Weight Reduction Finite Element Study of Selected Power Train Components for Heavy Duty Vehicles

Submitted to FIERF and AISI

By

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FOREWORD

This document has been prepared by the Advanced Metal Forming and Tribology Laboratory in the Department of Mechanical and Aerospace Engineering at North Carolina State University. The research focus of this Lab includes manufacturing process modeling and optimization, triboscience and tribotechnology, tool design, and computational tools. In addition to conducting industry relevant engineering research, the Lab has the objectives a) establish close cooperation between industry and the university, b) train students, and c) transfer the research results to interested companies.

This report, entitled “Weight Reduction Finite Element Study of Selected Power Train Components for Heavy Duty Vehicles”, examines the potential for the reduction in the weight of several heavy duty truck components through the finite element method. The loading conditions on the parts are first established via a combination of surveys of individual parts and larger systems and mathematical analysis. The several lightweight designs of some selected components are then compared to the conventional part design to determine the potential weight savings that are possible if the lightweight techniques were adopted.

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Chapter 1

Introduction, Motivation, and Objectives

The main focus of the project is to reduce the weight of the components which make up class 7 and 8 trucks using innovative forging techniques, improved heat treatment techniques, material substitution, and geometric changes. In order to determine which parts have the highest potential for weight savings and to determine the potential lightweight techniques which should be employed to achieve these weight reductions one must have a clear understanding of the geometry of and loading on various truck parts. Therefore this report is focused on the numerical modeling of several heavy duty truck components and the results of weight reduction studies carried out using these models.

The study will focus on various shafts used to transmit torque in heavy duty trucks. Namely, the input and output shafts, the countershaft, and the fully floating axle shaft. In the future, the study will be extended to the other components, such as the connecting rod, crank shaft, and cam shaft. The loads on these shafts will be determined through a combination of surveys of similar parts and mathematical analysis of the load transfer from part to part. The numerical analysis will be carried out using the commercial finite element software ANSYS Workbench and will disregard dynamic loads and fatigue analysis in favor of static analysis of the maximum loading condition. The light weight techniques which will be examined include making the shafts hollow or using a bimetallic billet which utilizes as low density core. The main objectives of the study will be to i) create a load map of the truck which can be used to determine the loads on the various components for future simulation, ii) carry out a series of baseline simulations on the selected parts to determine the stress level in a conventional part, iii) perform a series of simulations on the selected parts which are modified to reduce their weight, and iv) determine the potential weight savings that can be achieved by innovative lightweight fabrication techniques.

In order to accomplish these goals, the study is divided into two major parts. The first part is dedicated to understanding the distribution and magnitude of the loads on the major heavy duty truck systems and the components that make up these systems. In this part, the way that the loads are transferred though the power train of the truck are mapped out and free body diagrams of the individual components are created. The second part covers weight reduction finite element studies
of selected power train components. This part is divided into several chapters, each investigating individual power train parts. Chapter 3 covers the study of the axle shaft, chapter 4 covers the study of the gearbox input shaft, Chapter 5 covers the study of the gearbox output shaft, and Chapter 6 covers the study of the gearbox counteraft. Finally, the content of the report is summarized in a short conclusion chapter.
Chapter 2

Load Map and Free Body Diagrams

The loading of each of the class 7 and 8 truck components is critical for the evaluation of lightweight alternatives to the conventional parts. This will allow the performance of light weight designs to be compared to the conventional components. The loading conditions on each of the components can be determined by systematically determining the loads which are transferred from the larger assemblies to the smaller assemblies and finally to the individual parts. In this way, loads on individual engine components, gearbox parts, etc. can be determined from information about the loads on the truck itself. The load maps presented in this chapter are based on a five axle tractor trailer style truck with an engine capable of producing 2440 N m [1800 lb ft], a gearbox with 10 speeds and rated at 2400 N m [1800 lb ft], and a tandem rear axle rated at 3550 N m [2600 lb ft].

The loads acting on the reference truck are depicted in Figure 2-1 and a description of the loads is given in Table 2-1. There are numerous state and federal regulation concerning the dimensions and weight of heavy duty trucks operating on the highway, so in order to be consistent the weight limits of the truck will be picked according to the regulations of the state of North Carolina [1].

Figure 2-1: Loads acting on a Typical Tractor Trailer Configuration
Table 2-1: Loads Acting on the Reference Vehicle

<table>
<thead>
<tr>
<th>Load Type</th>
<th>Load Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Steering Axle Load</td>
<td>MAX 89,000 N [20,000 lbs]</td>
</tr>
<tr>
<td>Tandem Axle Load</td>
<td>MAX 169,000 N [38,000 lbs]</td>
</tr>
<tr>
<td>Trailer Axle Load</td>
<td>MAX 169,000 N [38,000 lbs]</td>
</tr>
<tr>
<td>Truck Weight</td>
<td>Variable, Gross Vehicle Weight = 356 kN [80,000lbs]</td>
</tr>
<tr>
<td>Trailer Weight</td>
<td>Variable, Gross Vehicle Weight = 356 kN [80,000lbs]</td>
</tr>
<tr>
<td>Drive Force</td>
<td>38,000 – 15,000 N [3,370 – 8,540 lbs]</td>
</tr>
<tr>
<td>Drag</td>
<td>Dependent of Velocity and Wind Conditions</td>
</tr>
</tbody>
</table>

The focus of this investigation is to reduce the weight of the components in the heavy duty truck, but this will not necessarily reduce the gross weight of the vehicle because as the weight of the truck increases the weight of the cargo will be increased to maximize profits, in accordance to state and federal weight regulations. Therefore, this study will mainly focus on components which contribute to the weight of the unloaded truck with a special focus on the parts which can be lightened using forging practices. By detaching the trailer and the cab to focus on these components, a simplified free body diagram is created as shown in Figure 2-2.

Figure 2-2: Loads on the Chassis and Drive Train Assembly [2]
2.1 Engine Load Map

The engine is a complicated system with a high number of parts moving rapidly and with precise timing. Based on a survey of major engine suppliers, the engines typically used in class 7 and 8 trucks are in an inline 6 configuration and have displacements of 11 to 16 L [670 to 980 in³]. They can produce a maximum torque of anywhere from 1690 – 2780 N m [1250 – 2050 lb ft] and a maximum power output of between 240 and 450 kW [325 and 600 HP] [3]- [5]. The parts associated with power generation in the engine are the piston head, connecting rod, and the crankshaft. The piston head creates a moving seal on the cylinder walls and transfers the force of the expanding gasses from the combustion chamber to the connecting rod, which, in turn, transfers the load onto the eccentric lobes of the crankshaft creating a torque. The majority of the torque is then transferred to the gearbox, with the remaining torque being diverted to running the cam shafts or auxiliary systems (e.g. air conditioning). The load transfer between the various engine components is depicted in Figure 2-3 and information about the individual loads is given in Table 2-2.

The pressure load on the piston head depends on several parameters including the types of fuel being used, compression ratio, type of aspiration, etc. and varies as a function of the angle of the crank shaft. Large diesel engines are typically in a 4 stroke configuration meaning that loading on the engine repeats itself every 720°. A typical pressure vs. crank angle plot is shown in Figure 2-4 [6]. Each of the cylinders is offset by a certain number of degrees on the crank angle to establish the most even loading possible on the crankshaft, meaning that, in general, the pressures in the various cylinders are not equal. Because the system is reciprocating rapidly, there are dynamic loads which must be accounted for when determining the loads on the crank journals and piston pins. A simplified way of accounting for the dynamics of the system is to consider the engine to behave like a simple slider crank mechanism which is operated a constant angular velocity, as shown in Figure 2-5.
Table 2-2: Summary of Forces Acting on the Major Engine Components

<table>
<thead>
<tr>
<th>Force</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pressure Load</td>
<td>MAX~15 MPa [2,175 psi] Function of Crank Angle</td>
</tr>
<tr>
<td>Gearbox Torque</td>
<td>2440 N − m [1800 ft − lbf]</td>
</tr>
<tr>
<td>Accessory System Torque</td>
<td>Depends on Valve Actuating System + Other Loads from Engine</td>
</tr>
<tr>
<td>Cylinder Wall Load</td>
<td>Function of Crank Angle and Engine Dynamics</td>
</tr>
<tr>
<td>Connecting Rod Load</td>
<td>Function of Crank Angle and Engine Dynamics</td>
</tr>
<tr>
<td>Piston Head Load</td>
<td>Equal and opposite to Connecting Rod Load</td>
</tr>
<tr>
<td>Crankshaft Load</td>
<td>Function of Crank Angle and Engine Dynamics</td>
</tr>
<tr>
<td>Crankshaft Journal Loads</td>
<td>Equal and Opposite to Individual Crankshaft Loads</td>
</tr>
</tbody>
</table>

Figure 2-4: In Cylinder Pressure as a Function of the Crank Angle
The gearbox is an intricate system designed to transmit an appropriate torque to the drive shaft while keeping the engine speed in the optimal range. The gearboxes of modern day heavy duty trucks can have as many as 15 forward speeds and 3 reverse speeds. This large number of gear ratios is accomplished by combining 5 individual gear ratios with 3 range selectors. A simpler example of a gearbox with a range selection is shown in Figure 2-6 [7]. The gearbox depicted is a 10 speed Mack T310 gearbox [8]. The torque from the engine is transmitted to the input shaft through the splines and then transferred to the countershaft through the gear on the opposite side of the shaft. The counter shaft receives the torque from its input gear which rotates the shaft and all the gears on the shaft at the same speed, according to the gear ratio between the input gear and the input shaft. The output gears of the countershaft are always engaged with the corresponding gears on the output shaft, but only one gear transmits torque from the countershaft to the output shaft at a time. The gear which is engaged is selected by meshing a sliding clutch on the output shaft with the desired gear which forces the gear to rotate with the shaft, the rest of the gears on the output shaft are free to rotate at a different speed than the shaft. The output shaft will either be directly engaged with the drive shaft, or it will be engaged with another gear set with two to 3 speeds which allows the operator to select a gear range.

In order to transfer torque from gear to gear the meshing teeth apply forces to each other. The magnitude of these forces is dependent on the diameter of the gear and the geometry of the teeth. Modern day gearboxes typically have helical gears, though some gearboxes still utilize spur gears, and most gears have a pressure angle of 20°. Table 2-3 summarizes the forces which might be found on gears with the proportions shown in Figure 2-6. It should be noted that these loads are determined based on the assumption that there is only one countershaft in order to properly demonstrate the forces that are created from the meshing teeth. However, the real Mack T310
gearbox has 3 countershaft and many other gearboxes have 2 countershafts which allows them to reduce the loads on the gear teeth by $\frac{1}{\text{# of countershafts}}$ and eliminate the bending loads on the input and output shafts through proper balancing of the radial and circumferential loads.
Figure 2-6: Gearbox Load Map and Loading of Individual Gearbox Components

Table 2-3: Summary of Forces Acting on the Major Gearbox Components

<table>
<thead>
<tr>
<th>General</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Engine Torque</td>
<td>2,440 N – m [1,800 ft – lbf]</td>
</tr>
<tr>
<td>Drive Shaft Torque (6th Gear)</td>
<td>6,520 N – m [4,810 ft – lbs]</td>
</tr>
<tr>
<td>Input Shaft</td>
<td></td>
</tr>
<tr>
<td>Gear Radial Load</td>
<td>18,200 N [4,090 lbs]</td>
</tr>
<tr>
<td>Gear Circumferential Load</td>
<td>41,700 N [9,730 lbs]</td>
</tr>
<tr>
<td>Gear Thrust Load</td>
<td>25,000 N [5,620 lbs]</td>
</tr>
<tr>
<td>Countershaft</td>
<td></td>
</tr>
<tr>
<td>Input Gear Radial Load</td>
<td>18,200 N [4,090 lbs]</td>
</tr>
<tr>
<td>----------------------------</td>
<td>----------------------</td>
</tr>
<tr>
<td>Input Gear Circumferential Load</td>
<td>41,700 N [9,730 lbs]</td>
</tr>
<tr>
<td>Input Gear Thrust Load</td>
<td>25,000 N [5,620 lbs]</td>
</tr>
<tr>
<td>Output Gear Radial Load</td>
<td>41,400 N [9,300 lbs]</td>
</tr>
<tr>
<td>Output Gear Circumferential Load</td>
<td>98,800 N [22,200 lbs]</td>
</tr>
<tr>
<td>Output Gear Thrust Load</td>
<td>56,900 N [12,800 lbs]</td>
</tr>
</tbody>
</table>

**Output Shaft**

<table>
<thead>
<tr>
<th>Drive shaft Torque</th>
<th>6,520 N – m [4,810 ft – lbs]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gear Radial Load</td>
<td>9,320 N [6,870 lbs]</td>
</tr>
<tr>
<td>Gear Circumferential Load</td>
<td>22,200 N [16,300 lbs]</td>
</tr>
<tr>
<td>Gear Thrust Load</td>
<td>12,800 N [9,440 lbs]</td>
</tr>
</tbody>
</table>

**2.3 Rear Axle Load Map**

The rear axles are the final step in the power train as they deliver the torque to the wheels which ultimately creates the driving force for the truck. The rear axle assembly consists of a differential assembly, a pair of axle housings, the axle shafts, and the various braking and mounting components. The differential assembly splits the load between the two sides of the axle and, in the case of both of the tandem axles being powered, between the front and rear tandem axles. It is also the source of another gear reduction. Figure 2-7 shows how the torque is transmitted from the drive shaft to the individual axles shafts and finally to the wheels themselves. Note that, the axle shafts depicted in this figure are full floating axles, which means that they only carry torque and that all of the bending loads are carried by the axle housing. Under cruising conditions, the torque would be divided evenly between the axle shafts (4 ways for a 6X4 configuration and 2 ways for a 6X2 configuration). Likewise the total load on the tandem axle will be divided evenly between the 4 halves of the tandem axle pair. A summary of the loads applied to the rear axle assembly is shown in Table 2-4. These numbers were arrived at by carrying out a survey of typical rear axles to determine the torque rating and gear ratios commonly used in modern day for the rear axle assemblies [9].
Figure 2-7: Rear Axle Load Map and Loading of Individual Gearbox Components
Table 2-4: Summary of the Loads Acting on the Rear Axle Assembly

<table>
<thead>
<tr>
<th>Load Type</th>
<th>Load Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Drive Shaft Torque</td>
<td>3,550 N m [2,620 lb ft]</td>
</tr>
<tr>
<td>Axle Shaft Torque</td>
<td>8,630 to 21,900 N m [6,370 to 16,200 lb ft]</td>
</tr>
<tr>
<td>Tire Torque</td>
<td>Equal and Opposite to Axle Shaft Torque</td>
</tr>
<tr>
<td>Suspension Load</td>
<td>42,300 N [9,500 lbs]</td>
</tr>
<tr>
<td>Tire Load</td>
<td>42,300 N [9,500 lbs]</td>
</tr>
</tbody>
</table>

2.4 Summary

The loads on the truck are used to determine the loads on the major systems which make up the truck powertrain. The loads are determined through a combination surveys of engines, gearboxs, and rear axles and calculations based on gear ratios, geometry, and dynamic analysis. Using the loads mapped out in this chapter detailed finite element models of the components outlined in this report were created and employed in the study of lightweight designs of selected power train components.
Chapter 3
Axle Shaft: Finite Element Study on Lightweight Shafts

3.1 Introduction

Using the loads determined in chapter 2, it is possible to create a finite element model of the axle shaft. This finite element model is first used to examine the distribution of stresses in the conventional, solid axle shaft. These stresses are then used as a baseline to define the required performance of a lightweight design. Two methods for reducing the weight of the shaft are then investigated and evaluated in regards to the baseline study. Finally, the chapter is concluded with an outlook on the possible weight savings that could be achieved by adopting the lightweight techniques investigated.

The axle shaft examined in this section is a fully floating axle shaft, meaning that it serves to transmit only torque to the wheels and does not help to support the weight of the truck in any way. The axle shaft consists of a splined side which mates with the differential and a flanged side which is attached to the wheels. The axle shaft receives torque from the differential in accordance to the gearing ratios of the gearbox and the differential and transmits it to the tires. Based on a quick survey of commercial axle shafts the torque rating for tandem axle shafts is 3550 N-m (2618 ft-lbf) and rear axle ratios can get as high as 6.17:1. Assuming that the truck is in a 6X4 configuration (i.e. both the front and rear tandem axle are driven) and the torque is divided evenly between the axle shafts, the torque on each shaft can be determined by multiplying the torque by the axle ratio and then dividing by the number of axles (4). Using this method the torque on each shaft can be determined to be about 5476 N-m (4039 ft-lbf). Additionally, because the shaft is rotating there will be an additional inertial load on the shaft, though the effects of this load are expected to be small. Assuming that the truck’s top speed is 105 km/h (65 mph) and the tire diameter is 1.15m (45.4 in) the rotational velocity of the shaft is 50.4 rad/s. These loads are summarized in Figure 3-1.
3.2 Solid Axle Shaft (Baseline)

The baseline simulations of the axle shaft serve to establish the nominal stresses in a conventional axle shaft under its maximum loading conditions. These stresses can then be compared to the stresses observed in later simulations which utilize lightweight concepts. This allows for the evaluations of individual light weight designs and provides some idea of the necessary performance of the light weight component.

The geometry of the simulated axle shaft was taken from drawings of a full floating axle shaft and the diameter of the smallest cross-section of the shaft was 47.6 mm [1.875 in]. However, features like splines were ignored in order to simplify the simulation. It was meshed with tetrahedral elements and the mesh was refined in areas where stress concentration were expected (i.e. the radii) as shown in Figure 3-2. The material was approximated by ANSYS’s structural steel model which has a Young’s Modulus of 200 GPa [29,000 ksi] and a Poisson’s ratio of 0.3. Only the material’s elastic behaviors were simulated, as plastic deformation of the part would be considered failure. The torque load was applied as a moment boundary condition on the area of the shaft into which splines would be cut, and the shaft was held in place via fixed displacement boundary conditions in the holes to which the tires are mounted on the flange. A rotational velocity was applied to the entire shaft to simulate the inertial loads on the shaft.
After the completion of the simulation the stress contours were plotted as shown in Figure 3-3. The maximum stress fell into the 480 - 420 MPa [69.6 – 60.9 ksi] range and occurred in the thinnest section of the axle shaft. Due to stress concentrations caused by the changes in diameter on the shaft the stress was slightly higher around the transitions from the small diameter section of the shaft to the flange and the splines. It is interesting to note that the stresses in the shaft are concentrated on the outer diameter of the shaft and that the stress in the middle of the shaft is almost zero, as shown in Figure 3-4. Thus the material in the middle of the shaft increases the weight of the part without contributing significantly to its strength. The weight of the shaft could be significantly reduced by removing this parasitic material from the inside of the shaft or by replacing it with a lower density material.
3.3 Hollow Axle Shaft

The following simulations demonstrate how the performance of the shaft can be maintained while significantly reducing the weight of the part by removing the parasitic material from the center of the shaft. For the purposes of these simulations, the shaft was hollowed by removing a cylindrical section of material from the center of the axle shaft. In practice this would be difficult to fabricate as a single part, as it would require the forged blank to be have a hole drill through its entire length (about 1.2 m [4 ft]) which would be time consuming and difficult. Another way to
fabricate this shaft would be to start with a thick tube and weld a flange onto its end although this too would come at the cost of additional required equipment manufacturing steps. Perhaps the most attractive way to fabricate this shaft would be to come up with an innovative way to forge a flange onto a tubular blank, eliminating the need for time consuming joining and machining steps.

The first simulation which was carried out had the same outside dimensions as the baseline axle shaft, but had a 6.35 mm [0.25 in] hole through the center of the part. The stress distribution in this part is given in Figure 3-5. The change in the maximum stress in the part is nearly unnoticeable with the maximum stress again falling into the 480 to 420 MPa [69.6 to 60.9 ksi] range. However, there was also not a significant weight savings from the removal of this small amount of material (the weight of the part was only reduced by 0.18 kg [0.4 lbs]). Clearly more material must be removed to see significant weight savings.

Figure 3-5: Stress Distribution in a Hollow Shaft with an Outer Diameter of 47.6 mm [1.875 in] and an Inner Diameter of 6.35 mm [0.25 in]

The effects of removing a significant amount of material were investigated in the next simulation. Once again the outside geometry of the shaft was kept the same as the baseline
simulation, but this time a 31.75 mm [1.25 in] diameter cylinder of material was removed from the center of the shaft. This resulted in a weight savings of 4.4 kg [9.7 lbs], but came at the cost of increasing the stress in the shaft to nearly 580 MPa [84.1 ksi]. This rise in stress will have to be mitigated by either changing materials to one which is stronger, or by increasing the diameter of the shaft. Fortunately, the stress in the shaft is proportional to $\frac{1}{r^3}$ while the weight in the shaft is only proportional to $r^2$, meaning that the stress in the shaft will decrease faster than the weight will rise as a result of the added material. As can be seen from the distribution of the stresses in the shaft (Figure 3-6), there is very little parasitic material in this axle shaft. The material in the center of the shaft is carrying a stress of about 360 MPa [52.5 ksi] which is nearly 2/3 of the maximum load in the shaft.

![Stress Distribution in a Hollow Shaft](image)

Figure 3-6: Stress Distribution in a Hollow Shaft with an Outer Diameter of 47.6 mm [1.875 in] and an Inner Diameter of 31.75 [1.25 in]

The next simulations show how the stresses in the shaft change as a result of a slight increase in the diameter of the shaft. The aspect ratio between the inner and outer diameter were kept the same as in the previous simulations, but the diameter was increased to 50.8 mm [2 in] (an increase
of 3.175 mm (0.125 in)). The stress distribution in the thick and thin walled shafts is given in Figure 3-7 and Figure 3-8. The increase in the diameter resulted in a decrease in the stress of about 17% in both cases and resulted in a weight increase of a little less than 10%. Clearly this strategy of increasing the diameter of the shaft to compensate for the increasing stress that results from removing parasitic material can provide a significant weight savings, but in order to fully determine its potential a larger sample of the design space must be analyzed.

Figure 3-7: Stress Distribution in a Hollow Shaft with an Outer Diameter of 50.8 mm (2 in) and an Inner Diameter of 6.78 mm (0.267 in)
To determine the potential weight savings of a hollow shaft a parametric study was run which examined the effects of varying the outer diameter and wall thickness to outer diameter ratio of the axle shaft on the maximum stress in the shaft and the weight of the shaft. Figure 3-9 demonstrates how the stress in the shaft changes with respect to the stress in the baseline study as the wall thickness ratio and shaft diameter change. The point where the stress curves for the various diameters are equal to the baseline stress represents the wall thickness ratio that will create a shaft with equal performance to the conventional, solid shaft for the shaft diameter corresponding to the particular curve. These wall thickness ratios can then be used in conjunction with Figure 3-10 to determine the weight savings of each of the equal performance designs.

The stress in the shaft is clearly not sensitive to small holes in the shaft, but as the hole increases in diameter it removes material that is responsible for carrying an ever increasing amount of the load, resulting in rapidly rising stresses as the wall thickness ratio approaches zero. Additionally, increases in the outer diameter of the shaft serve only to shift the stress curve down, without significantly affecting the shape of the curve. The weight of the equally performing shaft tends to decrease as the outer diameter of the shaft is allowed to increase, so the designer will have to weigh
the cost of reduced weight against the drawback of a less compact shaft. Based on the study performed, the weight of the axle shaft could be reduced by 25-35\% by switching to a hollow shaft, depending on the outer diameter of the replacement shaft. This is a shavings of 5.6 to 7.3 kg [12.3 to 16.1 lbs] per shaft for a total savings of 22.5 to 29.4 kg [49.6 to 64.8 lbs] on the entire vehicle.

Figure 3-9: Maximum Stress in the Hollow Axle Shaft as a Function of Hole Aspect Ratio for Various Diameter Shafts

Figure 3-10: Mass of the Hollow Axle Shaft as a Function of Hole Aspect Ratio for Various Diameter Shafts


### 3.4 Bimetallic Axle Shaft

The next weight-reduction technique to be investigated is the bimetallic axle shaft. These simulations used the same geometry as the hollow axle shaft simulations above, except the hollow region has been filled with a lighter material, in this case Aluminum, while the outer material remains steel. The aluminum was assumed to have a Young’s Modulus of 71 GPa [10,300 ksi] and a Poisson’s ratio of 0.33, while the steel had a Young’s Modulus of 200 GPa [29,000 ksi] and a Poisson’s ratio of 0.3. These simulations were carried out in the same manner as the hollow shaft simulations discussed in section 3.3.

Figure 3-11 shows the stress distribution for the bimetallic shaft with an outer diameter of 47.6 mm [1.875 in] and an inner diameter of 6.35 mm [0.25 in]. The maximum stress in this simulation is in the 480-420 MPa [69.6 to 60.9 ksi] range, similar to the stress in the hollow shaft with these dimensions. Figure 3-12 shows the results for the simulation for the bimetallic shaft with an outer diameter of 47.6 mm [1.875 in] and an inner diameter of 31.75 mm [1.25 in]. The maximum stress in this simulation is in the 540-480 MPa [78.3 to 69.6 ksi] range, higher than the previous simulation, but lower than the hollow simulation with these dimensions. The results of the simulation for the shaft with an outer diameter of 50.8 mm [2 in] and an inner diameter of 6.78 mm [0.267 in] is shown in Figure 3-13. The maximum stress in this simulation is in the 420-360 MPa [60.9 to 52.2 ksi] range, an improvement upon the previous simulation and, again, less than the hollow simulations with the same dimensions. Finally, Figure 3-14 shows the results of the shaft with an outer diameter of 50.8 mm [2 in] and an inner diameter of 33.8 mm [1.33 in]. The maximum stress in this simulation is also in the 480-420 MPa [69.6 to 60.9 ksi] range.
Figure 3-11: Stress Distribution in a Bimetallic Shaft with an Outer Diameter of 47.6 mm [1.875 in] and an Inner Diameter of 6.35 mm [0.25 in]
Figure 3-12: Stress Distribution in a Bimetallic Shaft with an Outer Diameter of 47.6 mm [1.875 in] and an Inner Diameter of 31.75mm [1.25 in]
Figure 3-13: Stress Distribution in a Bimetallic Shaft with an Outer Diameter of 50.8 mm [2 in] and an Inner Diameter of 6.78 mm [0.267 in]
Another parametric study was performed on the bimetallic axle shafts in a similar manner as the study performed on the hollow shafts. Figure 3-15 below shows the maximum stress in the bimetallic axle shaft as a function of the aspect ratio of the hole (wall thickness to diameter) for three different outer diameter shafts. It can be seen that for each aspect ratio, the larger outer diameter has a lower maximum stress. It can also be seen that for each outer diameter, the stress decreases as the hole aspect ratio increases (wall thickness increases). Figure 3-16 shows the mass of the bimetallic axle shaft as a function of the hole aspect ratio. At each aspect ratio, it can be seen that the shaft with the larger outer diameter will have the highest mass. For each outer diameter, as the aspect ratio increases, the mass will also increase. By using Figure 3-15 one can determine what the internal dimensions of the shaft are that will create a shaft that performs equally...
with a solid steel shaft, for a given outer diameter. This information can be used in conjunction with Figure 3-16 to determine the potential weight savings that can be achieved by switching from a solid steel axle shaft to a slightly larger bimetallic axle shaft. In this case it appears that the weight of the axle shaft can be reduced by 4 to 4.8 kg [8.8 to 10.55 lbs], for a total vehicle savings of 16 to 19.2 kg [35.2 to 42.2 lbs].

![Graph showing stress and mass as a function of wall thickness/diameter for various diameter shafts.](image)

**Figure 3-15:** Maximum Stress in the Bimetallic Axle Shaft as a Function of Hole Aspect Ratio for Various Diameter Shafts

![Graph showing mass as a function of wall thickness/diameter for various diameter shafts.](image)

**Figure 3-16:** Mass of the Bimetallic Axle Shaft as a Function of Hole Aspect Ratio for Various Diameter Shafts
3.5 Conclusion

The total weight of the four axle shafts used in tandem rear axles is 86 kg [189 lbs], meaning there is a possibility for significant weight savings in the heavy duty axle shafts. Because the weight of the vehicle is supported by the axle housing in the fully floating axle design the only load on the axle shaft is the torsional load. This means that the stress is concentrated on the outside surface of the part, leading to a high amount of material that carries almost no load. The removal of this parasitic material has the potential to remove 22.5 to 29.4 kg [49.6 to 64.8 lbs] of unnecessary weight from the vehicle, depending on the degree to which the diameter of the shaft is allowed to expand. Making the axle shaft out of a bimetallic billet also has the potential to reduce the weight of the vehicle by 16 to 19.2 kg [35.2 to 42.2 lbs], depending on the acceptable level of diameter increase for the shaft. Clearly the hollow shaft would be the ideal choice, but the decision on what modifications to make to the axle shaft cannot be made until the methods (and costs) of manufacturing these shaft variants have been determined.
Chapter 4

Input Shaft: Finite Element Study on Lightweight Shafts

4.1 Introduction

The loads on the gearbox input shaft determined in chapter 2 are used as the starting point for the finite element study of the shaft. However, because the gearbox transmission system investigated has a triple countershaft configuration the bending loads can be ignored on the input shaft, as explained in section 2.2. The finite element model will first be employed to study the stress distribution in the conventional, solid shaft. This stress distribution will then be used as a baseline for which the performance of lightweight input shafts can be compared. Two lightweight techniques will be investigated, hollow shaft geometry and a bimetallic shaft configuration in which the center of the shaft is composed of a low density material. The chapter will be concluded with some remarks about the potential weight savings brought about by these lightweight designs and a summary of the study itself.

The input shaft takes the torque from the engine and sends it to the gearbox. The engine torque is transferred to the input shaft via the splines on the small diameter side of the shaft. The torques is then given to the counter shaft(s) in the gearbox via a gear which is either mounted on to the large end of the shaft or machined into a flange created during the forging of the shaft. Additionally, there is an inertial load on the shaft created by the rotational velocity of the shaft, which is equal to the rotational speed of the engine. The shaft is supported by a tapered roller bearing where it meets the gearbox casing.

In order to model the input shaft these loads and supports were transferred to a finite element model of the shaft operating under its maximum loading condition (i.e. maximum torque rating for the gearbox). The geometry used in the finite element model is shown in Figure 4-3 along with the boundary conditions applied to the model. The torque from the engine was simulated as a moment about the splined area of the shaft and the reaction from the counter shaft(s) is simulated by a displacement boundary condition that prevents the shaft from rotating, but leaves it free to move radially and axially. The rigid body motion on the shaft is prevented by applying a
frictionless support to the areas of the shaft in contact with the bearing. A rotational velocity was applied to the entire shaft to simulate the centripetal forces on the shaft.

![Diagram showing Input Shaft in the Truck Gearbox](image1)

**Figure 4-1:** Input Shaft in the Truck Gearbox

![Diagram showing Input Shaft Free Body Diagram](image2)

**Figure 4-2:** Input Shaft Free Body Diagram
4.2 Solid Input Shaft (Baseline)

In order, to gain an understanding of the maximum stresses present in the conventional part, a finite element simulation will be run on the solid steel input shaft. This simulation result will be used as a baseline to evaluate the performance of the lightweight input shafts simulated later in this chapter. The geometry was approximated from a picture of a gearbox (Figure 4-1 above) and was modeled (splines were omitted for simplification), meshed with tetrahedral elements, and was refined in areas with an expected high stress concentration. Figure 4-3 shows the mesh that was applied to the geometry and the area where the mesh was refined. The material used in the simulation was ANSYS’s structural steel which has a Young’s Modulus of 200 GPa [29,000 ksi] and a Poisson’s ratio of 0.3. A moment of 2,440.5 N m [1800 lb ft] was applied to the section of the shaft where the splines would be located, a frictionless support was added to the section where the bearing is located, and a fixed rotation of 209.4 rad/s was applied to the shaft to simulate the centripetal forces on the part. The rotational displacement of the shaft was fixed on the gearbox side of the shaft to simulate the counter torque from the gear on the end of the shaft.

![Geometry, Boundary Conditions, and Mesh used in the Baseline Simulations](image)

Figure 4-3: Geometry, Boundary Conditions, and Mesh used in the Baseline Simulations
Figure 4-4 and Figure 4-5 below show the results of the simulation of the baseline input shaft. The maximum stress from the simulation is 645 MPa [93.5 ksi]. This stress concentration occurs at the radius of the flange due to the diameter change. Most of the stress on the shaft is concentrated along the outer diameter, with very little stress being located in the center, implying that material in the shaft’s center increases the weight of the shaft without adding significant strength to the part. The weight of the input shaft could be reduced by applying weight reduction techniques such as hollowing out the center, or filling the center with a lighter metal material.

Figure 4-4: Stress [MPa] in the Solid Input Shaft
The following simulations aim to reduce the weight of the input shaft, while still maintaining performance by simply removing the dead weight from the center of the shaft. The input shaft was modeled in the same way as the baseline simulation, except now a cylindrical section was removed from the center. The geometry and the mesh of the hollow shaft are shown in Figure 4-6.
Figure 4-7 shows the results for the hollow shaft with an outer diameter of 38.1 mm [1.5 in] and an inner diameter of 6.35 mm [0.25 in]. The maximum stress for this hollow shaft is 645 MPa [93.5 ksi], identical to the stress in the baseline shaft. The results for the hollow shaft with an outer diameter of 38.1 mm [1.5 in] and an inner diameter of 18.3 mm [0.75 in] is shown in Figure 4-8. The maximum stress in this simulation is 674 MPa [97.8 ksi] which is higher than the baseline stress but still in the same 612 – 680 MPa [88.8 – 98.6 ksi] range. Figure 4-9 shows the simulation results for the hollow shaft with an outer diameter of 43.9 mm [1.73 in] and an inner diameter of 7.37 mm [0.29 in]. The maximum stress in this simulation is 421 MPa [61.1 ksi], which is much lower than the stress in the baseline shaft. Finally, Figure 4-10 shows the results of the simulation for the input shaft with an outer diameter of 43.9 mm [1.73 in] and an inner diameter of 22.1 mm [0.871 in]. The maximum stress in this simulation is 441 MPa [64.0 ksi], also lower than the baseline, but slightly higher than the previous simulation.
Figure 4-8: Stress Distribution in a Hollow Shaft with an Outer Diameter of 38.1 mm [1.5 in]
and an Inner Diameter of 18.3 mm [0.75 in]
Figure 4-9: Stress Distribution in a Hollow Shaft with an Outer Diameter of 43.9 mm [1.73 in] and an Inner Diameter of 7.37 mm [0.29 in]
A parametric study was run to determine the potential weight savings of a hollow input shaft. The study varied the outer diameter and wall thickness of the shaft to see how the stress and weight were affected.

Figure 4-11 shows the maximum stress in the hollow input shaft as a function of the aspect ratio of the hole, (wall thickness to outer diameter) for three different values of the shaft’s outer diameter. It can be seen that at any given aspect ratio, the smaller outer diameter shafts experience a higher maximum stress, while the larger outer diameter shafts experience a lower maximum stress. It can also be seen that for each outer diameter, the maximum stress experienced decreases
exponentially as the wall thickness to outer diameter ratio increases (as the inner diameter hole becomes smaller).

Figure 4-12 shows the mass of the input shaft as a function of that same aspect ratio. It can be seen that the larger outer diameter shafts have more mass than the smaller outer diameter shafts at any given aspect ratio. It is also shown that for each outer diameter, as the aspect ratio increases (the hole gets smaller), the mass also increases. By using both graphs together the potential weight savings for the hollow input shaft can be determined. First the dimensions of the hollow shaft with equal performance to the solid steel shaft must be determined, and then the weight of this geometry must be determined. Based on these results, a potential weight savings of 1.75 kg (3.85 lbs) or more can be achieved, which is a savings of 48%.
Figure 4-11: Maximum Stress in the Input Shaft as a Function of Hole Aspect Ratio for Various Diameter Shafts

Figure 4-12: Mass of the Input Shaft as a Function of Hole Aspect Ratio for Various Diameter Shafts
4.4 Bimetallic Input Shaft

For the bimetallic simulations, the two materials used were steel for the outer shaft material and aluminum for the inner shaft material. The same steel used for the baseline and hollow simulation was used here. The aluminum material had a Young’s Modulus of 71 GPa [10,300 ksi] and a Poisson’s ratio of 0.33. These shafts were modeled in the same way as the hollow shafts, but instead of leaving the cylindrical section in the center empty, it was filled in with the aluminum material, while the outer geometry of the shaft had the steel properties applied. The geometry and the mesh can be seen in Figure 4-13 below.

![Figure 4-13: Geometry, Boundary Conditions, and Mesh used in the Bimetallic Simulations](image)

Figure 4-14 shows the stress distribution for the bimetallic shaft with an outer diameter of 38.1 mm [1.5 in] and an inner diameter of 6.35mm [0.25 in]. The maximum stress in this simulation is
645 MPa [93.5 ksi], similar results to the hollow shaft with the same dimensions. Figure 4-15 shows the results for the simulation for the bimetallic shaft with an outer diameter of 38.1 mm [1.5 in] and an inner diameter of 18.3 mm [0.75 in]. The maximum stress in this simulation is 663 MPa [96.2 ksi], which is lower than the stress in the hollow shaft with the same dimensions. The results of the simulation for the shaft with an outer diameter of 43.9 mm [1.73 in] and an inner diameter of 7.37 mm [0.29 in] is shown in Figure 4-16. The maximum stress in this simulation is 418 MPa [60.6 ksi], which is only 2 MPa lower than the stress in the hollow shaft with these dimensions. Finally, Figure 4-17 shows the results of the shaft with an outer diameter of 43.9 mm [1.73 in] and an inner diameter of 22.1 mm [0.871 in]. The maximum stress in this simulation is 433 MPa [62.8 ksi], also slightly lower (8 MPa) than the hollow shaft with these dimensions. Based on the results of these simulations it is possible to replace a substantial amount of steel in the input shaft with aluminum and significantly reduce the weight of the bimetallic shaft.
Figure 4-14: Stress Distribution in a Bimetallic Shaft with an Outer Diameter of 38.1 mm [1.5 in] and an Inner Diameter of 6.35mm [0.25 in]
Figure 4-15: Stress Distribution in a Bimetallic Shaft with an Outer Diameter of 38.1 mm [1.5 in] and an Inner Diameter of 18.3 mm [0.75 in]
Figure 4-16: Stress Distribution in a Bimetallic Shaft with an Outer Diameter of 43.9 mm [1.73 in] and an Inner Diameter of 7.37 mm [0.29 in]
Figure 4-17: Stress Distribution in a Bimetallic Shaft with an Outer Diameter of 43.9 mm [1.73 in] and an Inner Diameter of 22.1 mm [0.871 in]
Figure 4-18: Maximum Stress in the Input Shaft as a Function of Hole Aspect Ratio for Various Diameter Shafts

Figure 4-19: Mass of the Input Shaft as a Function of Hole Aspect Ratio for Various Diameter Shafts
Another parametric study was performed on the bimetallic shafts, investigating the same parameters as the study done on the hollow shafts.

Figure 4-18 shows the maximum stress in the bimetallic input shaft as a function of the aspect ratio of the hole for the three different outer diameter shafts. It can be seen that for each aspect ratio, the smaller outer diameter shafts experience a higher maximum stress, while the larger outer diameter shafts experience a lower maximum stress, similar to the results of the hollow shafts. It can also be seen that for each outer diameter, the maximum stress experienced decreases exponentially as the wall thickness to outer diameter ratio increases, also similar to the hollow shaft results.

Figure 4-19 shows the mass of the input shaft as a function of the hole aspect ratio. It can be seen that the larger outer diameter shafts have more mass than the smaller outer diameter shafts at any given aspect ratio and that for each outer diameter, as the aspect ratio increases, the mass also increases. These results follow the same trends observed in the parametric study done on the hollow shafts. Based on the amount of weight which can be saved by switching from the solid steel shafts to the bimetallic shafts is about 0.9 kg [2 lbs] or 25%.

4.5 Conclusion

The mass of a single input shaft is small (only about 3.7 kg or 8.14 lbs), so the total weight which can be saved by switching to a lightweight design is also small. However, the reduction in the shaft’s weight is an important part of a larger total weight reduction which can be achieved when considering all of the small savings in the gearbox. The finite element studies carried out revealed that the weight of the shaft could be reduced by 1.75 kg [3.85 lbs] if hollow geometry was employed and 0.9 kg [2 lbs] if the bimetallic variant of the shaft was used. These savings would require the diameter of the shaft to be increased by a mere 15%. Greater savings could be expected for shafts with larger outer diameters. The small weight savings achievable on the input shaft must be weighed against the cost of updating the manufacturing process to create the lightweight components.
Chapter 5

Output Shaft: Finite Element Study on Lightweight Shafts

5.1 Introduction

Finite element studies are carried out on the gearbox output shaft based on the loading conditions determined in chapter 2. The first simulation to be carried out is a study of the stress distribution in the conventional, solid, shaft. Hollow geometry was then investigated as a possible means of lowering the weight of the shaft, followed by an investigation of a bimetallic construction of the output shaft. The performance of these two light weight designs was measured against the performance of the conventional shaft and conclusions were drawn about the possible weight savings that could be achieved by adopting one of two lightweight designs.

The output shaft takes the torque from the counter shafts and sends it to the output axles. The output shaft in the gearbox is shown in Figure 5-1. And the free body diagram of the shaft is shown in Figure 5-2. The engine torque is transferred to the input shaft, and then given to the countershafts. The countershafts transfer the torque to the output shafts through the gears that are mounted on the countershafts and meshed to the output gears on the output shaft. There are three sliding clutches on the output shaft which are able to slide on the shaft along axial direction but are forced to rotate with the output shaft. The sliding clutch is shifted to engage with the desired output gear causing the shaft to rotate at the same speed as the gear. Each of the sliding clutches has a neutral position, in which it is not engaged with any gear, and two positions that engage it with a particular gear, resulting in 5 forward speeds and one reverse speed. The maximum torque that can be applied to the output shaft is achieved when the engine is applying the maximum rated torque to the input shaft and the first gear is engaged with the sliding clutch. In order to simulate the maximum torsion loading condition on the output shaft, a moment load will be applied to the area where the sliding clutch would be when it is engaged with the first gear. The shaft is supported by roller bearings in the gearbox. Frictionless support will be applied on the bearing contact area around the shaft tip on the left. In order to simulate the torque which resists the acceleration of the shaft, the rotational degree of freedom is removed from the end of the shaft where it would mate with its output gear. Additionally, there is an inertial load on the shaft created by the rotational velocity of the shaft, which is equal to the rotational speed of the input shaft multiplied by the
specific gear ratio. Since there are three balanced countershafts, the bending loads that result from the meshing of the gears of the countershafts with the gear on the output shaft cancel out and only a net torsional load is applied to the output shaft.

Figure 5-1: The output shaft in the truck engine

Figure 5-2: Output shaft FBD
5.2 Solid Output Shaft (Baseline)

The baseline simulations of the output shaft serve to establish the nominal stresses in a conventional output shaft under its maximum loading conditions. These stresses can then be compared to the stresses observed in later simulations which utilize lightweight concepts. This allows for the evaluations of individual light weight designs and provides some idea of the necessary performance of the light weight component.

The geometry of the simulated output shaft was taken from drawings of an actual output shaft and the diameter of the cross-section on the shaft where the first ratio gear was located was 49.78mm [1.96 in]. As shown in Figure 5-3, the output shaft with solid center was modeled in Static Structural of ANSYS Workbench as Baseline Simulation. The geometry of shaft’s outside surface was simplified by removing features like splines and snap ring seats to ease the simulation process. The corners on the step surfaces were filleted with a 1.27mm [0.05 in] radius. The material was approximated by ANSYS’s structural steel model which has a Young’s Modulus of 200 GPa [29,000 ksi] and a Poisson’s ratio of 0.3. Only the material’s elastic behaviors were simulated, as plastic deformation of the part would be considered failure.

As illustrated in Figure 5-3, the shaft was meshed with tetrahedral elements and the mesh was refined in areas where the stress concentration was expected to be high, i.e. the fillets at the steps on the shaft. The left side of the shaft, where the shaft is mated with its output gear, has its rotational degree of freedom removed via a displacement boundary condition and a moment boundary condition of 6,520 N m [4,810 lb ft] was applied to the area where the sliding clutch meshes with the first gear and output shaft. Additionally, a rotational velocity of (76.1 rad/s) is applied to the shaft to simulate the centripetal forces on the part and the area where the rest of the rigid body modes of motion are removed by applying a frictionless boundary condition to area of the shaft which is in contact with the roller bearing.
Figure 5-3: Geometry, Boundary Conditions, and Mesh used in the Baseline Simulations of the Output Shaft

After the completion of the simulation the stress contours were plotted as shown in Figure 5-4. Due to numerical error cause by the simplification of the real loading conditions to the boundary conditions used in the finite element simulations, specifically the fixed rotation condition, the stress was higher around the transitions from the free surface to the fixed surface on the shaft. Ignoring the numerical error around the fixed boundary condition, the maximum stress on the output shaft was about 466 MPa [67.6 ksi]. The shaft only sees stress in a relatively localized area because only the first gear is engaged. If, for instance, the fourth gear were engaged, instead of the first, the entire shaft would be subjected to stress.

As was the case in the previous chapters, the stresses in the output shaft are concentrated on the outer diameter of the shaft and that the stress in the middle of the shaft is almost zero, as shown in Figure 5-4. Thus the material in the middle of the shaft increases the weight of the part without contributing significantly to its strength. The weight of the shaft could be significantly reduced by removing this parasitic material from the inside of the shaft or by replacing it with a lower density material.
5.3 Hollow Output Shaft

The following simulations demonstrate how the performance of the shaft can be maintained while significantly reducing the weight of the part by removing the parasitic material from the center of the shaft. As shown in Figure 5-5, for the purposes of these simulations, the shaft was hollowed by removing a section of material from the center of the output shaft which was similar to the outer diameter of the shaft. The ratios between the outside diameters and inside diameters for each main cross section are kept the same ($D_o/D_i = \text{Constant}$). In practice this would be difficult to fabricate as a single part, as it would require the forged blank to have a drilled hole with multiple diameters which would be time consuming and difficult for machining. A possible and attractive way to fabricate this shaft would be to start with a thick tube and forge the tube into shafts with the correct inner to outer diameter ratios.

The first hollow shaft simulation which was carried out had the same outside dimensions as the baseline output shaft, but had a 6.35mm [0.25 in] hole on the large diameter section of the part where the first ratio gear was connected. The stress distribution in this part is given in Figure 5-6.
The change in the maximum stress in the part is nearly unnoticeable with the maximum stress again was around 466 MPa [67.6 ksi]. However, there was also not a significant weight savings from the removal of this small amount of material (the weight of the part was only reduced 0.079 kg [0.174 lbs]). Clearly more material must be removed to see significant weight savings.

Figure 5-5: Geometry, Boundary Conditions, and Mesh used in the Simulations of the Hollow Output Shaft
The effects of removing a more significant amount of material were investigated in the next simulation. Once again the outside geometry of the shaft was kept the same as the baseline simulation, but this time a 25.4 mm [1.0 in] diameter cylinder of material was removed from the large diameter section the shaft. This resulted in a weight savings of 1.24 kg [2.73 lbs], but came at the cost of increasing the stress in the shaft to nearly 500 MPa [72.5 ksi]. This rise in stress will have to be mitigated by either changing materials to one which is stronger, or by increasing the diameter of the shaft. As discussed in chapter 3, the stress in the shaft will decrease faster than the weight will rise as a result of the increased diameter. The distribution of the stresses in the shaft (Figure 5-7) also shows, there is very little parasitic material in this output shaft. The material in the center of this section of the shaft is carrying a stress of about 254 MPa [36.8 ksi] which is nearly 1/2 of the maximum load in the shaft.
The next simulations show how the stresses in the shaft change as a result of a slight increase in the diameter of the shaft. The aspect ratio between the inner and outer diameter were kept the same as in the previous simulation, but the main shaft diameter was increased to 57.15mm [2.25 in] (an increase of 6.35mm [0.25 in]). The stress distribution in the thick and thin walled shafts is given in Figure 5-8 and Figure 5-9. The increase in the diameter resulted in a decrease in the stress of about 21.5% and resulted in a weight increase of about 17.5% in both cases. Clearly this strategy of increasing the diameter of the shaft to compensate for the increasing stress that results from removing parasitic material can provide a significant weight savings, but in order to fully determine its potential a larger sample of the design space must be analyzed.
Figure 5-8: Stress Distribution in a Hollow Output Shaft with an Outer Diameter of 57.15mm [2.25 in] and an Inner Diameter of 7.14mm [0.281 in]

Figure 5-9: Stress Distribution in a Hollow Output Shaft with an Outer Diameter of 57.15mm [2.25 in] and an Inner Diameter of 28.58mm [1.125 in]
To determine the potential weight savings of a hollow output shaft, a parametric study was run which examined the effects of varying the outer diameter and wall thickness to outer diameter ratio of the output shaft on the maximum stress in the shaft and the weight of the shaft. Figure 3-9 demonstrates how the stress in the shaft changes with respect to the stress in the baseline study as the wall thickness ratio and shaft diameter change. The point where the stress curves for the various diameters are equal to the baseline stress represents the wall thickness ratio that will create a shaft with equal performance to the conventional, solid shaft for the shaft diameter corresponding to the particular curve. These wall thickness ratios can then be used in conjunction with Figure 3-10 to determine the weight savings of each of the equal performance designs.

The stress in the shaft is clearly not sensitive to small holes in the shaft, but as the hole increases in diameter it removes material that is responsible for carrying an ever increasing amount of the load, resulting in rapidly rising stresses as the wall thickness ratio approaches zero. Additionally, increases in the outer diameter of the shaft serve only to shift the stress curve down, without significantly affecting the shape of the curve. The weight of the equally performing shaft tends to decrease as the outer diameter of the shaft is allowed to increase, so the designer will have to weigh the cost of reduced weight against the drawback of a less compact shaft. Based on the study performed the weight of the axle shaft could be reduced by 35-44% by switching to a hollow shaft, depending on the outer diameter of the replacement shaft. This is a savings of 1.7 to 2.2 kg [3.74 to 4.84 lbs] for the output shaft.
5.4 Bimetallic Output Shaft

The following simulations demonstrate how the performance of the shaft can be maintained while significantly reducing the weight of the part by replacing the parasitic material in the center with light metals like aluminum. As shown in Figure 5-12, the geometry of the bimetallic output
shaft is nearly identical to the hollow shaft geometry, except for the empty portion of the shaft is replaced with an aluminum material, which was assumed to have a Young’s Modulus of 71 GPa [10,300 ksi] and a Poisson’s ratio of 0.33. This would be a challenging part to fabricate as the two parts must be well bonded so that the core can assist in carrying the torsional load applied to the shaft. It is possible that a high quality bond could be established by forging the shaft out of a bimetallic billet, which could make the bond as strong as the shear strength of the weaker material.

The first bimetallic simulation which was carried out had the same outside dimensions as the baseline output shaft, but had a 6.35 mm [0.25 in] aluminum core in large diameter portion of the shaft. As a reminder, the geometry of the aluminum core is identical to the geometry of the hole in the hollow shaft simulations. The stress distribution in this part is given in Figure 5-13. The change in the maximum stress in the part is nearly unnoticeable with the maximum stress again was around 466 MPa [67.6 ksi] range. However there was also not a significant weight savings from replacing this small amount of parasitic steel material with aluminum (the weight of the part was only reduced 0.05 kg [0.11 lbs]). Clearly more material can be replaced to see significant weight savings.
Figure 5-12: Geometry, Boundary Conditions, and Mesh used in the Simulations of the Bimetallic Output Shaft.

Figure 5-13: Stress Distribution in a Bimetallic Output Shaft with an Outer Diameter of 49.78mm [1.96 in] and an Inner Diameter of 6.35mm [0.25 in]

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The effects of replacing a significant amount of steel material were investigated in the next simulation. Once again the outside geometry of the shaft was kept the same as the baseline simulation, but this time an aluminum core with a maximum diameter of 25.4 mm [1.0 in] was used. This resulted in a weight savings of 0.8 kg [1.76 lbs], but came at the cost of increasing the stress in the shaft to nearly 487 MPa [70.6 ksi]. This rise in stress will have to be mitigated by either changing outer shaft material to one which is stronger, or by increasing the diameter of the shaft. As can be seen from the distribution of the stresses in the shaft (Figure 5-14), there is very little parasitic material in this output shaft. The material in the center of the shaft is carrying a stress of about 248 MPa [36.0 ksi] which is nearly 1/2 of the maximum load in the shaft.

Figure 5-14: Stress Distribution in a Bimetallic Output Shaft with an Outer Diameter of 49.78 mm [1.96 in] and an Inner Diameter of 25.4 mm [1 in]

The next simulations show how the stresses in the shaft change as a result of a slight increase in the diameter of the shaft. The aspect ratio between the inner and outer diameter were kept the same as in the previous simulation, but the main shaft diameter was increased to 57.15 mm [2.25 in] (an increase of 6.35 mm [0.25 in]). The stress distribution in the thick and thin shell shafts is given in Figure 5-15 and Figure 5-16. The increase in the diameter resulted in a decrease in the
stress of about 21.5% and a weight increase of a little less than 18% in both cases. Clearly this strategy of increasing the diameter of the shaft to compensate for the increasing stress that results from replacing high density parasitic material with light metal can provide a significant weight savings, but in order to fully determine its potential a larger sample of the design space must be analyzed.

Figure 5-15: Stress Distribution in a Bimetallic Output Shaft with an Outer Diameter of 57.15mm [2.25 in] and an Inner Diameter of 7.14mm [0.281 in]
Figure 5-16: Stress Distribution in a Bimetallic Output Shaft with an Outer Diameter of 57.15mm [2.25 in] and an Inner Diameter of 28.58mm [1.125 in]

To determine the potential weight savings of a hollow shaft with light aluminum core a parametric study was run which examined the effects of varying the outer diameter and wall thickness to outer diameter ratio of the output shaft on the maximum stress in the shaft and the weight of the shaft. Figure 5-17 demonstrates how the stress in the shaft changes with respect to the stress in the baseline study as the wall thickness ratio and shaft diameter change. The point where the stress curves for the various diameters are equal to the baseline stress represents the wall thickness ratio that will create a shaft with equal performance to the conventional, solid shaft for the shaft diameter corresponding to the particular curve. These wall thickness ratios can then be used in conjunction with Figure 5-18 to determine the weight savings of each of the equal performance designs.

The stress in the shaft is clearly not sensitive to changes in the diameter of the aluminum core when the diameter of the core is small, but as the core increases in diameter it removes steel material that is carrying an increasing amount of the torsional load. Additionally, because the stiffness of the aluminum is much less than the stiffness of steel, a majority of this load is redistributed to the steel material. This results in the stress increasing rapidly as the wall thickness
ratio approaches zero, but the stress does not approach infinity as it did with the hollow shaft. Rather, it approaches the stress that a steel would reach as it approached the strain in a 100% aluminum shaft under the same loading. It is also noted that increases in the outer diameter of the shaft serve only to shift the stress curve down, without significantly affecting the shape of the curve. The weight of the equally performing shaft tends to decrease as the outer diameter of the shaft is allowed to increase, so the designer will have to weigh the cost of reduced weight against the drawback of a less compact shaft. Based on the study performed the weight of the axle shaft could be reduced by 26% by switching to a hollow shaft, depending on the outer diameter of the replacement shaft. This is a savings of 1.29 kg [2.84 lbs] for the output shaft.

Figure 5-17: Maximum Stress in the Bimetallic Output Shaft as a Function of Hole Aspect Ratio for Various Diameter Shafts
The weight of the conventional output shaft is about 4.9 kg or 10.8 lbs. The finite element study carried out in this chapter demonstrates how two lightweight designs of the shaft can reduce the weight of the part. If the shaft was made hollow, a great deal of dead weight could be removed from the center of the shaft with only a small increase in the outer diameter of the shaft. This could reduce the weight of the shaft by 1.7 to 2.2 kg [3.74 to 4.84 lbs]. Similarly the material contributing to the dead weight in the shaft could be replaced by a lower density material, which could produce weight saving of around 1.29 kg [2.84 lbs]. The decision to employ one of these light weight techniques will need to be carefully weighed against the costs of implementing design, which have yet to be determined. Although the potential savings are relatively small, one must look at them as a piece of the larger scale total reduction of the heavy duty truck mass.

Figure 5-18: Mass of the Bimetallic Output Shaft as a Function of Hole Aspect Ratio for Various Diameter Shafts

5.5 Conclusion

The weight of the conventional output shaft is about 4.9 kg or 10.8 lbs. The finite element study carried out in this chapter demonstrates how two lightweight designs of the shaft can reduce the weight of the part. If the shaft was made hollow, a great deal of dead weight could be removed from the center of the shaft with only a small increase in the outer diameter of the shaft. This could reduce the weight of the shaft by 1.7 to 2.2 kg [3.74 to 4.84 lbs]. Similarly the material contributing to the dead weight in the shaft could be replaced by a lower density material, which could produce weight saving of around 1.29 kg [2.84 lbs]. The decision to employ one of these light weight techniques will need to be carefully weighed against the costs of implementing design, which have yet to be determined. Although the potential savings are relatively small, one must look at them as a piece of the larger scale total reduction of the heavy duty truck mass.
Chapter 6
Countershaft: Finite Element Study Light Weight Shafts

6.1 Introduction

This chapter describes the finite element study carried out on both the conventional countershaft and two lightweight designs of the countershaft. The loads on the countershaft are detailed in chapter 2, although due to the triple countershaft layout of the reference gearbox the loads are lower than those described previously. The first part of the finite element study was to determine the stress distribution in the conventional, solid countershaft. This information was then used as a baseline to compare the results of the finite element analysis of the lightweight designs. The two lightweight designs investigated were a hollow shaft geometry and a bimetallic shaft, which utilized a low density core material.

The countershaft is a gearbox component which transmits torque from the input shaft to the output shaft. Gears are fixed to the shaft either through a press fit or machined into the shaft itself, so that they always rotate at the same speed as the countershaft. Additionally, the gears are always engaged with their corresponding gears on the input and output shafts. Typically, the input and output shaft are coaxial and the countershaft rotates next to the shafts on a parallel axis. Heavy duty vehicle gearboxes come with a large variety of countershaft configurations having as few as one and as many as three counter shafts. Having multiple countershafts serves two main purposes. First, because the load being transmitted from the input shaft to the output shaft is divided into multiple paths, the load on the gear teeth and the counter shaft itself is reduced by a factor of the number of counter shafts engaged. Second, having multiple countershafts allows the bending forces from the meshing gears to be balanced in such a way that there is not bending moment on the input or output shafts. However, bending moments on the countershaft cannot be avoided.

The countershaft in a Mack 10 speed Maxitorque is highlighted in Figure 6-1. This particular gearbox has three countershafts, a torque rating of 2,440 N m [1,800 lb ft], and the first/sixth gear ratio (not including the high/low range adjustment) is 2.67. Based on measurements taken of the shaft the gear ratio between the input shaft and the countershaft is about 1.43 meaning that the total torque carried by the countershafts is 3,490 N m [2,580 lb ft], or 1,163 N m [858 lb ft] per shaft. As stated previously there will be bending loads on the counter shaft. These loads can be
calculated based on the torque carried by the shaft and the size of the input and output gears. In this case the input gear and output gear corresponding to first/sixth gear have pitch circle diameters of about 163 mm [6.43 in] and 96.8 mm [3.81 in], respectively. The circumferential forces on the gears can be calculated by dividing the torque produced by the gear by the radius of the gear and the radial forces can be calculated by multiplying the circumferential force by the tangent of the pressure angle of the gears. Assuming a pressure angle of 20°, the circumferential and radial loads on the input gear are 14,230 N [3,200 lbf] and 5178 N [1,164 lbf], respectively. The corresponding loads on the output shaft are 24,060 N [5410 lbf] and 8760 N [1969 lbf], respectively. It is important to remember that the circumferential loads from the gears must balance the torque on the shaft, so they will be applied in opposite directions. Additionally, if the gears are helical, instead of spur gears, the radial load will be slightly different and there will be an axial load on the shaft as well, due to the additional helix angle on the gear teeth. For the purposes of this investigation the loading conditions for spur gears will be used because the reference gearbox uses spur gears. Finally, the shaft has inertial loads corresponding to rotational at speeds of 1400 RPM and is supported on either end by tapered roller bearings.
Figure 6-1: Mack 10 Speed Maxitorque ES T310 Gearbox with Simplified Boundary Conditions used in Finite Element Analysis

6.2 Solid Countershaft (Baseline)

The geometry of and loads applied to the countershaft in the Mack ES T310 gearbox shown in Figure 6-1 are summarized in Figure 6-2. In order to establish a baseline from which studies of light weight countershafts can be compared, a solid shaft with the dimensions shown in Figure 6-2 will be simulated via the finite element method. In addition to the shaft itself, the gears, spacers and key must simulated. This is because the gears and spacers have no space to slide on the shaft and will, therefore, serve to help the shaft resist the bending load created by the meshing gears. The key and keyway on the shaft are simulated because they transmit the load to the shaft unevenly and may have a significant effect on the stresses in the shaft. Contact conditions between the various simulated parts are as follows: i) the gears have frictionless contact with each other and the countershaft, ii) the key is bonded to the gears and the key way in the countershaft. The
circumferential and radial loads are applied as force boundary conditions to the outside surfaces of the input and output gears (this is acceptable because the analysis is focused on the countershaft and not the gears). The torque on the shaft is applied as a moment on the surface of the input gear and is countered by a using a displacement boundary condition to prevent the output gear from turning. The shaft is held in place by two frictionless supports on the ends of the shaft where the bearing would be placed on the real shaft. The simulated geometry and boundary conditions are illustrated in Figure 6-3. The mesh used to discretize the shaft, gears, and key is given in Figure 6-4.

Figure 6-2: Geometry and Loading on the Countershaft
Figure 6-3: (left) Countershaft with Gears, Spacers, and Key Simulated in the Baseline Finite Element Analysis (right) Boundary Conditions applied to the Countershaft Assembly

Figure 6-4: Mesh used in the Finite Element Simulation of the Solid Countershaft

After the simulation was run, the stress distribution on the shaft was plotted (Figure 6-5) and examined. The maximum stress on the shaft was 565 MPa [81.8 ksi] and this stress occurred on the outer radius of the keyway around the area where the output gear counters the torque produced by the input gear. The maximum stress elsewhere on the shaft is relatively low (about 110 MPa [16 ksi]). This is interesting as this stress is much lower than the stresses observed in the
simulations of the other parts examined. It is likely that the stresses in the shaft could achieve a more appreciable fraction of the yield strengths of typical automotive grades of steel if the key way was removed in favor of a coupling method which produced a smaller stress concentration. Unlike the studies of the other investigated parts, a cross sectional view of the countershaft shows a high amount of ‘parasitic’ material on not just in the middle of the shaft, but also in the areas which are far from the keyway. It is very likely that the shaft’s weight can be reduced without significantly affecting its strength by removing this material, or replacing it with a lower density material. Additionally, more efficient use of the strength of the material could be made by changing the joining method of the gears and shaft.
The following simulations demonstrate the behavior of the countershaft as material is removed from its center. The diameter of the cylindrical section of material removed from the center of the shaft and the outer diameter of the shaft were varied to determine how the weight of the shaft can be reduced without affecting the performance of the shaft. These simulations used the same boundary conditions and meshing parameters as the baseline study, so that the only difference between the two studies is the geometry. A representative mesh is given in Figure 6-6. Due to the complexity of the simulation the amount of time required for each simulation was quite high and
so the number of individual designs studied was lower than in some of the other studies. Still, some interesting conclusions can be drawn from the simulations.

Figure 6-6: Representative Mesh used in the Finite Element Study of the Hollow Countershaft

The first studies were carried out on a shaft with the same outside dimensions as the original shaft, but with different internal diameters. These stress distributions in the countershaft predicted by these finite element simulations are given in Figure 6-7 and Figure 6-8. The maximum stress in both simulations occurs at the outer radius of the keyway underneath the output gear. The magnitude of this stress is very close to the stress from the baseline simulation in this area for both simulations, while on the other side of the shaft the stresses vary with the internal diameter of the shaft. This occurs because the high stress in the keyway is due to the contact between the key and the shaft and the total force carried by the key is directly proportional to the outer diameter of the shaft. In contrast, the stress on the side of the shaft opposite to the key is under a torsional load instead of a contact load, so the stress in this area depends on the countershaft’s polar moment of inertia and outer diameter. It is important to point out that the effects of bending on the shaft are significantly reduced by the presence of the gears and spacers. Essentially, the gears create an additional resistance to bending deformation on the shaft because pitching of the individual gears would cause them to come into contact with the other gears.
Figure 6-7: Stress Distribution in a Hollow Countershaft with an Outer Diameter of 44.1 mm [1.735 in] and an internal Diameter of 6.35 mm [0.25 in]
The second set of hollow shaft studies demonstrated the effects of increasing the outer diameter of the hollow shaft on the stress distribution in the shaft. There were two simulations carried out one with a small hole and another with a large hole (the material removed from each shaft was in proportion to the material removed from the smaller diameter shafts). The results of these simulations are recorded in Figure 6-9 and Figure 6-10. The maximum stress on the shaft occurs at the outside radius of the keyway underneath the output gear, just as it did in the baseline study and the previous hollow simulations. However, in this set of studies the stress were slightly lower than the stress in smaller diameter shaft. This is likely due to the decrease in the contact force required to produce the torque on the shaft due to the increase in the average diameter of the keyway. The stress on the side of the shaft opposite to the keyway, decreased as a result of the increased diameter and was lower in both cases than the corresponding stress on the lower diameter shafts. The change in stress in this area was about 33%, but because the stresses are already so low in this area the total change in the stress was only 40 MPa [5.8 ksi].
Figure 6-9: Stress Distribution in a Hollow Countershaft with an Outer Diameter of 50.8 mm [2 in] and an internal Diameter of 7.32 mm [0.288 in]

Figure 6-10: Stress Distribution in a Hollow Countershaft with an Outer Diameter of 50.8 mm [2 in] and an internal Diameter of 29.3 mm [1.153 in]
A few more simulations were carried out on the hollow countershaft with even more material removed from the shaft so that trends in the changes in the stresses could be better examined. The numerical results of these simulations are plotted in addition to the results of the four simulations examined previously in this section in Figure 6-11 through Figure 6-13. One can see a very tight grouping between the maximum stress in the baseline and the 44.1 mm [1.735 in] and 50.8 mm [2 in] diameter hollow shafts. The stress on the opposite side of the keyway, however, demonstrates a much larger spacing (percentage wise). The amount of weight which could be removed from the shaft depends on whether one makes the decision based on the stresses on the keyway or based on the stresses elsewhere in the shaft. If the maximum stress is the deciding factor, the weight can only be reduced by about 0.5 kg [1.1 lbs] per shaft or 1.5 kg [3.3 lbs] total. However, if the stress in the keyway is ignored, the weight saved could be upwards of 2 kg [4.4 lbs] per shaft or 6 kg [13.2 lbs] total. The major source of stress on the countershaft in this particular gearbox is the keyway. Using a lower stress alternative to the keyway, like a splined shaft or directly machining the gears onto flanges in the shaft, would allow the designer to take full advantage of the material’s strength to greatly reduce the shaft’s weight.

![Figure 6-11: Maximum Stress in the Countershaft as a Function of the Wall Thickness to Outer Diameter Ratio](image)
In this section the simulations are focused on bimetallic counter shafts. Instead of removing the material from the inside it will be replaced with material which has a much lower density. This may allow for the weight reduction of complicated parts that are difficult to forge in a hollow configuration. The geometry used in these simulations is identical to the geometry used in the
hollow shaft simulations, except the void in the middle of the steel shell is replaced by an aluminum core which is bonded to the steel shell. An example of the mesh used to discretize the bimetallic geometry is given in Figure 6-14.

The results of the simulations of the bimetallic shaft are summarized in the stress contours shown in Figure 6-15 through Figure 6-18. The first two figures depict the stress distribution in the shaft with outer geometry identical to the baseline simulation and the last two figures show the stress distribution in shafts with larger outer diameters. The same trends that were seen in the simulation of the hollow shafts are observed in the bimetallic shaft simulations. Namely, i) the overall stress in the body decreases as the outer diameter is allowed in increase, ii) the overall stress in the body increases as the wall thickness of the steel shell decreases, iii) the maximum stress in the shaft occurs at the outside of the key way just under the output gear, and iv) the maximum stress does not vary significantly as the wall thickness decreases. If the keyway is ignored the stress in the bimetallic shaft is lower than its corresponding hollow shaft, due to the extra support added by the light core material. However, this comes at the cost of increased weight for a given geometry of the steel shell.
Figure 6-15: Stress Distribution in a Bimetallic Countershaft with and Outer Diameter of 44.1 mm [1.735 in] and an internal Diameter of 6.35 mm [0.25 in]

Figure 6-16: Stress Distribution in a Bimetallic Countershaft with and Outer Diameter of 44.1 mm [1.735 in] and an internal Diameter of 25.4 mm [1 in]
Figure 6-17: Stress Distribution in a Bimetallic Countershaft with an Outer Diameter of 50.8 mm [2 in] and an internal Diameter of 7.32 mm [0.288 in]

Figure 6-18: Stress Distribution in a Bimetallic Countershaft with an Outer Diameter of 50.8 mm [2 in] and an internal Diameter of 29.3 mm [1.153 in]
A few additional simulations were run on the bimetallic counter shafts to better determine the trends in the stress and weight of the component. The numerical results are plotted with the numerical results from the simulations which were previously discussed in this section. Figure 6-19 illustrates how the maximum stress in the counter shaft varies as the ratio of the steel wall thickness to the outer diameter of the shaft changes. The trend in these simulations is less clear than it is in the simulations of the other parts in this study, but there appears to be a slight increase in the stress as the wall thickness in the steel shell decreases. The stress on the opposite side of the keyway is plotted against the steel wall thickness to outer diameter ratio in Figure 6-20. The trend in this plot is much clearer and follow a similar path as the other parts simulated in this study. However, it is clear that the material is not being loaded to anywhere near yield strength of typical automotive steel grades, which means that it is not being fully utilized. Based on Figure 6-21 the weight which can be saved by switching from a solid shaft to a bimetallic shaft is about 0.5 kg [1.1 lbs] per shaft, or 1.5 kg [3.3 lbs] total, if the stress at the keyway is made to match the baseline simulations. If the magnitude of the stress concentration is reduced by replacing the key with a lower stress joining technique a great deal more weight can be saved. A better joining technique would allow the material to be loaded to higher stress levels, reducing the need for large shaft and gear dimensions.

![Graph showing maximum stress in the bimetallic countershaft as a function of the wall thickness to outer diameter ratio.](image)

**Figure 6-19:** Maximum Stress in the Bimetallic Countershaf as a Function of the Wall Thickness to Outer Diameter Ratio

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6.5 Conclusions

The weight reduction of the countershaft is particularly attractive because there are frequently more than one in the gearbox. Thus any savings in this part could be doubled or even tripled in some instances. In the gearbox examined in this chapter the combined weight of the countershafts was 10.5 kg or 23.1 lbs. The stress on the countershaft was highly concentrated in the keyway and
was well below the yield strength of typical automotive grades of steel elsewhere in the shaft. This lead to a situation where essentially all of the material was acting in a parasitic fashion. The investigations of the hollow and bimetallic countershafts showed that a weight savings of only 0.5 kg [1.1 lbs] per shaft could be produced by switching to the hollow and bimetallic shafts if the stress in the keyway was used as the deciding factor. However, savings of upwards of 2 kg [4.4 lbs] per shaft could be achieved if the stress on the opposite side of the keyway was being compared to the stress in the same area of the baseline study. Thus, if the stresses in the baseline study are to be maintained a total weigh savings of 1.5 to 6 kg [3.3 to 13.2 lbs] could be achieved. However, it is likely that even greater weight savings could be achieved if the keyway was replaced with a joining method which created a lower stress concentration. By switching to another joining technique, or even machining the gears directly onto flanges forged onto the shaft, the average stress in the shaft could be brought much closer to the yield strength of the material, reducing the total amount of material needed to resist the loads on the countershaft.
Chapter 7

Conclusions and Future Work

In the first section of the report, load maps are created that depict how the loads are transmitted from system to system and part to part. The forces and moments which are applied to the individual heavy duty truck components are evaluated in this section. The next section is composed of several investigations into the weight reduction of several gearbox shafts and the axle shafts. Two lightweight design techniques were examined: i) a hollow shaft geometry, and ii) a bimetallic shaft configuration which utilized a low density core material.

The largest weight savings in the study were achieved by replacing the conventional, solid axle shaft with lightweight axle shafts. The weight savings were very large because the combined weight of the axle shafts in the heavy duty truck in much greater than the weight of the other parts examined. Specifically, 22.5 to 29.4 kg [49.6 to 64.8 lbs] of material can be removed if the hollow shaft design is used and 16 to 19.2 kg [35.2 to 42.2 lbs] can be removed from the vehicle if the bimetallic variant is used. These savings are dependent on the acceptable increase in the shaft’s diameter and feasibility of fabricating the designs.

The lightweight designs for the gear box input shaft also showed some promise, but due to the low weight of the conventional part the actual weight removed from the vehicle is small. The hollow input shaft design saved 1.75 kg [3.85 lbs] of weight and the bimetallic input shaft design reduced the weight of the component by 0.9 kg [2 lbs]. It is important to note that even this small shavings could be an important part of an aggregate weight savings created by making a series of minor weight improvements.

The output shaft has slightly higher prospects for reducing the weight of the heavy duty truck than the input shaft due to its larger initial weight. It was determined that a weight savings of 1.7 to 2.2 kg [3.74 to 4.84 lbs] could be achieved by switching from a solid shaft to a hollow shaft and a savings of 1.29 kg [2.84 lbs] could be achieved if a bimetallic variant of the output shaft was utilized. This is, of course, assuming that the lightweight designs could be fabricated and that a slight increase in the diameter of the shaft is acceptable.
The countershaft examined in this report is harder to make a judgement about than the other power train parts examined due to the high stress concentrations induced by the keyway. It was discovered that a majority of the material in the countershaft in the reference gearbox was carrying a stress which was much less than the yield strength of typical automotive grades of steel. The exception to this generalization is the stress of the material in the keyway which reached very high levels. If the stress is to be maintained at levels near the solid countershaft examined combined weight savings of only 1.5 kg [3.3 lbs] are possible. However, if a method of joining the gears to the shaft was adopted that had a smaller stress concentration than the keyway, it is very likely that the size of the countershaft and gears on the shaft could be significantly reduced. This would result in a better utilization of the strong, but heavy, steel and an overall reduced weight.

Overall, it was determined that by applying these techniques to the investigated parts that a total weight savings of 38.4 kg [86.6 lbs] could be reached. This is, of course, subject to the feasibility of actually fabricating the lightweight designs.

In the next phase of the project will focus on the forging techniques which will be needed to create some of the light weight parts suggested in this report. It will help to determine the feasibility of the designs in light of up to date forging practices and materials as well as identify any potential problems or limitations that may arise during the development of these new heavy duty light weight parts.
References


