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White, G., Cunningham, G., & Doyle, D. (2019). Design of an Electric Drive Transmission for a Formula Student Race Car. In *Advances and Current Practices in Mobility: SAE International Conference Proceedings: WCX SAE World Congress Experience 09/04/2019 → 11/04/2019 Detroit, United States* (SAE Technical Papers). SAE International. <https://doi.org/10.4271/2019-01-1295>

Published in:

Advances and Current Practices in Mobility: SAE International Conference Proceedings: WCX SAE World Congress Experience 09/04/2019 → 11/04/2019 Detroit, United States

Document Version:

Peer reviewed version

Queen's University Belfast - Research Portal:

[Link to publication record in Queen's University Belfast Research Portal](#)

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Design of an Electric Drive Transmission for a Formula Student Race Car

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Abstract

This paper presents a methodology used to configure an electric drive system for a Formula Student car and the detailed design of a transmission for in-hub motor placement. Various options for the size, number and placement of electric motors were considered and a systematic process was undertaken to determine the optimum configuration and type of motor required. The final configuration selected had four 38 kW in-hub motors connected through a 14.8:1 reduction transmission to 10" wheels. Preliminary design of the transmission indicated that the overall gear ratio would be best achieved with a two-stage reduction, and in this work an offset primary spur stage coupled to a planetary second stage was chosen. Detailed design and validation of the transmission was conducted in Ricardo SABR and GEAR, using a duty cycle derived from an existing internal combustion Formula Student car. The analysis was conducted in line with ISO 6336 and permitted the examination of the stresses in gear teeth and the prediction of gear and bearing life. A detailed design was proposed with due regard to ease of manufacture and assembly, and a full-scale prototype was manufactured to facilitate physical validation of the design. The design analysis showed all gears and bearings had a suitable predicted lifetime with a minimum factor of safety of 1.8 on gear wear.

Introduction

Formula Student (FS) is an educational motorsport competition held every year at Silverstone race circuit in the United Kingdom (UK). In recent years the competition has seen an increasing number of electric vehicle (EV) entries which have been hugely successful. This has mainly been driven by the worldwide demand for consumer EVs and the massive reduction in the cost of the technology. Queen's Formula Racing (QFR) are a FS team from Queen's University Belfast who have competed with an internal combustion engine (ICE) vehicle for almost 20 years. After reviewing the performance advantages of an EV, the decision was made to begin a 3-year development cycle for an EV leading to a first entry in the 2020 competition.

Many teams around the world have, or are, making this move to electric drive and overcoming the technical challenges associated with this. Some of the students working on these vehicles have featured in previous SAE publications and have presented papers on topics such as "Design and Optimization of Planetary Gearbox for a Formula Student Vehicle" [1]. These solutions presented to the problem of designing a transmission often feature bespoke machined gears which can be very costly. This paper presents an alternative

approach, where the transmission is designed around as many off-the-shelf components as possible to allow much smaller budget teams to enter EVs.

Another feature of this report is the high gear ratio of the design. Many FS transmission designs are limited to a 10:1 ratio as this is the highest ratio geometrically achievable with a planetary gear set [2]. This report shows how higher GRs can be achieved by using a two-stage reduction which allows lower torque, smaller motors to be utilized, saving mass and package space.

Conceptual designs for the vehicle layout, and most importantly the motor placement, were evaluated on key performance attributes. A duty cycle was then extrapolated from ICE data before a 5-year fatigue simulation was carried out on the proposed transmission in line with ISO 6336. A detailed CAD model was produced, and a working prototype manufactured.

System Specification

Before undertaking a detailed design of the transmission, the drive line architecture was evaluated with an appropriate motor(s) selected.

Vehicle/System Architecture

Torque vectoring allows for control of the vehicles yaw response by controlling the amount of torque delivered to either side of the vehicle at any given time [3]. This perfectly complements an electric powertrain in which separately controlled power sources (motors) can be distributed throughout the vehicle with only a small mass and efficiency penalty. Although greater performance may have initially been achievable by using a single motor and a mechanical limited slip differential, this would hinder the development of the QFR electric car long term, as QFR would not be able to develop and test torque vectoring algorithms. Four concepts were proposed, shown in Figure 1.

Max. Speed	20,000 rev/min
Volumetric Size	0.98 l

Optimal Gear Ratio

From a very basic tire model, and by considering the amount of torque required to slip the wheel under peak acceleration, a nominal gear ratio (GR) can be determined. This GR will be optimized for peak acceleration and will make the car traction limited. Using an estimated vehicle mass and a tire friction coefficient of 1.4 [4], a GR of 15.5 was calculated. This target GR was updated near completion of the project to 14.5 after a more extensive vehicle simulation model was completed by a current project [5].

Evaluation of Transmission Options

Given an outboard upright mounted motor, the transmission had to be packaged within a small volume. This perfectly suits a planetary transmission as the sun tooth loads are shared amongst the planets meaning the gear teeth can be much smaller. A single stage planetary style transmission, however, has a maximum GR of 10:1 due to geometric constraints, therefore if it is to be utilized it must be used in conjunction with another reduction stage. Many configurations were investigated, and the best options were:

- Compound planetary
- Double planetary
- Single planetary with Initial Reduction
- 3 stage compound gear train

The options were evaluated based on (in order of importance) complexity/cost, efficiency, reliability, mass and size. After reviewing these factors, the more common solutions of using either compound or a double planetary gearbox were deemed to be too costly due to the difficulty in ensuring correct mesh timing and the increased number of parts respectively. A single planetary stage with an initial reduction was identified as the most suitable for the application. A simplified version of this concept is shown in Figure 2.

Figure 2. Final transmission configuration - spur gear stage followed by a planetary stage

Using a planetary stage transmission presents two further options for the output drive. As a reduction gear ratio is required, the input must be the sun gear. The output however, can be either the planets or the annulus. Concept designs for both were produced, with detailed models for the upright, hub and brakes included to ensure that they were manufacturable.

Figure 1. Concept powertrain layouts

To evaluate the different concepts a spreadsheet tool was developed to estimate the vehicle mass, unsprung mass, yaw inertia, rotational inertia and mechanical efficiency of each concept. A model of the previous ICE car was used as the benchmark.

The different concepts were then evaluated through a Pugh matrix on several performance, design and manufacturing criteria. The Pugh matrix highlighted concept four as the best layout for the vehicle due to its superior mass, efficiency, manufacturing and cost from having the transmission inside the wheel rim for each wheel. The mass and efficiency estimates are listed in Table 1. Based on this analysis the decision was made to proceed with upright mounted motors for each wheel with a rear wheel drive only car being developed initially.

Table 1. Concept powertrain evaluation

Concept Powertrain	Vehicle Mass (kg)	Mechanical Efficiency (%)
- IC Engine	294.0	88.1
1. Inboard Rear	300.8	86.4
2. Outboard Front, Inboard Rear	305.6	88.4
3. Inboard Front, Outboard Rear	304.8	88.4
4. Outboard Front, Outboard Rear	301.1	90.3

Motor Selection

After extensive research, it was found that motors suitable for FS were difficult to source. For motors to be appropriate for FS they need to be packaged small, extremely lightweight and have both a good output torque (to reduce the need for a large transmission) and a high-power rating. Several potential motors were sourced and evaluated against each other with the FS kit, of motors, inverters and controllers, provided by AMK being selected due to their outstanding specifications, as listed in Table 2. This report hence focuses on the design of an electric drive system around this AMK DD5-14 motor.

Table 2. AMK DD5-14 motor specification

Power	38 kW
Torque	20 Nm

Figure 3 shows the difference between an external and internal hub design. It's shown that an external hub design favors the annulus as the output, while the internal hub favors the planets as the output. An internal hub design with the planets used as the output was selected as it packaged much more favorably allowing standard rims, brakes and bearings to be used which further reduced the cost and complexity.

Figure 3. Comparison of an external hub and an internal hub design

Transmission Analysis

Lewis Formula

To validate a transmission for its intended design life some level of simulation must be performed. In its most basic form, hand calculations can be performed with life and wear factors using equations such as the Lewis formula, Equation 1 [6].

$$\sigma = \frac{K_v \cdot F_t}{b_a \cdot m_G \cdot Y} \quad (1)$$

Where

$$F_t = \frac{T \cdot 2}{PCD} \quad (2)$$

$$m_G = \frac{PCD}{No.of\ teeth\ on\ gear} \quad (3)$$

Also, for cut or milled gears:

$$K_v = \frac{6.1+v}{6.1} \quad (4)$$

And for hobbed/shaped gears:

$$K_v = \frac{3.56+\sqrt{V}}{3.56} \quad (5)$$

Initially these formulae were used to develop a basic spreadsheet model of the transmission which allowed the gear tooth stress to be approximated given some geometric input such as GRs, tooth

numbers and modules. This was very useful for performing a quick feasibility analysis of any concept.

SABR and GEAR Software

Although the Lewis formula provides values for tooth stress which is useful for selecting materials and rough sizing, due to the complexity of tooth wear, this method would have to be used with a large safety factor which would significantly increase the size and mass of the transmission. To improve the accuracy of the results SABR and GEAR transmission analysis software packages, developed by Ricardo for automotive transmission design were used. This allowed for much more comprehensive analysis in accordance with ISO 6336.

SABR is focused on shaft and bearing analysis and has an extensive library of bearings from suppliers such as SKF, Timken and KOYO. Detailed shafts can be drawn with features such as grooves, bores and keys. A duty cycle (Power Flow) can then be imported which is categorized into specified torque, speed and duration bands to allow shaft loads, bearing loads and deflections to be computed. This is then used to calculate the predicted life of all bearings alongside shaft stress.

The duty cycle, along with shaft deflection and gear misalignment, can be exported to GEAR to allow each gear set to be analyzed individually. GEAR uses both a Ricardo developed wear model and ISO 6336 guidelines to calculate the tooth bending stress and tooth wear for each gear. GEAR also allows extensive modification of standard gear profiles and features an optimize function to decrease stress and increase tooth life.

Transmission Duty Cycle

A life cycle of 1,000 km was selected given that the transmission is designed for a 5-year service life. The fastest sprint lap for the ICE car was extracted from the 2017 event data and analyzed. Although the ICE car data will not be completely representative of the EV data, for example due to the EV having different performance characteristics, it was assumed that the driver demand for power would be very similar between both the EV and ICE car. From this data, the statistics shown in Table 3 were produced.

Table 3. Extrapolated transmission duty cycle

Design Service Life	1,000 km
	21.99 hours
% time that torque is demanded (non-zero)	53.43 %
Number of cycles between positive and negative torque	29,680
Average rpm	9,260 rpm

The torque demand and vehicle speed were investigated further to understand typical loading conditions. It was found that during the time that the driver requested power (when the throttle pedal was over 5% activated) the driver wanted full power 35% of the time. The full distribution of demand is shown in Figure 4. Similarly, the vehicle speed was analyzed and converted (assuming a GR of 15.5) for the AMK DD5-14 electric motor speed. Given the calculated life of almost 22 hours, Figure 5 shows the distribution of time over the motor's speed range. The distribution is positively skewed with an average motor speed of 9,260 rev/min. It is predicted that the motor

would spend only 4.93% of its service life at speeds over 15,000 rev/min where transmission wear is highest.

Figure 4 - % time distribution of throttle position over 5%

Figure 5. Distribution of time spent across motor speed range over full 5-year life cycle.

The instantaneous torque demand and speed data for the sprint lap was then categorized into the amount of time spent at various speed and torque bands and imported as a Power Flow duty cycle into SABR and GEAR.

Transmission Optimization

The first step in optimizing the transmission was deciding on how the GR should be split between the spur gear stage and the planetary stage. The spreadsheet model which was developed using the Lewis formula, as mentioned previously, was used to perform a sensitivity study to determine how changes to key variables such as the GR, module, materials and manufacturing method effect gear stress and gearbox dimensions.

Using this study, it was decided that the transmission should be optimized in the following way. The initial GR should be optimized so that the pinion gear and sun gear tooth stress is almost equal (these are the two most heavily loaded gears). The smallest module should then be selected which gave appropriate tooth fatigue life. To supplement this spreadsheet model, GEAR was used to provide more detailed analysis. The transmission design was iterated between these

two different models to define the geometry (spreadsheet) and then ensure sufficient transmission life (GEAR). Figure 6 shows the SABR model of the transmission which was exported to GEAR.

Figure 6. SABR model of the transmission.

Another factor to consider is addendum modification, and how it can be used to improve gear meshing and reduce gear stress. Although this is standard practice, it requires gears to be custom machined which has a sizeable cost associated with it. With one of the aims of the project being to propose a cost-effective solution it was decided to use off-the-shelf gears available from a standard parts supplier. To ensure correct meshing of the teeth using standard gear profiles the minimum number of teeth per gear was set to 25.

The spreadsheet goal seek function was used to identify the respective GR's which equated the stress in the pinion and sun gear. Two other factors to consider were what teeth number of gears were available from the supplier, and if mating gears were common factors of each other. To reduce gear fatigue, it is good practice to ensure that any two corresponding teeth do not frequently mate with each other as a small imperfection on one tooth can increase wear if it is always contacting the same tooth. To optimize the gear module GEAR was used to evaluate gear stress and fatigue life. The geometric information generated in the spreadsheet model was input along with material data from the gear supplier and a commercial materials database. From research of similar designs, a face width of 12 mm was chosen as it was shown that for values greater than this, small tooth misalignment can cause excessive tooth wear [7]. The design was then iterated several times before a module of 1.5 was selected. A GR of 14.82 was achieved given an updated target GR of 14.5 [5]. A breakdown of the gear geometry is given in Table 4

Table 4. Gears selected with material specification

Stage	GR	Gear	Material	# of Teeth
Initial Reduction	3.84	Pinion	EN36	25
		Wheel	EN36	96
Planetary	3.86	Sun	EN36	28
		Planet	EN36	26
		Annulus	EN36	80

Bearing Selection

The transmission bearing loads were calculated using free body diagrams of the torque reaction from the gears using the maximum motor output torque of 21 Nm. To reduce stress in, and improve the life of the transmission, the wheel hub and the transmission were axially isolated by using a sliding polygon shaft connection. To support the transmission shaft axially a combination of deep groove ball (DGB) bearings and needle roller/angular contact ball bearings

(NR/ACBB) were chosen. NR/ACBB were required for the transmission shaft and the planets as these bearings can handle much higher radial loads than DGB yet still provide axial support.

Bearing life was then validated in SABR using the previously developed duty cycle.

Design for Manufacture

Having validated the gear stress, bearing loads and transmission life in SABR and GEAR a detailed CAD model was developed shown in Figure 7. This included concept designs for the upright and hub to ensure manufacturability and assembly of the transmission. The transmission shaft, shown in green in Figure 8, was modelled first as this determined the position of both gear stages.

To ensure bearings were positioned and held correctly throughout the transmission, abutments and bearing spacers were added to the shafts in accordance with the manufacturer's recommendations. All the bearings were then clamped against these abutments using screwed bearing clamps with fixings specified from a self-developed bolt stress calculator. Similarly, hand calculations were performed on both transmission shaft keys, drawn in accordance to BS 4235-1:1972, with appropriate materials defined to handle the shear stress. BS 3673-4:1977 was used to select standard circlips for holding the gears and bearings in place where an alternative means of support could not be used.

Figure 7. Isometric view of final transmission design.

Figure 8. Section view of final transmission design.

Prototype Manufacture

To validate the design and allow visualization of the transmission, a prototype was commissioned. The gears supplier also sold Nylon versions of their gears for low load applications. These are less expensive than the steel alternatives but have identical geometry. To reduce the cost of the prototype further, all steel components were instead made from Aluminum and where suitable, Acrylic and MDF sheet was laser cut. Tolerances were relaxed on many components to reduce the prototype cost with the specified bearings purchased to verify assembly and clearances on the shafts. The prototype also features a crank handle on the back to allow the transmission to be turned (Figure 9).

Figure 9. Picture of prototype transmission

Results

The system architecture chosen was for four outboard upright mounted motors. This configuration was lightweight, had a superior system efficiency and fewer transmission parts than the other concepts. The AMK FS motor package was selected as they are extremely high in performance and widely regarded by FS teams as the best motor package available. A transmission duty cycle was then extrapolated using ICE car data.

A key aim of the project was to deliver a cost-effective solution and so off-the-shelf gears were chosen over bespoke gears. A solution was found that could be packaged within a 10-inch wheel rim using standard gears. This transmission was then analyzed using the SABR and GEAR software provided by Ricardo. The minimum safety factor for any gear was 1.80 with the full results reported in Table 5. The predicted lifetime of all bearings exceeded the 22-hour requirement with a considerable safety factor to account for unknowns such as lubrication, contamination and manufacturing quality. The full bearing lifetimes are reported in Table 6.

Table 5: ISO 6336: Load capacity safety factors

Stage	Gear	Bending Safety Factor	Contact Safety Factor
Initial Reduction	Pinion	4.55	1.80
	Wheel	6.00	2.11
Planetary	Sun	3.57	1.85
	Planet	2.64	1.89
	Annulus	4.90	2.97

Table 6. Selected transmission bearings

Use	Product Name	Hours to Failure
Central Transmission Shaft	SKF NKIB 5902	1649

End of Transmission Shaft	SKF 61900	7542
Planets	SKF NKIB 5901	1406

A detailed design solution was then proposed for the transmission shaft with concepts presented for the hub, upright and brakes (Figure 10). A prototype transmission was manufactured which verified manufacturability and provided a prototype for future development.

A dynamic analysis such as an analysis of the cyclic bending load or the shaft's natural frequencies was not performed however. This dynamic analysis can only truly be assessed when all the rotating components have been completely defined as the natural frequency is mass dependent. Another assumption made was that the transmission casing (upright) was infinitely stiff. In reality, it will deform under load which may increase tooth wear from tooth misalignment.

Other factors that have not been considered are the effects of the external environment on the components such as vibration, corrosion or contamination. These too could significantly reduce transmission life if they are not evaluated and managed closely.

Figure 10. Isometric view of packaged rear left transmission with cut away to show proposals for the hub and upright.

Summary/Conclusions

1. The AMK FS kit featuring four DD5-14 motors and inverters was selected as the highest performance, most cost-effective package.
2. A detailed electric drive system has been proposed with four upright mounted motors being selected as the optimum powertrain configuration.
3. Transmission concepts were evaluated, with an initial reduction and planetary stage configuration being selected as the best solution. A GR of 14.82 was achieved using standard gear sizes (nominal GR 14.50).
4. Detailed analysis was performed on the transmission suitable for a motor with 21 Nm of peak torque and 38 kW of peak power. All gears and bearings had suitable predicted

lifetimes with a minimum factor of safety of 1.8 on gear wear.

5. Bearing, shaft, key and fixing loads were calculated. Bearings were selected appropriately, with hand calculations performed on shaft, keys and fixings to ensure suitability.
6. Concept designs were presented for the hubs, uprights and brakes which verify the viability of the transmission design.

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Acknowledgments

The author would like to thank Dr Geoff Cunningham for his guidance and feedback over the past two years. Dr Cunningham has run QFR since its inception in 1999 giving myself, and countless other students an environment to develop our engineering and professional skills. Thanks also to Darryl Doyle and Connor McShane, the other two final year students working on the QFR electric car, who have facilitated long and fruitful technical discussions adding to the critical thought behind the report. Finally, a special thanks to Dr Stephen Glover and Ricardo engineers Paul Etheridge, Tomas Zemek and Jindrich Hruska for their technical support and software sponsorship for the project.

Definitions/Abbreviations

FS	Formula Student
UK	United Kingdom
EV	Electric vehicle
QFR	Queen's Formula Racing
ICE	Internal combustion engine
GR	Gear ratio
DGB	Deep groove ball
NR/ACBB	Needle roller/angular contact ball bearings