Design and Analysis of Power Take-Off (PTO) and Telescopic Driveshafts

By

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Dissertation submitted in partial fulfillment of
the requirements for the
Bachelor of Engineering (Hons)
(Mechanical Engineering)

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CERTIFICATION OF APPROVAL

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Approved by,

____

(Dr. Zainal Ambri Abdul Karim)

UNIVERSITI TEKNOLOGI PETRONAS

TRONOH, PERAK

July 2008

CERTIFICATION OF ORIGINALITY

This is to certify that I am responsible for the work submitted in this project, that the original work is my own except as specified in the references and acknowledgements, and that the original work contained herein have not been undertaken or done by unspecified sources or persons.

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MUHAMMAD ZUKNI BIN IBRAHIM

ABSTRACT

Power take-off (PTO) is widely used in automotive field but not commonly known to most people. PTO and telescopic driveshaft systems are used to transmit power from the engine to external machinery. The PTO system used by Light Fire Rescue Tender (LFRT) manufactured by CME Technologies Sdn Bhd is not the best because it is assembled by three telescopic driveshafts instead of one. The aim of this project is to simulate and analyze the PTO and telescopic driveshafts on Light Fire Rescue Tender trucks to identify area for improvement. Subsequently, this project also attempts to suggest a new design to improve the performance of this system. In achieving the objectives, simulation and analysis of the existing system will be performed and recommendation of a new system which will enhance the performance of the system will be proposed. It is hoped that the Design and Development Division of CME Technologies Sdn Bhd will refer to this research after the completion of the project.

ACKNOWLEDGEMENT

In the name of Allah, the Most Gracious and The Most Merciful.

With a humble heart, thanks to God: The Almighty for giving me the abundant strength to complete this Final Year Project. Without His Nourishing guidance, this project would not be a success.

A huge appreciation goes to Dr. Zainal Ambri Abdul Karim: the supervisor, for his never-ending assistance and supervision towards the project's completion. Without him, the cloudy would not be cleared.

Credit also goes to Mr. Nurzaki Bin Ikhsan, Mr Azman Bin Aziz, and Mr Fahmi Bin Ali, Formula SAE UTP team members who were really helpful in providing assistance with the software involved.

The next gratitude will be flown to my endless loving mother, Mrs. Nooriah Bt Tassim. Without her encouragement, flaws would be abundant.

Again, lot of thank you to all persons involved in this project, direct or indirectly. Only God could repay the deeds that had been showered. Thank you very much.

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LIST OF ABBREVIATIONS

PTO	Power Take-Off
ASAE	American Society of Agricultural Engineers (Society for Engineering in Agricultural, Food, and Biological Systems)
LFRT	Light Fire Rescue Tender

CHAPTER 1

INTRODUCTION

1.1 Background of Projects

Power take off (PTO) is a mechanical device used to transmit engine power to auxiliary equipment. Power take offs can be mounted either on main or auxiliary transmission to draw power from the main engine. If the power take off is operating on the transfer case and its operation is distance dependent, its rotational speed is given in revolutions per meter off distance covered. It can be categorized due to ratio between the output angular speed and engine speed. The auxiliary equipments like water pump or hydraulic pump will be attached to telescopic driveshafts and connected to PTO.

PTO can also be transmission mounted or engine mounted. In transmission mounted PTO, it is located on the side, bottom or rear of the transmission. For manual transmissions the PTO is generally driven from a countershaft gear, but can also be found being driven by the reverse idler gear. For automatic transmissions, the PTO will either be driven before the torque converter and be subjected to torque converter slip. In engine mounted PTO, it is located at the rear of the engine. The PTO can be driven from the timing gears, or a special gear train, and provides a constant drive for special applications.

The project undertaken here will focus on the PTO and telescopic driveshafts used by Light Fire Rescue Tender (LFRT) manufactured by CME Technologies Sdn Bhd where the author has gone for 32-weeks of industrial internship. This system is used to draw power taken from engine via PTO and telescopic driveshafts and driven the vehicle mounted water pump. Figure 1.1 shows the PTO mounted on the gearbox while Figure 1.2 shows the assembly of the driveshafts.

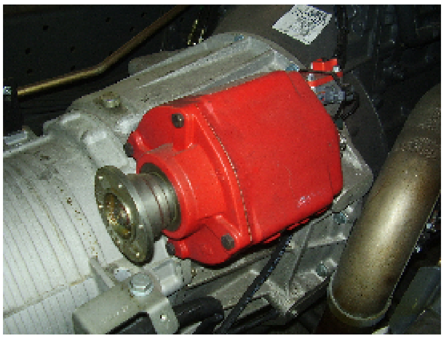


Figure 1.1: PTO

Figure 1.2: Propeller shafts assembly

1.2 Problem Statements

The PTO in LFRT is located at the side of the transmission box of the engine and the pump is located at the rear of the chassis. The PTO location is slightly offset and not perpendicular to the pump location. The pump connected to the PTO through an assembly of three telescopic driveshafts instead of one. This arrangement has caused the angular movement of the driveshafts to impose higher friction than if it is connected straightly. The project will simulate the shafts work and analyze the load, stress and works produced by this system. Appendix 1-1 shows the layout of the shafts assembly.

1.3 Significance of Study

The findings of the project would be offered as an option to the Design and Development Division (DDD) of CME Technologies Sdn Bhd in their design implementation for future projects and optimizes the performance of the water pumps. It will also increase their engineering knowledge and would be useful for tender proposal.

Students in engineering field also can refer to this projects in the future because the author believe that there are not many thesis produced by Malaysians in the area of PTO systems especially when it applies to trucks.

1.4 Objectives

The project aims to propose a new design that will give better efficiency and performance but still within the design and space limits of the LFRT specifications. In order to achieve this, the following objectives must be accomplished:

1. To simulate and analyze the power transmission from the main engine to the pump of the existing system.
2. To design and simulate an improved system with higher efficiency and performance of the PTO.

1.5 Scope of Study

In the first semester, efforts were focused on the literature review, data collection, simulation and analysis of the existing system. An interim report were produced to summarize all the tasks accomplished during the first semester and to identify future works. In the second semester, the author proposes a new design with higher efficiency. Software used to achieve these objectives is CATIA.

CHAPTER 2

LITERATURE REVIEW

2.1 Power Take-Off (PTO)

The Power Take-Off (PTO) shaft is an efficient means of transferring mechanical power between engine and implements. Figure 2.1 is a diagram of component parts of an implement PTO. Two typical PTO system arrangements are shown. The top drawing is of a PTO system involving a pedestal connection, such as is found on many types of pulled machinery (hay balers, forage choppers, large rotary mowers, etc.). The lower drawing is of a PTO system where the implement's input driveline connects directly to the tractor PTO stub. Examples of this type of connection include three-point hitch mounted equipment (post hole diggers, small rotary mowers, etc.) and augers.

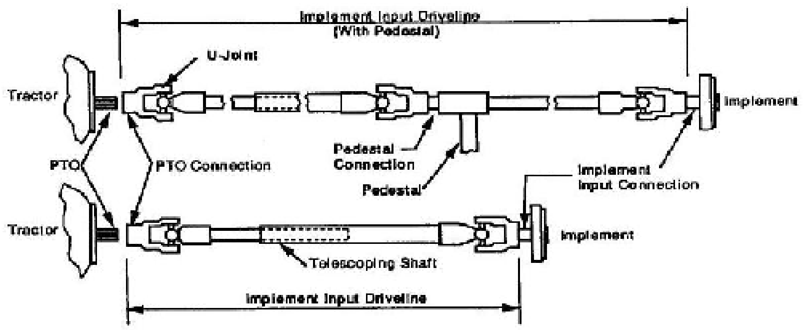


Figure 2.1: Major components of PTO system [4]

American Society of Agricultural Engineers (ASAE) defines terminology for three PTO configurations as continuous-running, independent and transmission-driven. For continuous-running, the power to operate both transmission and power take-off is transmitted through a master clutch. Both operate only when the master clutch is engaged. Auxiliary means are provided for stopping the travel of the tractor without stopping the power take-off. The continuous-running power take-off ceases to operate at any time when the master clutch is disengaged. For independent, the power to operate transmission and power take-off is transmitted through independent clutches. Travel of the tractor may be started or stopped by the operation of the transmission clutches without affecting the operation of independent power take-off. Likewise, the power take-off may be started or stopped without affecting tractor travel. For transmission-driven, the power to operate transmission and power take-off is transmitted through a master clutch, which serves primarily as traction clutch. The power take-off only operates when the master clutch is engaged. The transmission-driven power take-off ceases to operate at any time the master clutch is disengaged. [4]

2.2 Introduction to shaft

Shafts are used in a variety of ways in all types of mechanical equipment. A *shaft*, usually a slender member of round cross section, rotates and transmits power or motion. However, a shaft can have a noncircular cross section and need not be rotating. An *axle*, a nonrotating member that carries no torque, is used to support rotating members. A *spindle* designates a short shaft. A *flexible shaft* transmits motion between two points (e.g., motor and machine), where rotational axes are at an angle with respect to one another.

Most shafts are under fluctuating loads of combined bending and torsion with various degrees of stress concentration. Many shafts are not subjected to shock or impact loading; however some applications arise where such load takes place. Thus, the associated consideration of static strength, fatigue strength, and reliability play a significant role in shaft design. A shaft designed from the preceeding viewpoint satisfies strength requirements.

2.2.1 Theory of Designing shafts

In the design of circular slender shafts that transmit power at a specified speed, the material and the dimensions of the cross section are selected not to exceed the allowable shearing stress or a limiting angle of twist when rotating. Therefore, a designer needs to know the torque acting on the power-transmitting shaft. Equation below may be used to convert the power supplied to the shaft into a constant torque exerted on it during rotation. [2]

$$Power(kW) = \frac{Tn}{9549}$$

Where

T = torque (N.m)

n = shaft speed (rpm)

After having determined the torque to be transmitted, the design of circular shaft to meet strength requirements can be accomplished.

1. Assume that, as is often the case, shear stress is closely associated with failure.
2. An important value of the shear stress is defined by $t_{max} = Tc / J$.
3. The maximum usable value of t_{max} without failure is the yield shear strength S_{ys} or ultimate shear strength S_{us}.
4. A factor of safety n is applied to t_{max} to determine the allowable stress $t_{all} = S_{ys}/n$ or $t_{all} = S_{us} / n$. The required parameter J / c of the shaft based on the strength specification of is

$$\frac{J}{c} = \frac{T}{\tau_{all}}$$

Where

J = moment of inertia (m^4)

c = outer diameter of the shaft (m)

T = torque (N.m)

t_{all} = allowable stress (MPa)

2.2.2 Driveshaft

Drive shafts as power transmission tubing are used in many applications, including cooling towers, pumping sets, aerospace, trucks and automobiles. In metallic shaft design, knowing the torque and the allowable shear stress for the material, the size of the shaft's cross section can be determined. Metallic drive shaft has the limitations of weight, low critical speed and vibration characteristics. Composite drive shafts have solved many automotive and industrial problems accompanying the usage of the conventional metal ones because the performance is limited due to lower critical speed, weight, fatigue and vibration.

"When the length of steel drive shaft is beyond 1500 mm, it is manufactured in two pieces to increase the fundamental natural frequency, which is inversely proportional to the square length and proportional to the square root of specific modulus." (Badie, 2006).

An efficient design of composite drive shaft could be achieved by selecting the proper variables, which can be identified for safe structure against failure and to meet the performance requirements. As the length and outer radius of drive shafts in automotive applications are limited due to spacing, the design variables include the inside radius, layers thickness, number of layers, fiber orientation angle and layers stacking sequence. In optimal design of the drive shaft these variables are constrained by the lateral natural frequency, torsional vibration, torsional strength and torsional buckling.

2.3 Material Selection

To minimize deflections, *shaft materials* are generally cold-drawn or machined from hot-rolled, plain-carbon steel. The shaft end should be made with chamfers to facilitate forcing on the mounted parts and avoid denting the surfaces. Cold drawing improves the physical properties. It raises considerably the values of ultimate tensile and yield strengths of steel. Where toughness, shock resistance, and greater strength are needed, alloy steels are used. The foregoing materials can be heated to produce the desire properties. If the service requirements demand resistance to wear rather than extreme strength, it is customary to harden only the surface of the shaft, and carburizing grade steel can be used. Note that the hardening treatment is applied to those surfaces requiring it; the remainder of the shaft is left in the original conditions.

2.3.1 Steel properties

Steel is an alloy consisting mostly of iron, with a carbon content between 0.2 and 2.04% by weight (C:1000–10,8.67Fe), depending on grade. Carbon is the most cost-effective alloying material for iron, but various other alloying elements are used such as manganese, chromium, vanadium, and tungsten. Carbon and other elements act as a hardening agent, preventing dislocations in the iron atom crystal lattice from sliding past one another. Varying the amount of alloying elements and form of their presence in the steel (solute elements, precipitated phase) controls qualities such as the hardness, ductility and tensile strength of the resulting steel. Steel with increased carbon content can be made harder and stronger than iron, but is also more brittle. Table 2.1 shows the material properties of steel and othermetal

Properties	Carbon Steels	Alloy Steels	Stainless Steel	Tool Steels
Density (1000 kg/m^3)	7.85	7.85	7.75-8.1	7.72-8.0
Elastic Modulus (GPa)	190-210	190-210	190-210	190-210
Poisson's Ratio	0.27-0.3	0.27-0.3	0.27-0.3	0.27-0.3
Thermal Expansion (10^{-6}/K)	11-16.6	9.0-15	9.0-20.7	9.4-15.1
Melting Point (°C)			1371-1454	
Thermal Conductivity (W/m-K)	24.3-65.2	26-48.6	11.2-36.7	19.9-48.3
Specific Heat (J/kg-K)	450-2081	452-1499	420-500	
Electrical Resistivity (10^{-9}W-m)	130-1250	210-1251	75.7-1020	
Tensile Strength (MPa)	276-1882	758-1882	515-827	640-2000
Yield Strength (MPa)	186-758	366-1793	207-552	380-440
Percent Elongation (%)	10-32	4-31	12-40	5-25
Hardness(Brinell 3000kg)	86-388	149-627	137-595	210-620

Table 2.1: Properties of steel [1]

2.3.2 Composite properties

The nature of composites with their higher specific modulus (modulus to density), which in carbon/epoxy exceed four times that of aluminum, enables the replacement of the two pieces metal shaft by one piece composite one which resonate at higher speed and so keeping higher margin of safety. A drive shaft of composites offers excellent vibration damping, cabin comfort, reduction of wear on drive train components and increasing tires traction. Polymer matrix composites such as carbon/epoxy or glass/epoxy offer better fatigue characteristics as micro cracks in the resin not growth further like metals but terminated at the holes of fibers. Generally composites have less susceptibility to the effect of stress concentration such as those caused by notches and holes, than metals.

2.4 Torsional Analysis

Mechanical power transmission systems are often subjected to static or periodic torsional loading that necessitates the analysis of the torsional characteristics of the system. For example, the drive train of a typical automobile is subjected to a periodically varying torque. This torque variation occurs due to the cyclical nature of the internal combustion engine that supplies the power. If the frequency of the engine's torque variation matches one of the resonance frequencies of the engine/drive train system, large torsional deflections and internal shear stresses can occur. Continued operation of the machinery under such conditions can lead to early fatigue failure of system components. Thus, an engineer designing such a system needs to able to predict its torsional natural frequencies and be able to easily determine what effects design changes might have on those natural frequencies.

CHAPTER 3
METHODOLOGY

3.1 Simulation Tools

There will be no prototype produced at the end of this research. The following software will be used to accomplish the project. CATIA will be used to build real model of the telescopic driveshafts. Analysis can be done using CATIA where the required moment will be applied and the Von Mises Stress development and deflection will be observed.

3.1.1 3-D Model

3-D models of the driveshafts were built using the CATIA software. The models were disassembled into several parts to ease the process of the design. First of all, the models were design in the Part Design of CATIA. After all the parts have been complete, they are assembled in Assembly Design.

Here, all parts are combined into product space and the orientation of each part is rotated into desired orientation.

Then, after the assembly is completed, the products were converted back into part to define the material. For example, the front driveshafts consist of 4 parts that is main shafts, telescopic part, end of shafts and universal joint.

Each of these parts was built in the Part Design, and then these 4 parts were assembling in Assembly Design. After assembling and converting back into parts, the middle and rear driveshafts were built using the same technique.

3.1.2 Analysis

Analysis was conducted using CATIA software. After the models are completed in design, the material of these driveshafts was defined. In this project, the materials used are steels. Then, the models were exported into Analysis and Simulation section and generate structural analysis. The models were meshed into desired length and faces of the models that wanted to be analyzed were selected. Then, the torque was applied at a selected point and the software simulated the deformation, Von Mises Stress and displacement of the models.

3.2 Data Collection

The information of the LFRT has been collected during week 8 of semester 1. The detail is as below:

Date: 12th March 2008

Time: 11.00 a.m.

Venue: CME Technologies Sdn Bhd

The main data needed was the design of the telescopic driveshafts assembly and its dimension. It is because the model in CATIA requirement all of this information for the simulation. In addition, the specification for PTO and water also was collected for reference in the future. The verification letter is included in Appendix 3-1.

3.3 Project Planning

Gantt charts for both semesters are included in Figure 3.1 and Figure 3.2. The charts show the activities planned to complete this project in the allowed timeframe.

Milestone for first semester

No	Detail/Week	1	2	3	4	5	6	7		8	9	10	11	12	13	14
1	Selection of Project Topic															
2	Preliminary Research Work															
3	Literature Review															
4	Submission of Preliminary Report															
5	Seminar 1															
7	Software Training															
8	Document Preparation															
9	Data Collection															
10	Submission of Progress Report															
11	Seminar 2															
13	Simulation															
14	Analysis															
15	Submission of Interim Report Final Draft															
16	Oral Presentation															

	Process
	Milestone

Figure 3.1: Gantt chart Semester 1

Milestone for second semester

No	Detail/Week	1	2	3	4	5	6	7	8	9	10	11	12	13	14
1	Project work continue														
	Theory of designing														
	Calculation														
	Modeling of new design														
	Analysis of new design														
	Comparing existing and new design														
	Report preparation														
2	Seminar														
3	Poster Exhibition														

Process	

Figure 3.2: Gantt chart Semester 2

3.3 Research Flow

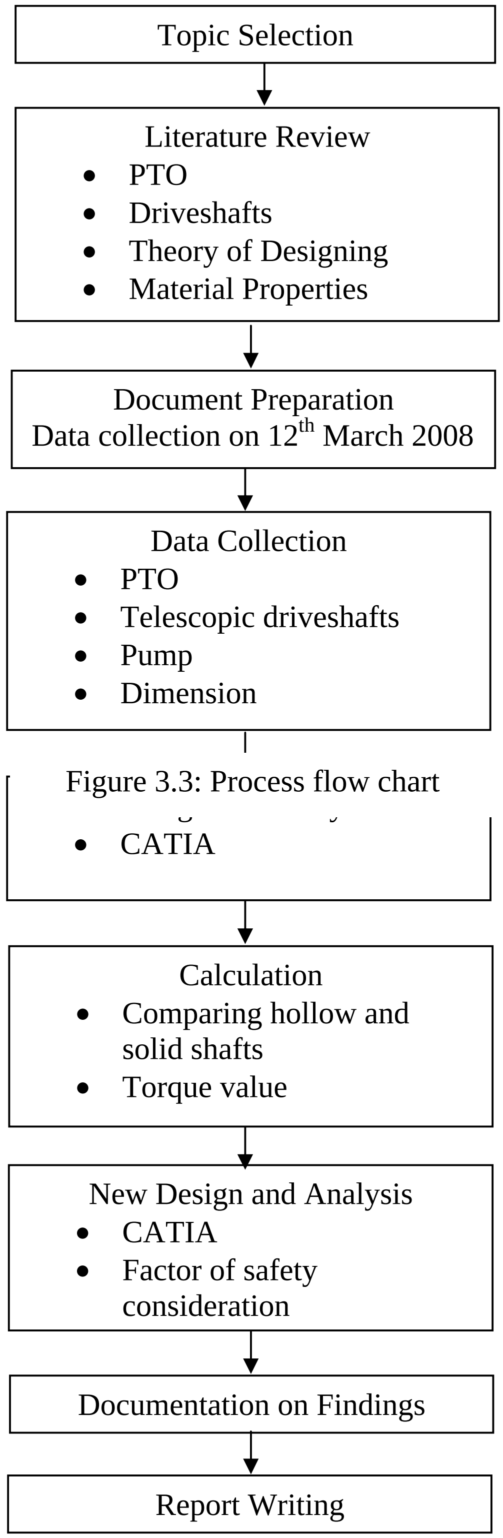
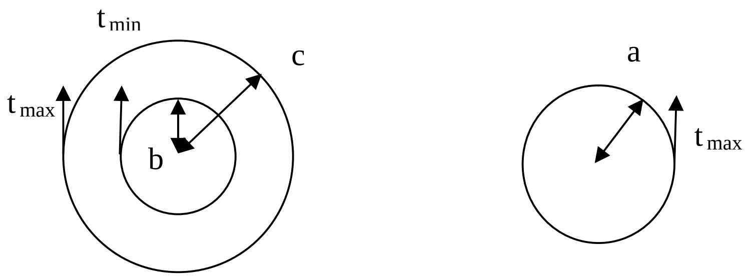


Figure 3.3: Process flow chart

CHAPTER 4

RESULTS AND DISCUSSION

4.1 Torque Transmission Efficiency of Hollow and Solid Shafts



Hollow shaft   Solid shaft

Given

$$\frac{c}{b} = \frac{45}{40}$$

$c = 1.125b$

<u>Solution</u>

The maximum shear stress t_{max} equals $t_{all.}$ Since the cross sectional areas of both shafts are identical,

$p\ (c^2 - b^2) = p\ a^2$

$a^2 = c^2 - b^2$

General equation for shaft

$$T = \frac{J}{c}\tau_{all}$$

For solid shafts

$$J_{solid} = \frac{\pi}{2} a^4$$

For hollow shafts

$$J_{hollow} = \frac{\pi}{2}(c^4 \quad b^4)$$

So, for solid shafts:

$$T_{solid} = \frac{\pi}{2} a^3 \tau_{all}$$

And for hollow shafts:

$$T_{hollow} = \frac{\pi}{2c}(c^4 \quad b^4)\tau_{all}$$

Dividing this 2 equations give:

$$\frac{T_{hollow}}{T_{solid}} = \frac{c^4 - b^4}{ca^3}$$

and $a^2 = c^2 - b^2$:

$$\frac{T_{hollow}}{T_{solid}} = \frac{c^4 - b^4}{c(c^2 - b^2)^{\frac{3}{2}}}$$

Substitute c = 1.125b gives:

$$\frac{T_{hollow}}{T_{solid}} = \frac{(1.125b)^4 - b^4}{(1.125b)\left[((1.125b)^2 - b^2)^{\frac{3}{2}}\right]}$$

$$\frac{T_{hollow}}{T_{solid}} = \frac{1.602b^4 - b^4}{0.154b^4} = 3.91$$

Result shows that hollow shafts are more efficient in transmitting torque than solid shafts. Interestingly, thin shafts are also useful for creating an essentially uniform shear. However, to avoid buckling, the wall thickness can’t be excessively thin.

4.2 PTO Calculation

Using data obtained from manufacturer, the torque rating can be calculated.

Standard Output Shaft Size	1.5" 10 spline with 1410 flange
Intermittent Torque Rating	575lbs.ft.
Horsepower Rating for Intermittent Service	
at 500RPM	54.7HP
at 1000RPM	109.4HP
Approximate Weight	66lbs

For the fire truck, PTO normally operated at 2500RPM

Using linear regression, horsepower rating is 273.5HP at 2500RPM and

$$1\text{HP} = 0.7457\ \text{kW}$$

$$273.5\text{HP} = 203.9\text{kW}$$

Power Equation:

$$kW = \frac{T.n}{9549}$$

Where

T = torque (n.m)

n = shaft speed (rpm)

so,

$$T = \frac{9549(kW)}{n}$$

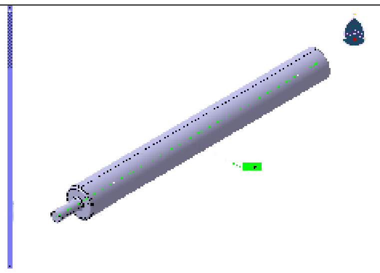
$$T = \frac{9549(203.9)}{2500}$$

$$T = 779N.m$$

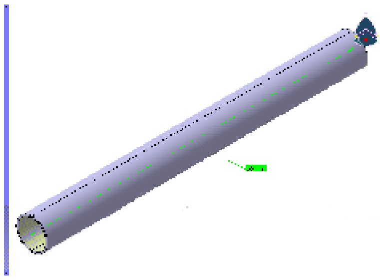
The torque rating obtained will be used in analysis to simulate the effect on shafts. This value of moments will be added in Finite Element Analysis in CATIA software to see the Von Mises Stress development.

4.3 Existing Model (Part)

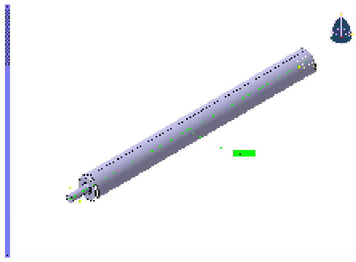
4.3.1 Front shaft

 Figure 4.1 : Front Driveshaft	Front driveshaft is the one that connects PTO to the middle driveshaft. It has one side of telescopic part that can vary its length and the end part can rotate due to the angle of assembly. The detail drawing is attached in Appendix 4-1.

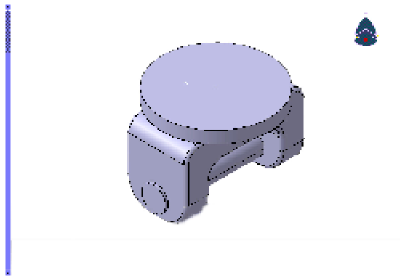
4.3.2 Middle shaft

 Figure 4.2 : Middle Driveshaft	Middle driveshaft is used to connect the front driveshaft to the rear driveshaft. It is a static driveshaft without any telescopic part. The detail drawing is attached in Appendix 4-2.

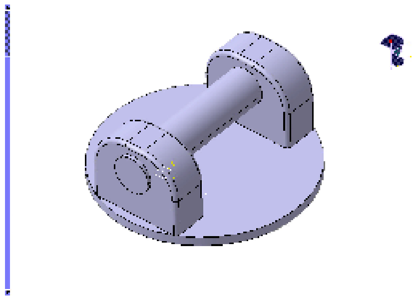
4.3.3 Rear shaft

 Figure 4.3 : Rear Driveshaft	Rear driveshaft is the one that connects pump to the middle driveshaft. It is almost the same as the front driveshafts. The detail drawing is attached in Appendix 4-3.

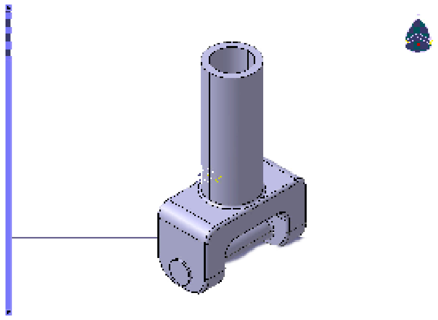
4.3.4 End of shaft

 Figure 4.4 : End of shaft	End of shaft is attached to one or both ends of the shaft based on its location. Then, it will be connected to the universal joint part. The detail drawing is attached in Appendix 4-4.

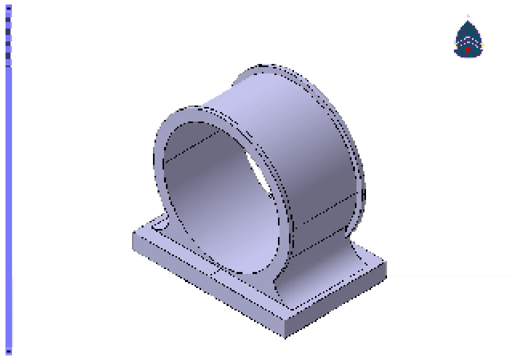
4.3.5 Universal joint

 Figure 4.5 : Universal joint	Universal joint is used to connect from one driveshaft to another. It can be rotated horizontal and vertically. The detail drawing is attached in Appendix 4-5.

4.3.6 Telescopic part

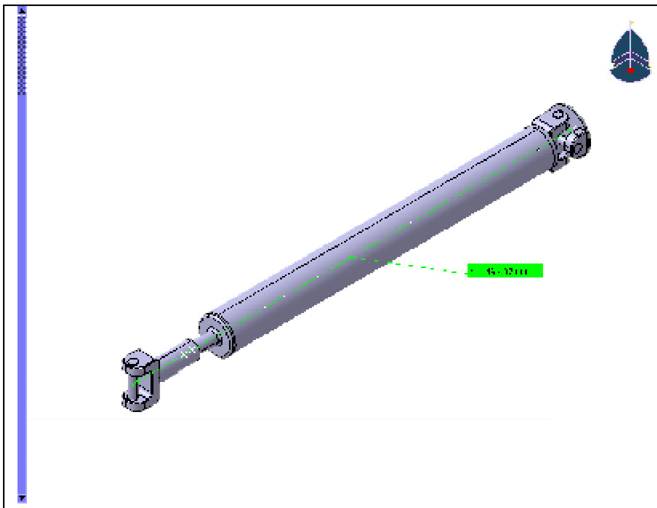
 Figure 4.6 : Telescopic part	Telescopic part is the part where the overall shaft length can vary according to the equipment and chassis length. The detail drawing is attached in Appendix 4-6.

4.3.7 Middle shaft mounting

 Figure 4.7 : Middle shaft mounting	Middle shaft mounting is used in middle shaft and mounted into the chassis. The front and rear do not need the mounting because the front shaft is connected to PTO while the rear to the pump. The detail drawing is attached in Appendix 4-7.

4.4 Existing Model (Assembly)

4.4.1 Front driveshaft assembly

 Figure 4.8: Front driveshaft assembly	The front driveshaft assembly consists of 4 parts that is the main shaft, telescopic part, end of shaft and universal joint. The telescopic will be connected to PTO and the universal joint will be connected to the middle driveshafts. The detail drawing is attached in Appendix 4-8.

4.4.2 Middle driveshaft assembly

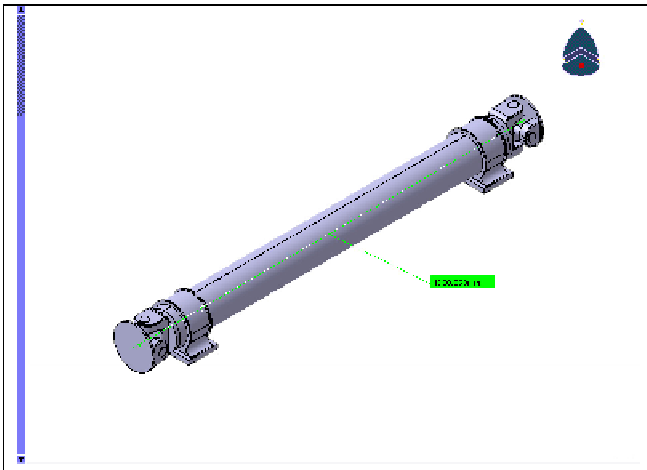


Figure 4.9: Middle driveshaft assembly

The front driveshaft assembly consists of 7 parts that are the main shaft, 2 mounting parts, 2 ends of shaft and 2 universal joints. This middle driveshaft will connect the front and rear driveshafts and the mounting parts will be mounted on the chassis. The detail drawing is attached in Appendix 4-9.

4.4.3 Rear driveshaft assembly

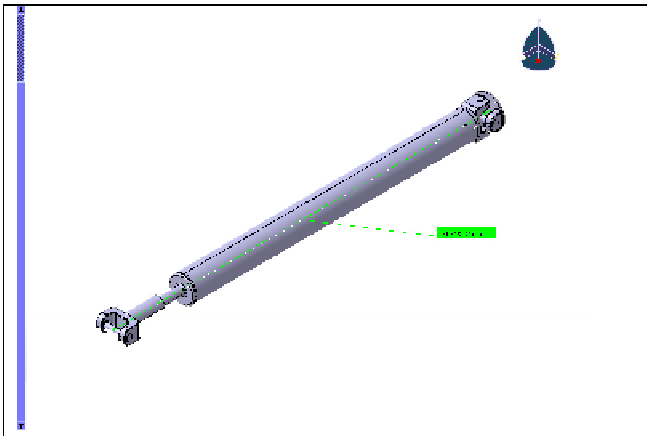


Figure 4.10: Rear driveshaft assembly

The rear driveshaft assembly consists of 4 parts that are the main shaft, telescopic part, end of shaft and universal joint. The telescopic will be connected to the pump and the universal joint will be connected to the middle driveshafts. The detail drawing is attached in Appendix 4-10.

4.4.4 Assembly of existing model

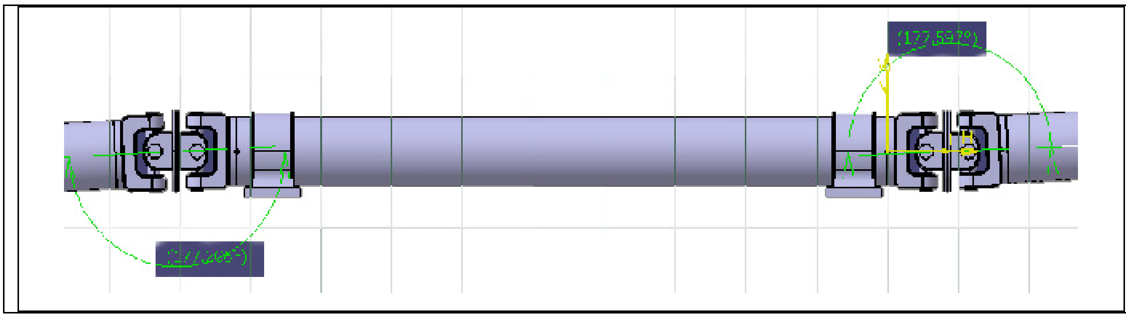


Figure 4.11: Driveshafts Assembly

All 3 shafts are assembled as shown in the Figure 4.11 above. It represents the real assembly of the driveshafts in LFRT.

4.5 Analysis of Existing System

Based on the calculation that was made before and the data obtained from reference book, structural analysis is made in CATIA software. The 779N.m torque value is obtained from previous calculation. The torque is applied to each of the 3 shafts to observe the outcome of the analysis. Von Mises Stress and displacement is obtained from the analysis.

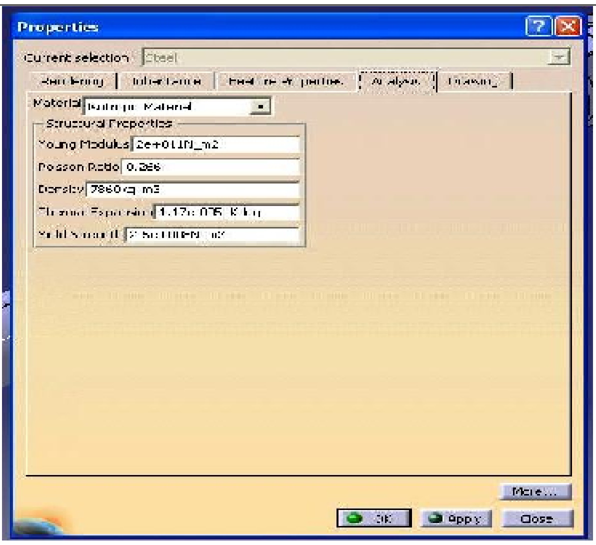


Figure 4.12: Steel Properties

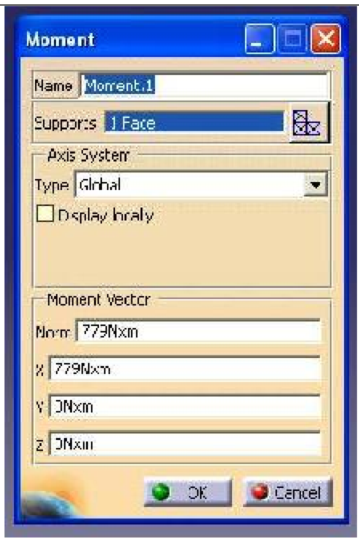


Figure 4.13: Moment value

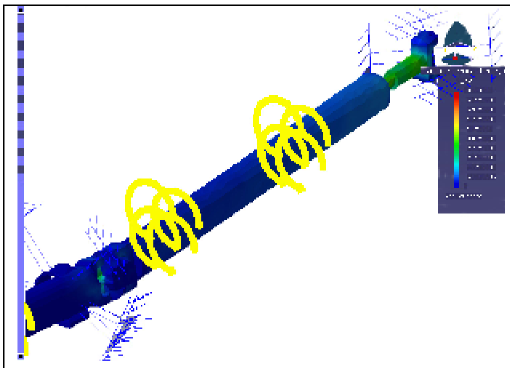


Figure 4.14: Rear shaft

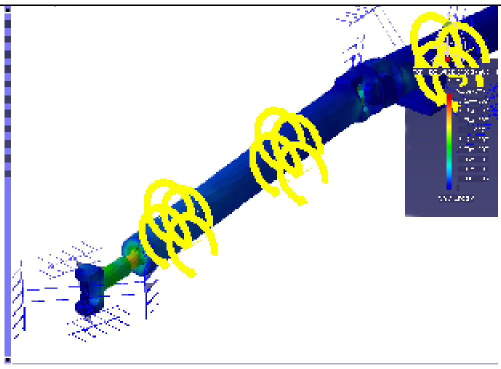


Figure 4.15: Front shaft

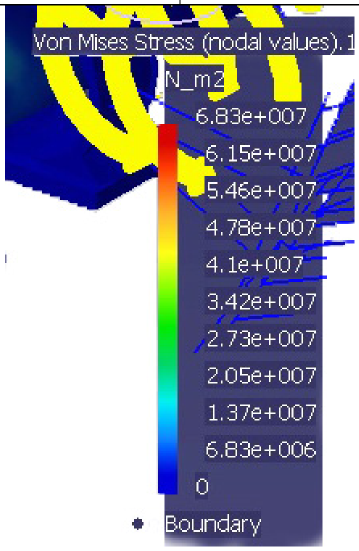


Figure 4.16: Von Mises Stress for existing system

From analysis of the assembled shaft, the highest stress developed is 6.83×10^7N/m. From the Figure 4.14 and Figure 4.15 above, the critical part in the shaft is the telescopic part and universal joint, but it is shown that none of the above parts experience the plastic deformation. Figure 4.16 shows the value of Von Mises Stress obtained from Figure 4.14 and Figure 4.15. The value is relevant for all parts of the driveshafts assembly of the existing model.

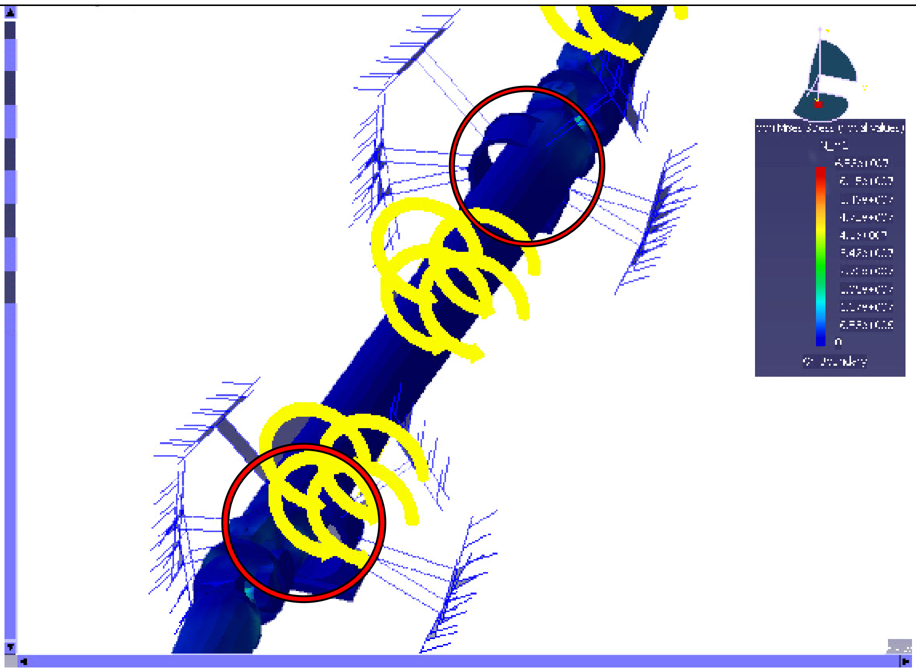


Figure 4.17: Middle shaft analysis

In Figure 4.17 above, it is observed that at the mounting of the middle shafts, the shaft tend to rotate out of the bearing ring. It is shown by the above red circle. It is the result of the assembly with gradient between the driveshafts.

Power to Weight Ratio

Power = 203.9kW

Weight = 50.931kg (obtained from CATIA)

$$ratio = \frac{203.9kW}{50.931kg}$$

$$ratio = 4kW / kg$$

This ratio will be used in the calculation to compare with new optimizes system.

4.6 New system dimension

Using shafts equation:

$$\frac{J}{c} = \frac{T}{\tau_{all}}$$

Where

J = moment of inertia (m^4)

c = outer diameter of the shaft (m)

T = torque (N.m)

t_{all} = allowable stress (Mpa)

For hollow shafts and factor of safety 4:

$$\frac{\pi}{2}\left[\frac{\left(c^4 - b^4\right)}{c}\right] = \frac{779(10)}{300\left(10^6\right)}$$

Substitute c = 1.25b (this value is selected to avoid buckling):

$$\frac{\pi}{2}\left[\frac{\left(2.4414b^4 - b^4\right)}{1.25b}\right] = 2.5967\left(10^{-5}\right)$$

$$\left[\frac{1.4414b^4}{1.25b}\right] = 1.6531\left(10^{-5}\right)$$

$$1.1531b^3 = 1.6531\left(10^{-5}\right)$$

$$b = \sqrt[3]{1.434\left(10^{-5}\right)}$$

$$b = 0.0245m \quad 24.5mm$$

$$d_i = 49mm$$

Substitute b = 24.5mm in c = 1.25b equation:

$$c = 0.0306m \quad 61.2mm$$

$$d_o = 61.2mm$$

Safety factor is 4 because of this consideration:

k_1 = 10 = safety coefficient

k_2 = 10 = vibration coefficient because system in dynamic.

$$k = \times\sqrt{(k_1 \quad k_2)}$$

$$k \approx 10$$

So, the inner diameter for new shafts will be 49mm and outer diameter will be 61.2mm.

4.7 Analysis of new model

For the model, the diameter is reduced to the new dimension as calculated, but the length remains the same as before. Then, the following discusses the outcome of new system.

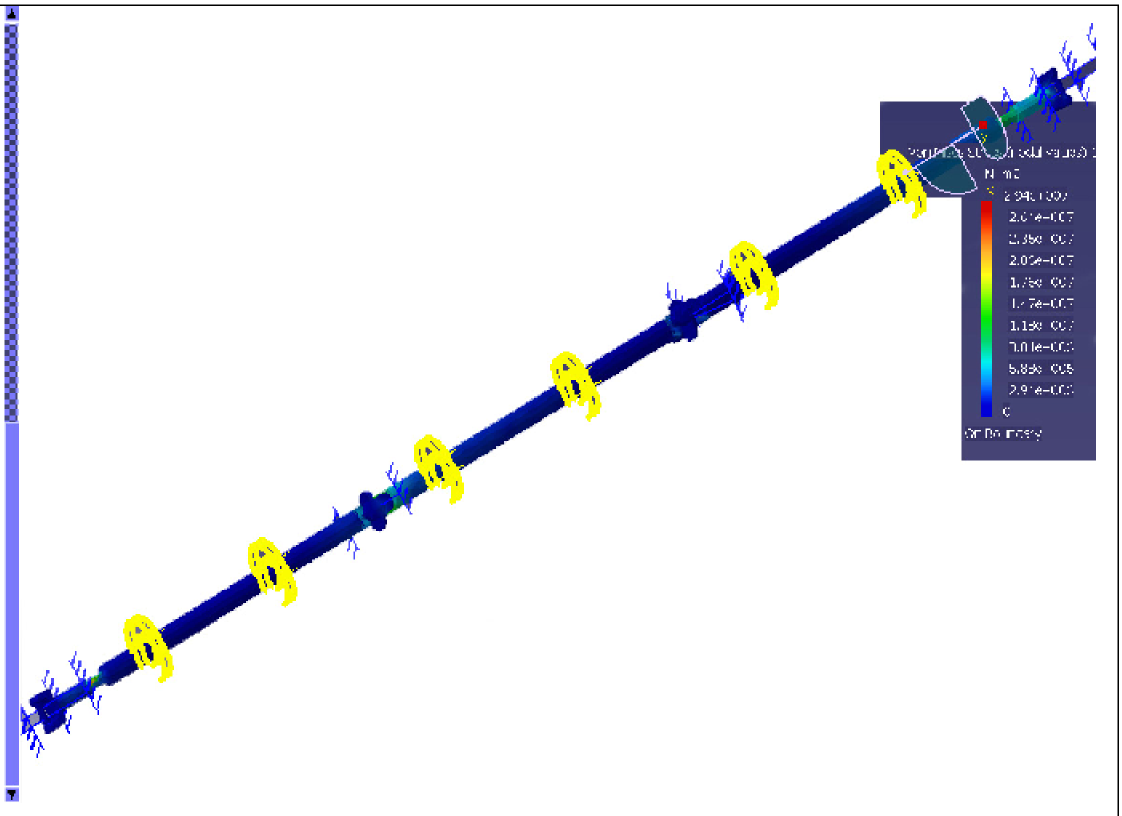


Figure 4.18: Analysis of new system

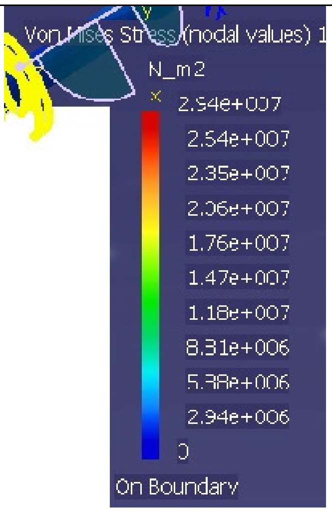


Figure 4.19 Von Mises Stress for new system

Figure 4.18 above shows that the analysis of the shaft after the moment has been applied to all 3 shafts. It is indicated by colour yellow. It shows that even though the same amount of moment is applied, this new shafts does not tend to rotate off its path. This is because all 3 shafts are axially linear. It also shows that the maximum Von Mises Stress is 2.94 x 10^7N/m. It is shown in Figure 4.19 above. This result will be discussed further.

<u>Power to Weight Ratio</u>

Power = 203.9kW

Weight = 30.713kg (obtained from CATIA)

$$ratio = \frac{203.9kW}{30.713kg}$$

$$ratio = 6.6kW / kg$$

4.8 Comparing the data

4.8.1 Power-to-Weight ratio

For the existing system, the power to weight ratio is 4kW/kg while for the new system design is 6.6kW/kg. To find the percentage increase:

$$Increment[\%] = \frac{6.6 - 4}{4} \times 100$$

$$Increment[\%] = 65\%$$

4.8.2 Von Mises Stress

Maximum Von Mises Stress in existing system: 6.83×10^7N/m

Maximum Von Mises Stress in new system : 2.94×10^7N/m

$$Reduction[\%] = \frac{\left|2.94(10^7) - 6.83(10^7)\right|}{6.83(10^7)} \times 100$$

$$Reduction[\%] = 56.95\%$$

From both calculations, it shows that the new system design is optimized in term of 3 data that's weight reduction, Von Misses Stress reduction and the observation of the driveshafts which can contribute to improve the existing system. The weight of the driveshafts was reduced from 50.931kg to 30.713kg. The joining between 3 shafts also did not tend to rotate out from the permitted range.

CHAPTER 5

CONCLUSION AND RECOMMENDATION

5.1 Conclusion

The project has accomplished the task of designing and optimizing of the existing system. It also has achieved both of the objectives which are:

1. To simulate and analyze the power transmission from the main engine to the pump of the existing system.
2. To design and simulate an improved system with higher efficiency and performance of the PTO.

The models of existing system have been built in CATIA and analysis has been conducted. The results were used in fulfilling the second objective.

From the new design model, it has been proved that it can be optimized to a more efficient system. Comparing existing and new system, the power-to-weight ratio has increased 65% because of the shafts weight have been reduced from 50.931kg to 30.713kg. The Von Mises Stress also has been reduced 56.95% with this new system.

5.2 Recommendation

It would be beneficial to CME Technologies to review the outcome of this research in the LFRT pump system. It is recommended that this research will be continued using more variables like materials, thickness ratio and factor of safety to observe the outcome.

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Appendices