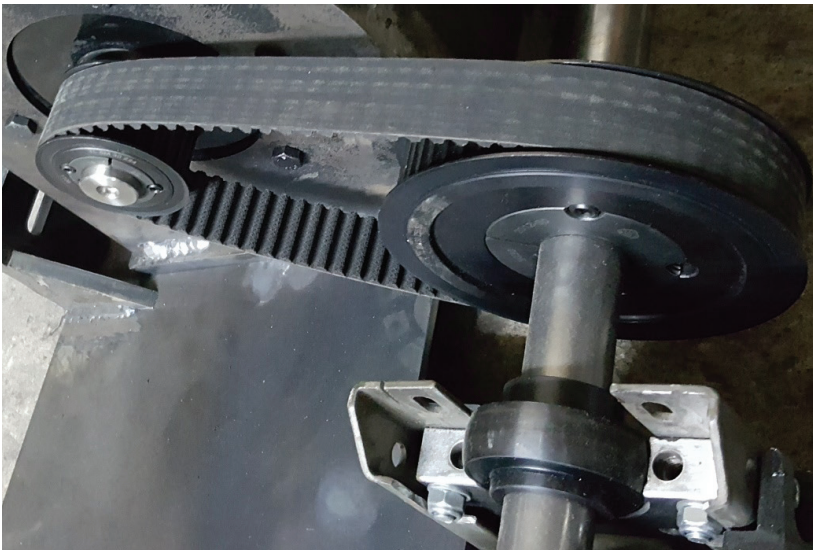


Fig. 5. Bearing seated in the bearing carrier and gear wheels mounted by Taper Lock fastening sleeve

Discussion and conclusions

For the first case, the assumption concerning the calculation of a full shaft replacing a pipe cross-section shaft showed a significant similarity of the results of both solutions. Based on the analytical analysis, we determined the correctness of the application of a given assumption. The first load case showed the possibility of using a shaft with a diameter of 17 mm for both analyzed elements. Both shafts met the strength assumptions, and the calculated strains and stresses were significantly below the safety limit of both elements. The deformations caused by the assumed loads during the given analysis turned out to be acceptable for the analyzed structure of the drive system.

For the second case, the analysis showed the necessity to use a full shaft with a minimum diameter of 24 mm. The considered load case showed that the second design solution assuming the use of a pipe cross-section shaft cannot be applied. Due to the conducted analyzes, the maximum deflection angles and displacements of both solutions have been determined, based on which we can unequivocally state that the pipe cross-section shaft shows too much slenderness and would be damaged under the conditions such as in the second case under consideration. The recommendation for the analyzed structures is to use a pipe cross-section shaft with an outer diameter of 50 mm and a wall thickness of 3.5 mm, for which the cross-sectional area is 25.5 mm². This solution will allow for much smaller displacements and deformations of the structure, maintaining the lightness of the element and keeping the maximum loads below critical values.

Racing go-karts use shafts with an outer diameter of 50 mm and a wall thickness of 2 mm. Thinner wall thickness reduces the weight, which results in better track times. The use of a smaller number of a factor of safety (in our case 2) may show a different result. Accurate calculations, the precision of technological shaft manufacturing, and knowledge of the experimental data characterizing the shaft load in real conditions may allow reducing the factor of safety number.

Full shaft, 30 mm thin, which was manufactured and assembly in prototype electric go-kart was chosen correctly. Drive train allowed for a ride without any problems on the in the straights, chicanes, harpin turns. The next step will be related to taking measurements during driving an electric go-kart and comparing the obtained results with numerical calculations.

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