Ultra-Lightweight Design of a Single Speed EV Transmission

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ABSTRACT

Reducing the mass of the electric drive-train directly improves the range and performance of an electric vehicle.

Substitutionary use of lower density materials may marginally improve the mass of a transmission but a full system approach is required to yield savings greater than 30% compared to conventional designs. In order to make large mass savings the damping, insulation, and processing qualities of alternative materials have to be maximised.

Overcoming the challenges of using composites in mainstream automotive use is possible if the system benefits are well understood. In this way the increased material costs can be traded against savings elsewhere in the design. One example is the inherent damping properties of the materials allow the efficiency of the drive train to be prioritised over NVH. This is a double win for the gearbox manufacturer.

A full system model of a single speed gearbox suitable for A/B segment vehicles is used to develop the design. The model calculates durability, efficiency and NVH performance simultaneously. Using the model to predict housing panel accelerations allows us to demonstrate how modifications to the gear geometry - previously introduced to lower excitation from gears - can be reversed in favour of lower mesh losses, reducing component cost, and complexity. It is these trade-offs that pay towards the higher costs of more exotic materials. This same process can be used to optimise the durability of the transmission leading to lower rotational inertia, less steel, and compounded mass and cost savings.

This paper presents what is possible for a single speed EV transmission. It shows the trade-off between NVH, efficiency and durability and their effect on the overall power train mass and cost. It also shows savings achievable by fully integrating an electric machine into the transmission. Materials and processes that make these technologies suitable for automotive applications in the near future have been chosen and utilised in the design.

INTRODUCTION

Since the turn of the century, average new passenger vehicle CO_2 emissions have decreased noticeably driven by the introduction of new legislation and penalties for manufacturers failing to meet these targets. Advances in engine combustion technology, the introduction of hybridised powertrains, and reductions in rolling resistance and aerodynamic drag have all contributed to this reduction. However, some technologies have bought with them significant weight penalties, such as the addition of the electrical architecture required for hybridisation.

Average new passenger car CO_2 emissions in Europe have reduced from 169g/km in 2001 to 123g/km in 2015, however average vehicle mass has increased from 1270kg to 1385kg over the same time period. If vehicle mass had stayed constant the average CO_2 emission in 2015 would likely have been close to 115g/km.

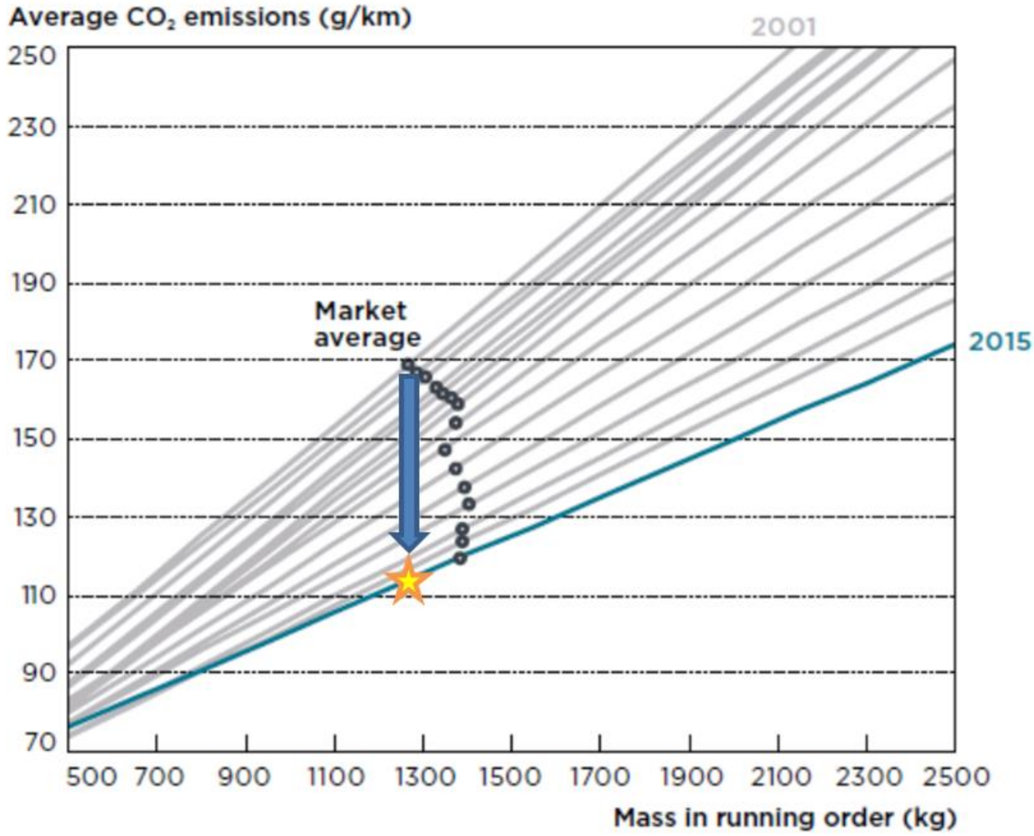

Figure 1 - Sales-weighted correlation CO_2 emissions and vehicle mass [1]

With the recent increase in popularity of electric vehicle in the passenger sector, the issue of vehicle weight manifests itself in a different way. Electric vehicles do not have a comparable measure to CO_2 emissions, and are rarely compared using a measure of energy consumption per unit of distance travelled. The primary comparator between electric passenger vehicles is range. The vehicle range, along with cost, is also one of the most influential parameters for customers when considering the purchase of an electric vehicle [2]. Range can be increased by increasing the amount of batteries in the vehicle, however this increases both the mass and the cost of the vehicle and actually increases the energy consumption per unit of distance travelled.

Reducing vehicle mass is therefore one the key areas of focus for automotive manufacturers and suppliers for both internal combustion and electrically powered vehicles. Reducing the mass of an electric vehicle can help to increase the available range from a specific battery module size, or even to reduce the battery size and mass whilst maintaining vehicle range. Whilst battery technology continues to progress in terms of increased power density and reduced specific cost, forecasts suggest that this alone will not be sufficient to generate the consumer uptake required to meet legislative objectives. The rest of the vehicle must therefore contribute to meeting these targets, including the transmission.

CASE STUDY TRANSMISSION

For the purposes of this study a typical single speed transmission for a B-segment passenger vehicle has been assumed. The transmission is driven by a 100kW, 220Nm electric machine and consists of a two stage speed reduction with an overall ratio of 11.8:1 and includes a differential for distribution of the power to two wheels. It is of conventional construction with two part die-cast aluminium casing, hobbed and ground gears manufactured from 20MnCr5 and case hardened, cast iron differential casing with bolted final drive joint and 2-pin bevel differential. A passive lubrication regime distributes of the lubricant and dissipates heat.

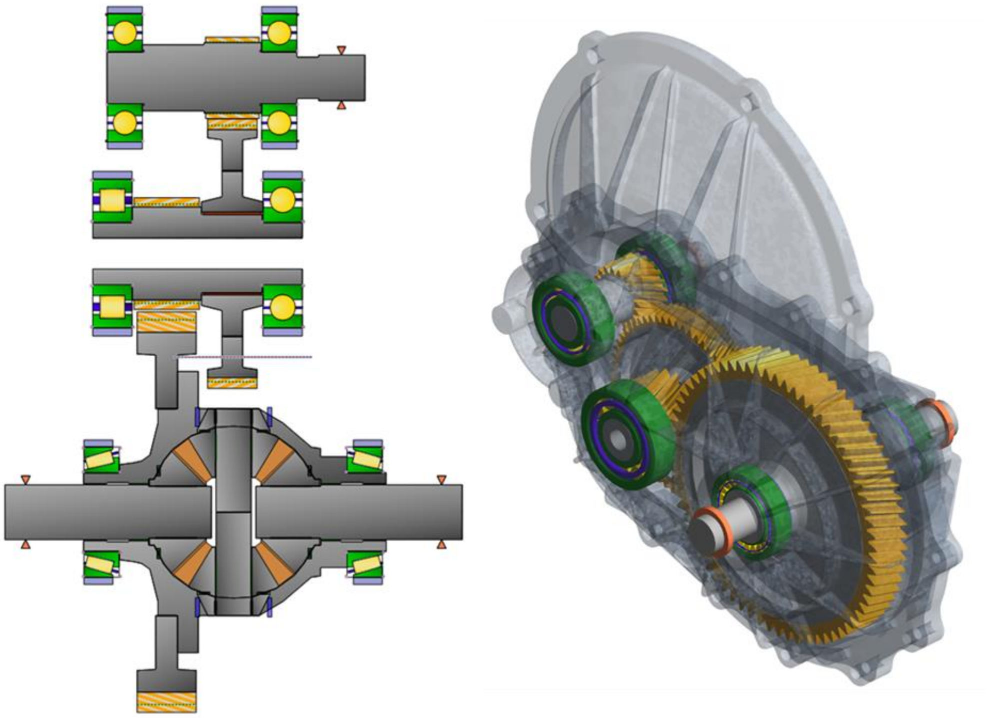

Figure 2 – Baseline transmission cross-section & 3D view

A full system analysis model has been used for the assessment of the transmission durability, efficiency and NVH performance. The analysis model includes shaft stiffness representations, non-linear bearing stiffness models, and FE based differential and casing stiffness models.

A duty cycle consisting of multiple load cases with defined output torque, output speed, and duration is applied to the analysis model and the loading of components is calculated based on the stiffness of the components. Additional external loads such as vehicle bump reactions can also be applied to the model. Durability analysis of gears and bearings has been conducted to recognised standards such as ISO 6336 and ISO/TS 16281, as well as contact stress analysis at the gear meshes and rolling element to raceway contacts. Casing loads, deflections, and resulting stresses may also be calculated.

NVH analysis has been performed through the calculation of gear mesh transmission error and subsequent application as an excitation and the system modal response assessed. Casing surface accelerations are calculated along with force functions at external interface points. Efficiency analysis has also been conducted using ISO/TS 14179 for gears, rolling element bearings, and seals. Power losses due to oil churning have also included based on estimated operating oil level.

The use of a single full system analysis model enables the rapid iteration and assessment of numerous design variants and the system performance in multiple areas to be analysed in a short time frame. It also allows the influence of modifying a single aspect of the design to be assessed on all other components within the system. This provides the ability to refine the design of the entire transmission to a much greater extent with a high level of confidence.

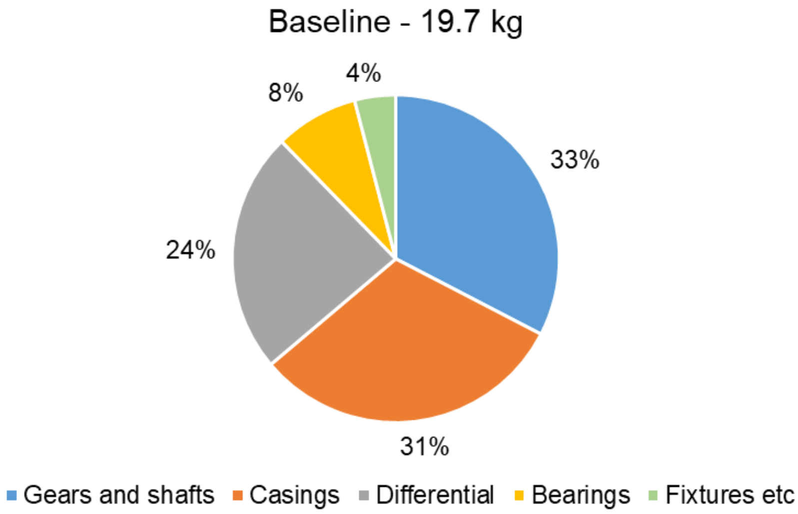

Figure 3 – Baseline transmission mass distribution

The transmission mass is quite competitive for the power output at 19.7kg. The gears and shafts represent the majority of the mass, closely followed by the casings, and just under a quarter of the total is due to the differential.

LIGHTWEIGHT DESIGN

ROTATING COMPONENTS

Modern automotive transmissions generally use some form of manganese chromium steel for the manufacture of gears, such as 20MnCr5, with additional heat treatment by case caburising. These can generally be classified in the MQ quality grade defined in ISO 6336-5. Material suppliers are increasingly offering engineering steels of higher grade and increased cleanliness in order to assist their customers in meeting ever more demanding targets for strength and durability. These materials also provide an opportunity to reduce the size of existing components whilst maintaining their current levels of durability.

The application of higher strength, higher quality gear steels can have a significant influence on the complete transmission design. Whilst the use of such materials comes at an increase in raw material cost it also enables a reduction in the amount of material required as it enables a decrease in the center distance of each of the gear sets. This not only reduces the diameter of the gears themselves but also reduces the amount of material required for the casing to enclose the gear train. Furthermore, the reduction in center distance enables a reduction in the angle between the two gear trains, required in the baseline design in order to achieve adequate gear durability within the package space defined. This reduces the amount of casing material required further still.

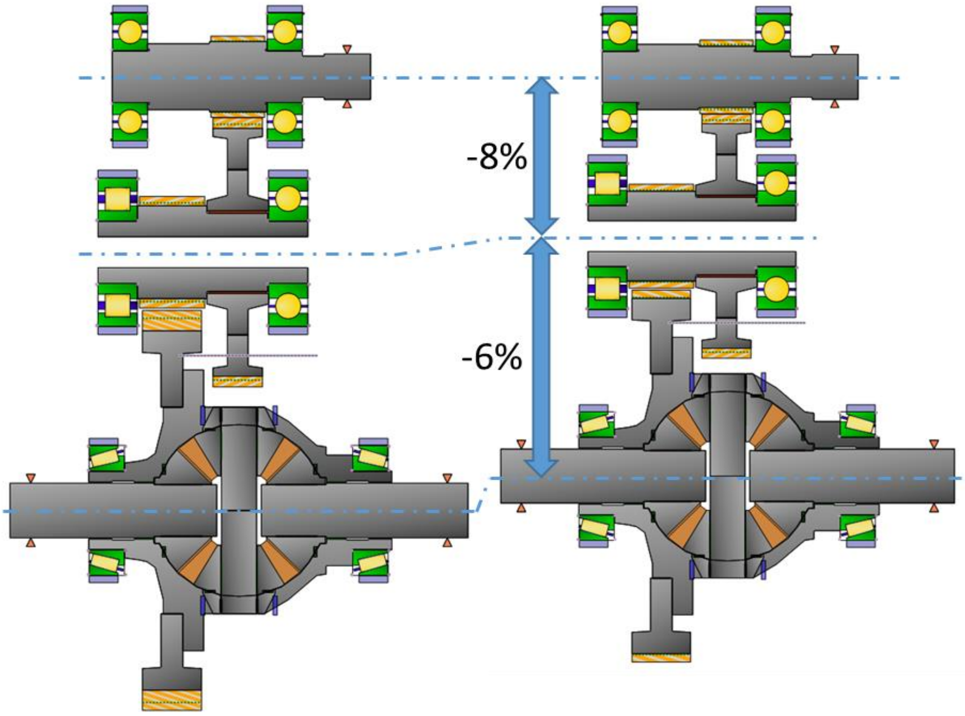

Figure 4 – Centre distance reduction through use of high grade steel

The reduction in gear center distance acts to increase the tangential load at the gear mesh due to the reduced pitch circle diameter. This in turn generates increased radial and axial separating loads at the gear mesh that must be reacted by the shaft support bearings. In addition, the change in intermediate shaft position influences the resultant load on the bearings supporting this shaft due to the change in relative angle between the working pressure angles of the two gear meshes acting upon the shaft.

In order to maintain durability of the shaft support bearings the gear geometry has been modified to adjust the balance of radial and axial load placed on the bearings. The bearing specification has also been revised in critical locations to utilise the latest range of high performance rolling element bearings available from European manufacturers. Such bearings are manufactured using highly homogeneous and clean steel, and have optimised internal geometry in order to provide increased load carrying capacity without additional weight, and only minimal additional cost.

The reduction in gear center distance also acts to reduce the tangential speed of the gears. This has a benefit for both the efficiency of the gear mesh itself, but also reduces the losses associated with any gears that rotate through the transmission lubricant. The gear mesh efficiency is improved by both the reduction in pitch line velocity and also the revised gear macro-geometry. The increased strength of the gear material enables a reduction in tooth size reducing the relative sliding velocity between the flanks of the two gears in contact.

The transmission employs a passive lubrication system to minimise complexity, cost, and power loss. The transmission relies on the gear rotation to distribute the lubricant to the various elements that require. As such it is not possible to completely remove the gear churning effects, however reducing the speed of the gears through the oil whilst maintaining the capability to distribute the lubricant can minimise the power loss associated.

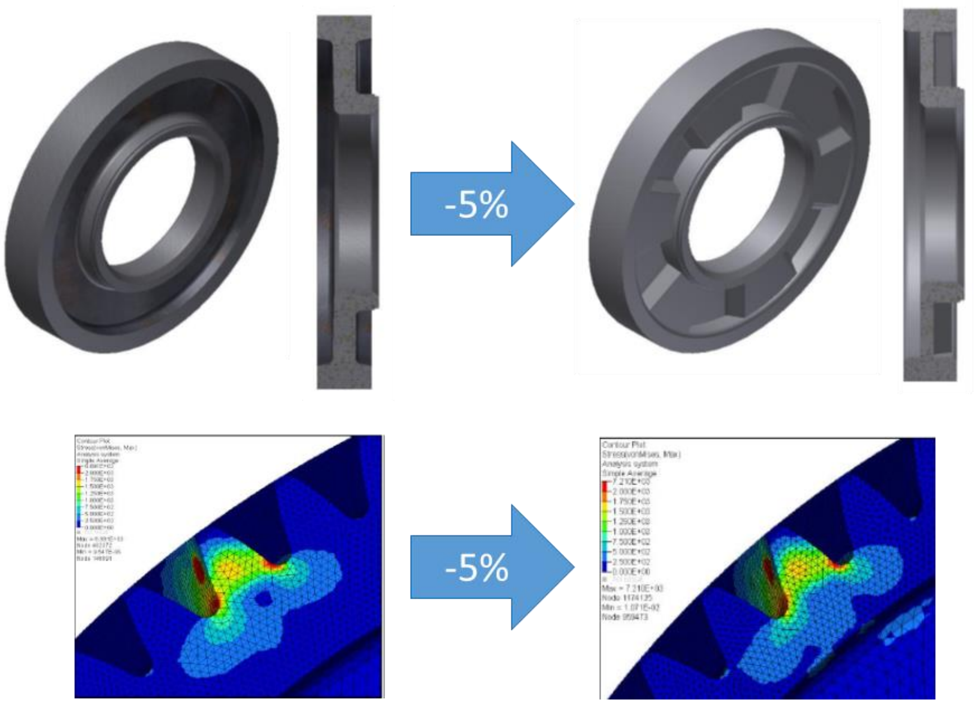

Figure 5 – Gear blank mass reduction

Detailed analysis of the gear blank and rim, coupled with the use of higher strength materials, also yields a further potential weight saving. By utilising a full FE contact model for the primary stage gear wheel rather than the more commonly employed rules of thumb, the thickness of the gear rim can be reduced without detriment to the gear tooth bending life. A rim thickness to tooth height ratio of as low as 0.6 can be achievable depending on the severity of the duty cycle, which can yield a component mass saving of up to 5%. Moving away from the conventional I-beam gear web design can also save a further 5% of component mass whilst maintaining stiffness and conventional manufacturing methods.

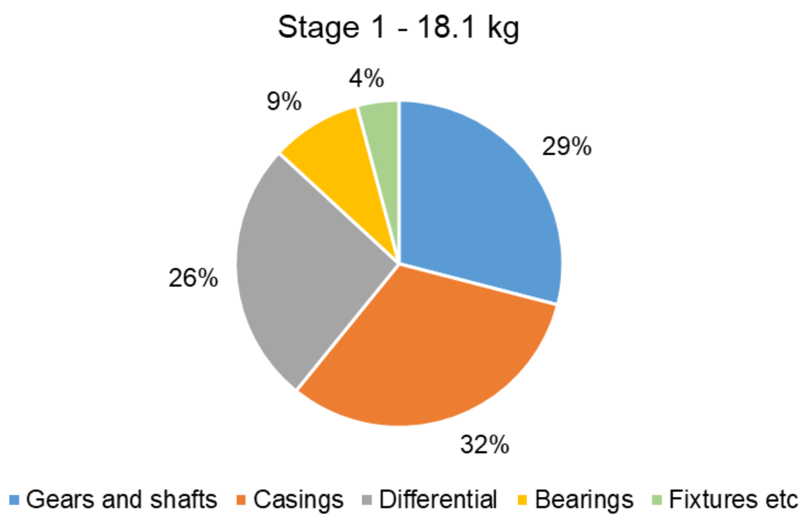

Figure 6 – Stage 1 transmission mass distribution

Through the application of higher grade gear steel and intelligent design of the mass of the rotating components has been reduced by 18%, a reduction in casing mass of 7% has also been achieved, plus a 6% reduction in the mass of the fixing required between the casings. In total the transmission mass has been reduced by 1.6 kg, equating to an 8% total reduction.

DIFFERENTIAL

The conventional bevel gear differential is a reliable and well-proven method of distributing the power from a single source to the two wheels of an axle of a vehicle and is commonplace in today's passenger vehicle market. Two differential pinions are most commonly used, mounted within a single piece differential cage constructed of cast iron. Symmetrical openings in opposite sides of the cage enable the bevel gears to be assembled into the differential with a single cross-pin inserted radially to support the planet bevel gears, whilst the side gears are generally located by the stub-axles or drive shafts inserted from either side of the differential. Greater torque capacity can be achieved by use of four planet bevel gears; however this then necessitates the use of two piece differential cage to facilitate assembly. The final drive wheel is normally mounted to the differential cage by use of a bolted connection, or may be welded in some applications.

Planetary differentials are far less common in passenger vehicles however they do present the potential to reduce transmission weight by better utilisation of the space within the final drive wheel. The implementation of a planetary differential as proposed by Schaeffler [3] not only reduces the mass of the differential assembly itself but also enables further weight savings through the redesign of the transmission casing.

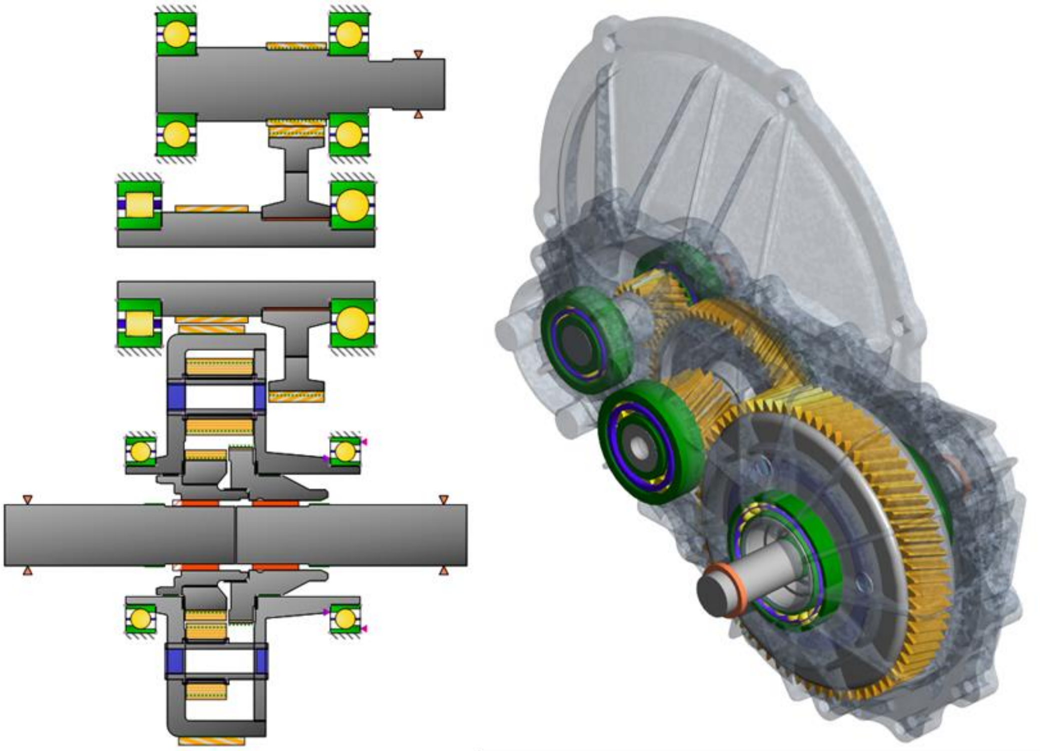

Figure 7 – Inclusion of a planetary type differential

The reduced span between the differential support bearings also enables the use of angular contact ball bearings for improved efficiency. Tapered roller bearings are often specified in this location due to the high load capacity required combined with the radial package restrictions created by the interface to the power source. For the transmission in question, the electrical windings of the motor restrict the radial space available for the differential bearing closest to the motor. The durability of angular contact ball bearings can be more sensitive to variations in preload resulting from the thermal expansion of the transmission casing however this is minimised by the reduced bearing span achieved by the planetary differential.

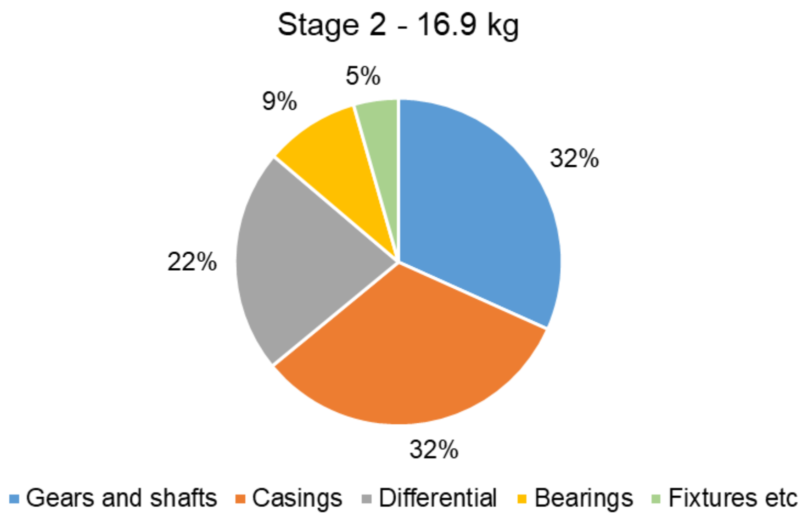

Figure 8 – Stage 2 transmission mass distribution

A differential of this form has been sized for this application with an appropriate factor of safety based on the maximum output torque of the transmission. This has achieved a mass saving of just under 1.0 kg from the differential, including the final drive wheel, equating to a 20% reduction from the baseline design. The differential support bearing selection has also been revised to use angular contact ball bearings saving another 0.04 kg (2%) and the casing has been redesigned to suit the new differential saving a further 0.29 kg from the stage 1 design, and a reduction of 11% from the baseline design. Application of the planetary type differential has reduced the transmission mass by 1.3 kg from the stage 1 design. Total weight saving from the baseline now stands at 2.9 kg (15%).

CASING

The vast majority of production passenger vehicle transmissions use die cast aluminium casings for the structural support of the rotating components. Depending on the architecture of the powertrain the transmission may also have pressed steel or injected molded plastic covers for certain areas, however these generally do not carry any of the internal or external loads experienced by the transmission.

A typical transmission casing assembly for a transverse powertrain arrangement consists of two casing joined by a bolted flange perpendicular to the axis of the rotating components. This flange must reacted the loads generated at the gear meshes within the transmission as well as the loads experienced due to vehicle bump or impact conditions. This causes the mass of the powertrain to accelerate and a force reaction to be generated at the powertrain mounts, typically at three points around the periphery of the powertrain.

It must also be noted that the transmission casings are required to fulfil a number of additional functions in addition to supporting the internal components. These include interfaces to various external components such as powertrain mounts, shift systems, electrical systems, and of course the power source. Retention and distribution of the transmission lubricant, exclusion of foreign contaminants, and dissipation of heat generated internally are also all critical. The casings can also be the primary path for unwanted noise generated at the gear meshes causing audible gear whine, an issue that is particularly relevant to electrical vehicles where the other traditional noise sources that previously masked low level transmission whine are no longer present.

Structural optimisation techniques can be employed to minimise the weight of the transmission casings through the identification of the most efficient locations in which to add structural features. The draw directions of each casing have a significant influence on the direction in which structure can be added in order to support the loads experienced by the casing, but also on the cost of the tooling required to cast the casings. It is therefore not always possible to place structural features such as ribs in the optimal location for supporting loads.

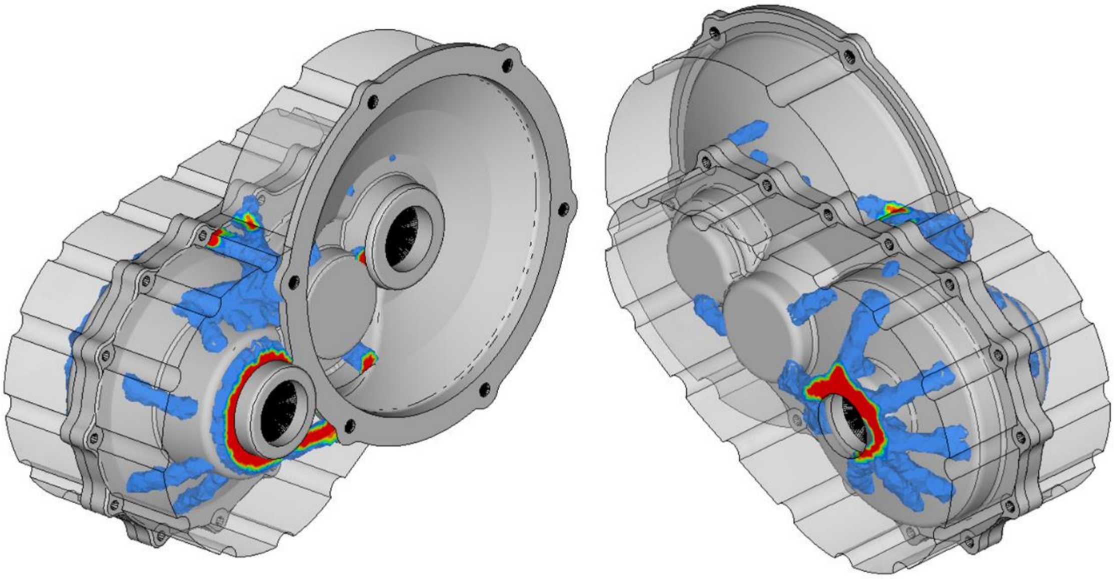

Figure 9 – Structural optimisation of a single speed EV transmission casing

Such optimisation can reduce the mass of a transmission casing assembly designed by conventional methods by up to 15%. This equates to a further 0.8 kg mass saving from our electric vehicle transmission. However, the casing must still fully enclose the gear train in order to retain lubricant and exclude contaminants, and thus has a continuous wall section between all shaft support bearings, seal location diameters, and the bolted flange. This wall also contributes to the structure of the casing although in many areas its sole function is to enclose the gear train. Manufacturing constraints generally present a lower limit to how thin this wall can be, limiting the amount of mass that can be removed from these areas of the transmission. Is there potentially an alternative approach to transmission casing design that could yield a greater mass saving?

The use of carbon composites in automotive applications is often proposed, but remains largely the preserve of prestige applications due to the cost. Previous research has shown that whilst it is possible to substitute an aluminium casing with a composite alternative it requires careful design and complex analysis. In order to achieve the equivalent strength and integrity provided by conventional aluminium casings, casings manufactured from composite require complex molded forms and over molding with structural plastics. Carbon composites can however offer significant weight savings because of their high specific strength, three to four times that of aluminium.

An alternative concept has therefore been developed that maintains the basic aluminium structure to provide the internal and external interfaces and react the radial separating loads, combined with carbon composite panels with a simple form, bonded to the aluminium structure to enclose the transmission and motor interface. The carbon composite panels also provide strength to the casing assembly by reinforcing the aluminium structure in torsion about the line between gear centers. This requires significant bonding strength between the aluminium and carbon composite to resist the shearing forces generated at the interface.

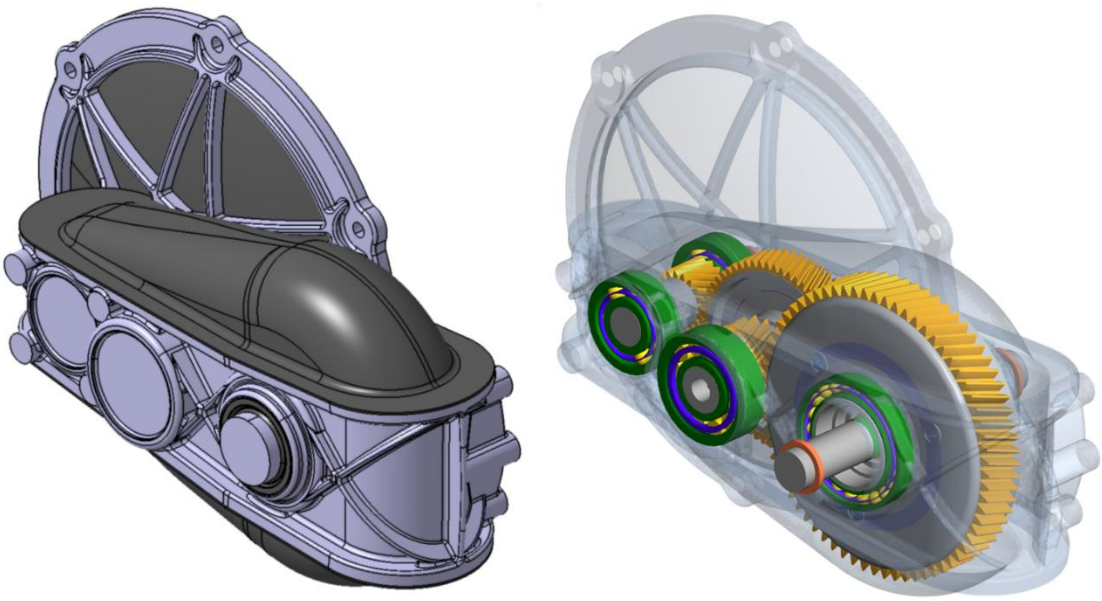

Figure 10 – Stage 3 casing design and analysis model

Inclusion of an FE model of the proposed casing design in the analysis model enables its influence on the durability, NVH, and efficiency of the transmission to be assessed. The change in casing stiffness influences the load sharing between components and also the misalignment of both the gear meshes and the rolling element bearings. This in turn impacts the component durability and also the gear mesh NVH performance. The use of carbon-composite in the casing also has an influence and introduces an additional dimension to transmission NVH performance as its stiffness and damping vary with temperature.

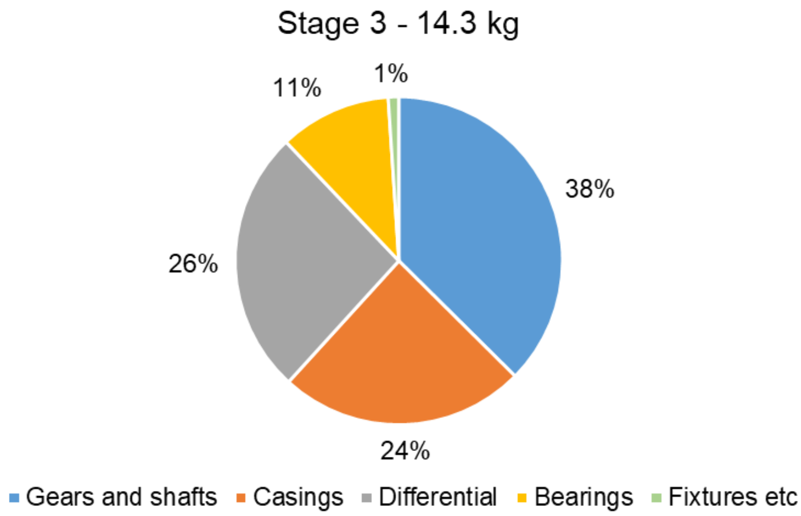

Figure 11 – Stage 3 transmission mass distribution

The introduction of the aluminium-composite casing achieves an impressive 36% reduction in mass over the stage 2 casing, more than double the saving achieved by structural optimisation. By bonding the carbon composite covers to the aluminium structure the majority of the fixings have also been eliminated saving a further 0.6 kg. The stage 3 design total mass has been reduced to 14.3 kg, a 27% reduction from the baseline design.

TRANSMISSION ANALYSIS

Whilst optimisation tools can be used to explore the design space for individual elements of the transmission, such as gear macro-geometry design, the complex interactions between the different elements of the transmission require the influence of numerous potential design changes to be considered together. This is especially true when aiming to extract the maximum performance in multiple respects from the various elements of the transmission.

An analysis model of the complete transmission system has been used to assess the influence of the additional technologies at each stage of the weight reduction process and ensure that the transmission continues to meet the required durability, efficiency, and NVH targets.

DURABILITY

Throughout the iteration of the design the durability of the gears and bearings have been assessed to ensure that the transmission continues to meet the required targets.

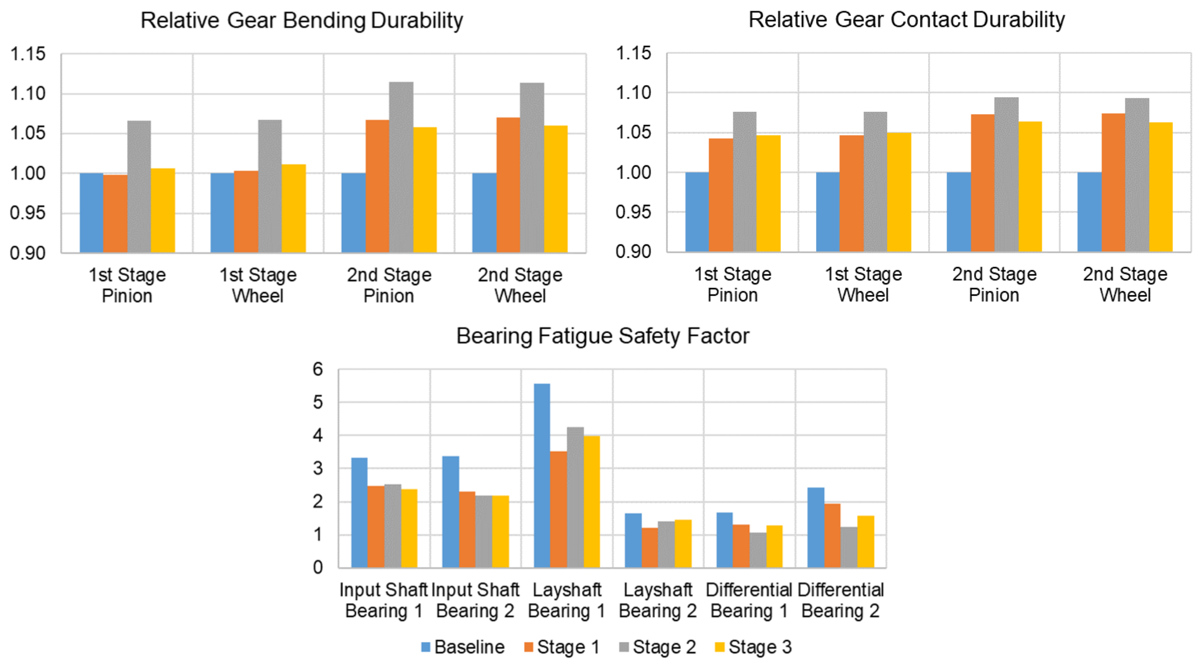

Figure 12 – Gear and bearing durability at each design stage

Gear durability has in fact been improved from the baseline. Bearing durability has reduced however this is indicative of the design changes that have been made and represents a more efficient use of the bearing capacity.

EFFICIENCY

Transmission power loss has also been analysed and considers the losses at the gear meshes, the rolling element bearings, and the oil churning losses.

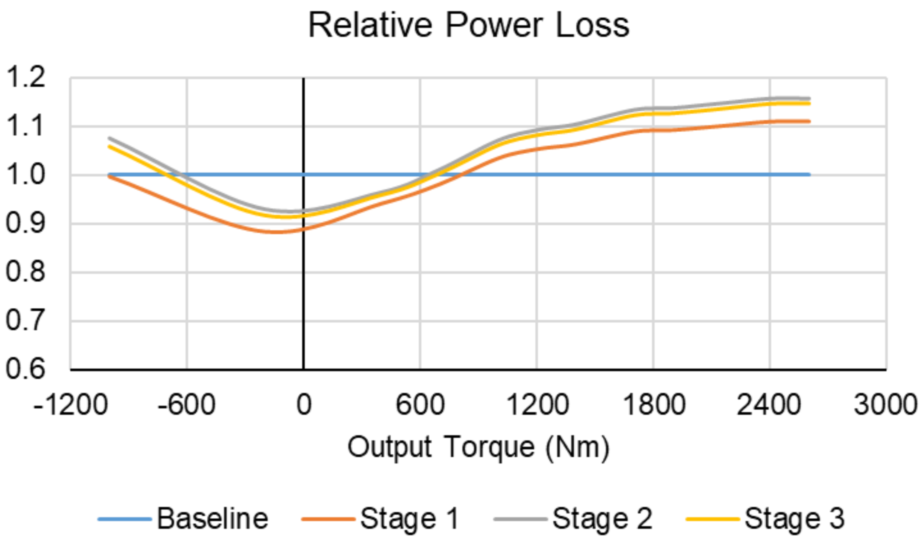

Figure 13 – Relative power loss at each design stage

The power loss of the transmission has been reduced in the low torque operating region with a slight increase observed at high torque levels, and area that is very influential on the published vehicle range.

NVH

Gear mesh misalignment and transmission error have also been analysed through the design process and have been maintained at very similar levels to the baseline design.

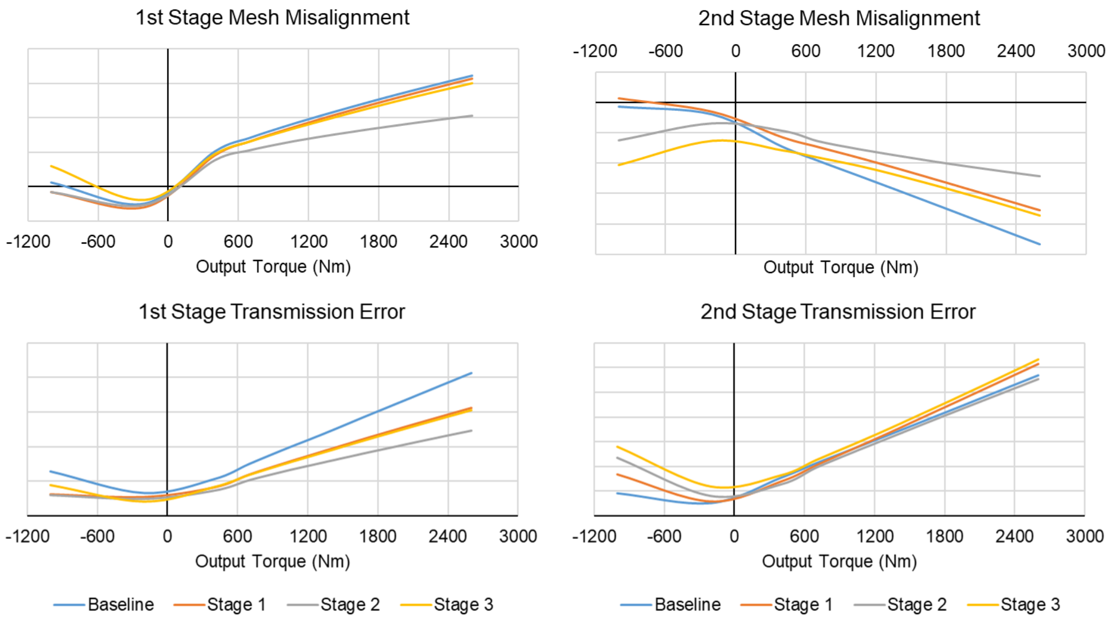

Figure 14 – Gear mesh misalignment and transmission error at each design stage

Further analysis of the casing response to the transmission error also suggests that the proposed casing design can achieve similar levels of NVH performance at ambient temperatures and actually reduces noise emissions at elevated temperatures due to the increase material damping as the matrix material approaches its glass transition temperature.

MASS

The main aim of the study was to achieve a significant reduction in transmission mass, with an ambitious target of 30% reduction from the baseline design.

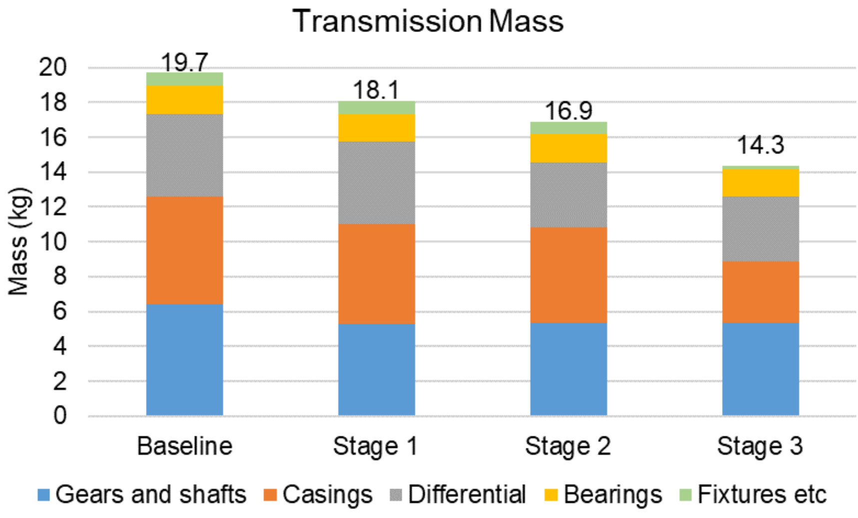

Figure 15 – Transmission mass distribution at each design stage

A total mass reduction of 5.6 kg has been achieved equating to a 27% saving.

COST

A basic costing model has been used to assess the impact of the various proposed design changes on the piece price of the transmission. In all cases, a production volume of 50,000 units per annum has been assumed. The mass of material and the manufacturing processes involved form the basis for the component cost calculation. An additional cost for assembly has also been included and modified dependent on the assembly complexity of the transmission design.

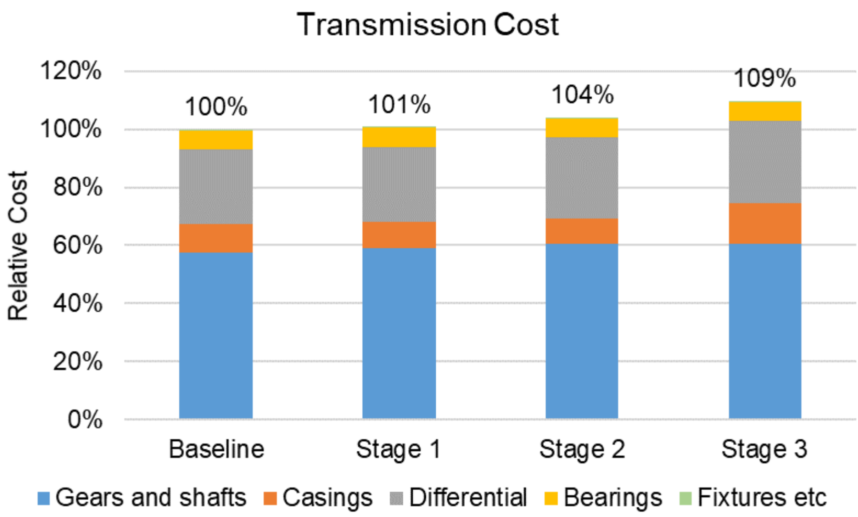

Figure 16 – Relative transmission cost at each design stage

Whilst a material cost increase of 25% has been assumed for the use of higher grade steel, the reduction in the mass of steel coupled with mass savings to the casing actually only translates to a 1% increase in the transmission piece cost. Similarly for the inclusion of the planetary differential, much of the additional cost associated with the more complex parts is actually offset in material savings in both the differential and casings resulting in 4% cost increase from the baseline design.

The cost of the carbon composite in the proposed casing design cannot however be offset by the reduction in aluminium casing due to the high material cost relative to the aluminium. Despite only comprising 13% of the casing mass the carbon composite elements account for nearly two thirds of the casing assembly cost. A 9% increase in transmission cost does however yield a 27% reduction in mass. With automotive electric vehicle transmissions costing in the region of €250 euros, depending on specification, and taking the 5.4 kg mass saving associated with these design proposals this equates to €4.17 per kilogram, below the €5 - €10 per kilogram generally regarded as the acceptable cost of lightweight design.

CONCLUSIONS

A number of new technologies have been applied to the design of a single speed electric vehicle transmission with the aim of achieving significant weight savings. Using a full system model approach to the analysis it has been shown to be possible to maintain durability, efficiency, and NVH targets whilst achieving an impressive mass reduction of 27%. The cost associated to this approach has also been considered and deemed as competitive with market requirements.

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