VALMAR

Fertilizer Spreader Gearbox Redesign

Team 7

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University of Manitoba
December 6th, 2011

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LETTER OF TRANSMITTAL

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December 6th, 2011

Dennis Rice
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Valmar Airflo Incorporated
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Dear Mr. Rice,

In response to Valmar Airflo Inc's 7600 pull type fertilizer spread gearbox redesign request we present the disclosed report. This report is submitted by Team 7 on December 6, 2011, entitled "Fertilizer Spreader Gearbox Redesign".

This report was composed to evaluate a fertilizer spreader gearbox that adjusts the delivery rate of a fertilizer product. This report includes background information on the product, the project definition, target specifications of the redesign, a final design, and our recommendations. The analysis and development undertaken by the team throughout this project is also rendered.

Team 7 would like to thank the following people for their time and help with this report: Dennis Rice – Research and Development at Valmar Airflo Incorporated; Christian Caron – Manufacturing Coordinator at Valmar Airflo Incorporated; Mercedes Caron – President of Valmar Airflo Incorporated; Dr. Qingjin Peng – Professor at the University of Manitoba; Dr. Paul Labossiere – Professor at the University of Manitoba; Aiden Topping – Sessional Instructor at the University of Manitoba.

Sincerely,

Henry Cortens	Steven Denby
James Kehler	Guangming Yang

Team 7

Enclosed One ["Valmar Fertilizer Spreader Gearbox Redesign"]

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Abstract

Team 7 has been commissioned by Valmar Airflo Incorporated to propose a suitable redesign of the 7600 pull type fertilizer spreader gearbox. The following report presents the full redesign process undertaken.

Three main problems were highlighted by Valmar to be addressed in the redesign. The first was the gear shifting mechanism which was malfunctioning. The gear engagement key design of the shifter was improved. The second issue was inadequate sealing of the gearbox cover which allowed oil to leak. More secure bolting at the lid to case interface, as well as better assembly and updates to the case lip are proposed. The third problem was the overall mass of the design. The gearbox needs to be lighter for easier assembly and mounting. The gear thickness was reduced and the overall case length decreased to help save weight without any sacrifice to performance or reliability.

Team 7 took the gearbox and analyzed the possible ways to address the problems presented. Each section provides a revision to be done to the production of the gearbox. A thorough cost analysis was performed on the new design and it was estimated that the redesign will cost 6.14% less than the current box. This combination of cost reduction and the many improved gearbox characteristics satisfy both the customer and Valmar Airflo Incorporated.

Therefore, Team 7 recommends implementing the gearbox redesign for future production variants.

1.0 Introduction

Team 7 has been commissioned by Valmar Airflo Incorporated to complete a redesign of the fertilizer spreader gearbox manufactured. There are three primary issues with the gearbox. These three problem areas are the gear shifting mechanism, oil leakage, and overall excessive weight. Various aspects of these problems are addressed throughout the design.

Valmar produces approximately twenty fertilizer spreader gearboxes a year. Since this product is produced in low quantities, Valmar wishes to fix the problems in their design without drastically increasing manufacturing or retooling costs. Additionally, the amount of employee training required to produce the new gearbox needs to remain at a minimum. A redesign of the product could fix the issues the gearbox has and provide a benefit not only to the customers of Valmar, but to Valmar's manufacturing group itself.

The project scope includes the redesign of the fertilizer spreader gearbox for Valmar to provide a better product in both manufacturing and sales. New designs could help Valmar reclaim the time and money for maintaining production for an item of such a limited demand. Any retooling and fabrication methods must be adapted for the new design, but remain feasible for production. Valmar requested that the costs of manufacture remain low. The fixes should be easy to implement and maintain for all future fabricated models.

1.1. Background

Valmar Airflo Incorporated is a company that manufactures farming equipment. The gearbox of our project is important to the company because it is used on one of their flagship pieces of machinery. Valmar requires an improvement to the design of a gearbox which is used on the 7600 Pull Type air boom fertilizer spreader, as shown in Figure 1. This gearbox is the major component responsible for adjusting the delivery rate of fertilizer product. An oil bath is used for lubrication and to minimize corrosion, and welded steel construction is used to manufacture the gear case. The shifting mechanism consists of several shafts, a tube and two spring-loaded keys. The shaft runs through the tube which runs through the top gear cluster, and a spring-loaded key is installed in the shaft which engages the selected gear with the keyway. There is a long slot cut into the tube to slide the key between gears, by which the gear ratio is adjusted and therefore also the fertilizer delivery rate.

Figure 1 - Pull Type Air Boom Fertilizer Spreader [1]

The full schematic of the current gearbox design is shown in figure 2. The schematic helps visualize the three main issues that are required to be resolved. The first problem is the issue of difficult shifting for the operator and time-consuming assembly of the shifting components. Valmar has also been experiencing manufacturing defects in the form of burring making it difficult for the key to properly slide from one gear to the next. The result is a gearbox that cannot shift gears or cannot engage completely in the selected gear. The problem components are shown circled on Figure 2. A more detailed image is available in Appendix B.

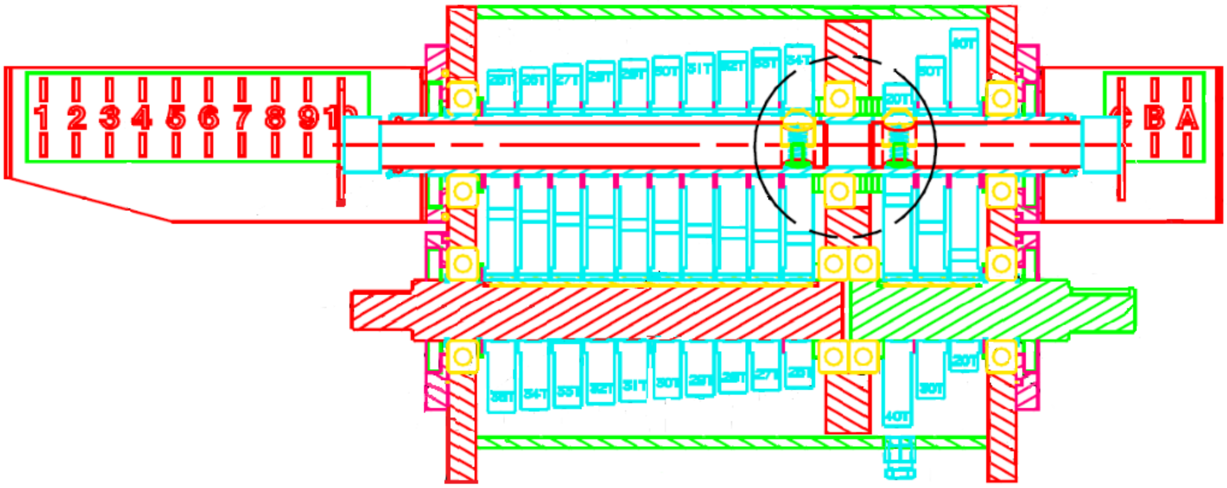


Figure 2 - Gearbox Schematic [2]

The second problem area is the tendency for the gearbox to leak oil. A change was made from a cork gasket after realizing it frequently leaked. Currently, Valmar is using a rubber gasket with silicone to seal the box, but leaking still occasionally occurs as the silicone does not bond to the rubber perfectly. Though the gearbox is machined for flatness as an extra step to ensure a good mating surface, leaking is still occurring.

The final issue is the gearbox weight. The gearbox case is constructed of overly thick steel, causing its mass to be very high. Gearboxes are typically installed on the 7600 model fertilizer spreader using a crane and as a result, assembly of the gearbox to the machine is precarious. Valmar requires a lighter gearbox to not only ensure lower manufacturing costs but to have a safer to install component.

Overall, Valmar requires the redesign of the gearbox to improve three different aspects, including the shifting mechanics, lubrication containment, and the excessive weight.

1.2. Target Specifications

Through communication with Valmar Airflo, a concise breakdown of their project requirement was determined. The three aspects of the gearbox redesign are its shifting mechanism, the tendency for it to leak oil, and its unnecessary weight. Gearbox shifting is the most important aspect to improve, because it affects equipment usability. Other specifications, such as reducing the gearbox weight will not affect how the customer uses the product, but are of benefit by simplifying assembly, installation, and reducing material costs. Table I identifies the client needs.

The gearbox shifting was improved in a number of ways. To quantify changing the gears to be easier, no more than a 5kg force should be required by the user to switch gears. A labor time decrease of 15% is the next improvement to be done. Next, a maximum of 1/1000 incident rate will be required for the shift shafts to dislocate from the gearbox.

The oil leakage needs to be addressed with one priority, which is the elimination of any lubrication leakage.

Finally, the gearbox weight issues must be quantified. A cast case can be considered but must be under 100% of the current weight of the welded case. Scaling is the next option, but a 50% scale factor maximum can only be applied. Overall, the weight of the gearbox should be 60 % to 80% of the current weight.

TABLE I - IDENTIFICATION OF CLIENT NEEDS

<table>
<tr><th>Aspect Needing Improvement</th><th>Sub-Improvement</th></tr>
<tr><td rowspan="3">Gearbox Shifting</td><td>Easier gear changes for the user
not require greater than 5 kg of force to switch
one hand</td></tr>
<tr><td>Reduce time and cost to manufacture components
reduce the labor time by 15%</td></tr>
<tr><td>Rectify tendency for shift shafts to dislocate from the gearbox
reduce to less than 1/1000 incident rate</td></tr>
<tr><td>Oil Leakage</td><td>Design a method of lubrication that is better contained
reduce leakage to 0%</td></tr>
<tr><td rowspan="3">Gearbox Unnecessarily Heavy</td><td>Opt for a cast case versus a welded case
Cast case to be under 100% weight of welded case</td></tr>
<tr><td>Scale parts where possible
scale down by a maximum of 50% of the width</td></tr>
<tr><td>Decrease overall weight decrease
decrease weight 20% to 40%</td></tr>
</table>

Some of the items above yield greater overall benefit than the others, so a prioritized table was prepared. The table is made up of the project deliverables sorted on their importance to the client with essential items being the most critical. Of the essential needs, both the shift shaft dislocation and the oil leakage have a direct effect on product performance. Conditional items provide valuable improvement to the gearbox, but these items must be weighed against their cost to improve. Optional items do not significantly affect the performance of the product, but their application will either make the gearbox easier to manufacture, less expensive, or more user friendly.

TABLE II - PRIORITIZATION OF CLIENT NEEDS

Importance	Specific Need
Essential	• Reduce tendency for shift shafts to dislocate from the gearbox
	• Eliminate oil leakage from the gearbox
Conditional	• Reduce time and cost to manufacture gearbox internals
	• Smoother gear shifting for the user
	• Reduce the weight of the gearbox
Optional	• Clearer identification of gearing selection
	• Cast gearbox case instead of a welded unit

1.3. Project Objective

The goal of this design project is to design and develop an improved metering gearbox for use by Valmar Airflo Incorporated on the 7600 Pull Type air boom fertilizer spreader. Consequently, the project will deliver the three primary requirements and expectations the client has laid out. The three objectives of the improved design are better shifting mechanics, better sealing to prevent leakage and a lighter gearbox weight.

The objective to redesigning the shifting mechanics is to allow easier gear changes for the operator, and faster assembly for the technicians. Additionally, the project addresses the operational failure that occurs when the shift shaft dislocates. To keep the gearbox compatible with the 7600 fertilizer spreader, any redesigning of components must not change the gearbox's speed ratios, or its mounting interface. The inside of the current gearbox design is pictured in figure 3. Figure 3 shows 2 sets of 13 gears separated into subsections of 10 and 3, allowing a total of 30 different gear configurations.

Figure 3 - Current Design of Gearbox Housing and Interior [3]

The next project objective is to redesign the gearbox in such a way as to eliminate oil leakage. The current method of a rubber gasket and silicone has proven ineffective. The rubber and silicone do not bond properly and the oil bath will then leak out. To accomplish this goal a number of solutions are analyzed. The design must also have no major retooling costs however, as there are so few gearboxes produced per year. Valmar Airflo usually manufactures only twenty 7600 gearbox units per year, so it is not beneficial to change any processes greatly such that the company has a long return on investment to deal with.

The final problem to solve is that the gearbox is extremely heavy due to components being oversized. Being used for metering and not power transmission, the gearbox is under a low amount of stress and has a relatively small amount of torque being placed on it. The weight does not help the gearbox in any way and only adds to the difficulty of installation and cost to produce.

To sum up, the design set to be produced must be a beneficial upgrade for Valmar so that it not only improves the product, but reduces the workload and cost of production for each unit.

2.0 Initial Research

An initial client interview was conducted to gain technical information on the gearbox and its application. The initial visit provided the problem background information that was analyzed for the project definition. Getting the best grasp possible of client expectations and project specifications successfully lead the transition into the project brainstorming phase. The goal of the first brainstorming session was to generate a short description of potential solutions to the client requirements. To each of many solutions, we provided their approximate feasibility which was insightful during the research process.

For better "hands-on" to the project, we conducted a second interview at Valmar with the manufacturing coordinator. During this site visit we observed the gearbox being assembled. Seeing first-hand some of the problems encountered during manufacture was extremely useful for subsequent concept analysis. We arrived with prepared questions and they were answered as best as possible by the manufacturing coordinator.

In addition to the brainstorming sessions described above, we implemented the Theory of Inventive Problem Solving, commonly referred to as TRIZ. The basis of TRIZ is a collection of 40 principles that can be applied to engineering problems in an attempt to encourage inventive thought leading to solutions. Customer requirements were divided into the three categories of weight reduction, shifting improvement, and retaining of lubrication. Each group member selected their top two TRIZ principles that apply to the category of improvement. The goal of this exercise was to gain direction for the solution to these problems. Compiling the selections from each group member resulted with between 3 and 5 different TRIZ principles, as there were overlapping selections.

2.1. Standards and Codes

The agricultural industry relies heavily on gearboxes to perform various functions for the vast amount of equipment present in modern farming. The American Gear Manufacturers Association (AGMA) is a body instituted to control all United States standards on gearing [6].

For a fee, gear standards can be purchased from AGMA to help in the design process. This gearbox utilizes spur gears which offer a good combination of efficiency, durability and cost. Other types of gears that we considered during brainstorming were helical, which would be impractical due to their axial load component, and also worm gears which would be unsuitable. Either would necessitate a complete redesign of the gearbox internals and case structure. The spur gear style currently used is very popular in gearbox design and will be our best option. Our concept analysis dictates adjusting the gear thickness to reduce mass and conserve material. Consulting with AGMA standards will help to determine the appropriate gear thickness. Due to the small production quantity of this gearbox, custom designs that drift slightly from standards, may be acceptable.

A corner point design standard is the factor of safety applied to stressed members. Critical engineering designs rely heavily on factors of safety, but failure of this gearbox isn't considered critical because no injury or personal harm would result in the event of failure. A factor of safety wasn't available from the initial gearbox design, do we don't have a value to compare new designs against.

Standards exist for the input and output shafts on the gearbox; however the scope of our project requires no modification to these pieces. These standard items are in place to allow easier part replacement for the end user. If the shafts were to be modified, their mounting interface would need to be adjusted accordingly. Engineering codes exist when standards are implemented by governing groups, or through contract negotiation. As such, they do not apply to our project.

The results of a patent search showed there are no existing patents that prevent us from completing the upgrades to the gearbox. As Valmar intellectual property, they should be able to pass this technology to the 7600 model fertilizer applicator without patent complications.

3.0 Proposed Redesign of the Gearbox

The gearbox was redesigned to meet the client's needs in three different ways. First the overall weight of the gearbox was reduced. The weight of the case itself was reduced, a stress analysis was done to justify a reduction in size of the gears and finally an overall weight analysis was done. The second need that was met was to design the improved gearbox to reduce leak occurrences. This was done in a number of ways: a grease substitute was considered, the bolt count along the case lid was analyzed, crush washers were used and the gasket material was changed. Finally, the shifting mechanism was analyzed. The shift shaft dislocation was evaluated and the shift shaft key assembly process was investigated. The final design changes are summarized after each need.

3.1. Case Weight

The case of the gearbox is comprised of thick steel plates welded together. The mass of the gearbox was measured to be 148.2lb, and Valmar requires this to be reduced. Valmar noted that a reduction in weight of between 20% and 40% would be ideal.

3.1.1. Reductions in Mass

The overall design goal is to reduce the total weight of the wet gearbox for the 7600 Pull Type air boom fertilizer spreader. The weight was lowered in three separate ways. The first method of weight reduction was the aforementioned reduction in gear thickness. The second method of weight reduction was the subsequent shortening of the gearbox length. Finally, a reduction in size in the internal bearing wall is done. The current measured weight of the gearbox is 148.2 lb and the design goal is to reduce this as much as possible.

There are a number of restraints on the gearbox that will not allow for a complete overhaul of the design. Since the wet gearbox is already in a certain configuration of the 7600 fertilizer spreader, the proposed gearbox design has to meet the same interface requirements. This limits the amount of changes can be made, to the x-direction as shown in figure 4. Changes in the y and z direction, shown in the figure, would result in the gearbox being unable to connect properly to the fertilizer machinery. Therefore, any major changes in the other two directions will result in the gearbox not meeting the configuration.

Figure 4 - Diagram of Gearbox Configuration Showing the Direction of Allowable Changes. [1]

Another restraint on the gearbox is that of cost. The wet gearbox has to be either less expensive or the same price as the current model. With a low production value of roughly 20 gearboxes a year, it will simply be unprofitable to make the gearbox more expensive. Therefore, complex machining and high strength materials are simply out of the design options. The materials costs of the redesign are kept equivalent or less than the original design in order to keep the new manufacturing cost low. Finally time restraints only allow the gearbox to be completed before December due to the nature of the task. This means that a total redesign of the gearbox would be unfeasible as complete miniature models would have to be constructed and tested. Therefore, the weight will be reduced as much as possible with the set constraints.

To justify a change in gear size a stress analysis of the gears has to be executed, in the following section. The gears are labeled for the stress analysis as shown in figure 5. These labels were arbitrarily assigned for ease of reference. The gears are labeled from 1 through 10 with 1 being the smallest gear of set 1, and from A to C with A being the smallest gear of set 2.

Figure 5 - Current Gear Configuration Labeled A-C and 1-10 with A and 1 Being the Smallest Gears.

3.1.2. Gear Redesign and Stress Analysis

A primary goal of weight reduction redesign is to use the thinnest gear possible. The main factor thus limiting the gear size is the allowable bending stress the gear is subject to. Therefore this value must be calculated to determine the smallest size available and of what material properties are required. Stress has an inverse relationship with revolutions per minute. This means that since the highest gear ratio is 24/17, the wet gearbox will have an output revolutions-per-minute higher than the input. Since the input moves slower, the stresses will be higher and calculations will therefore be made for the input alone. To do this, one must first find the material of the gear and the subsequent hardness. CD 1045 steel is used in the gears, which has yield strength of 77 ksi [12]. Using figure 6, one can convert the yield strength of steel to a hardness value in HB. Assuming a linear relation between close points, a value of 158 HB was obtained from the figure.

Figure 6 - Hardness Equivalent for Tensile Strength of Steel [12].

The hardness thus has an equivalent allowable stress number to allow proper material selection. The hardness is a function of stress as shown in figure 7. Using the hardness of 158 HB, and assuming the worst case scenario of a Grade 1 Material, an allowable stress, s_t, of 80 ksi is found from this figure.

Figure 7 - Allowable Stress as a Function of Hardness, HB [12]

Therefore, to find a suitable gear size, the stress must be determined. Stress, shown in equation 1, is a function of the transmitted load, W_t, the diametrical pitch, P_d, which is the number of teeth per inch of diameter, the thickness of the gear teeth, F, and the geometry of the tooth itself, J.

$$\sigma_t = \frac{W_t P_d}{FJ} \qquad \text{[eq 1]}$$

The transmitted load, W_t, is a function of power in horsepower, the number of teeth and diameter given. The load equation gives a result in pounds, shown in equation 2.

$$W_t = \frac{(126000)(P)}{(nD)} lb \qquad \text{[eq 2]}$$

Therefore to calculate the stresses acting on a gear, the horsepower must be estimated and the gear speed must be calculated. A high estimate of horsepower of 5 was initially estimated, though a range will eventually be determined. This conversion of speed is found from equation 3, which was derived from the hand-drawn diagram from Valmar shown in Appendix B. Since stress has an inverse relationship with the revolutions per minute, the lowest speed of the range was substituted. Greater stress occurs with lower speeds so the lowest value of the range was used for calculations.

$$870\frac{rev}{min} * 8\frac{miles}{hour} * \frac{1}{60} * \frac{Hours}{Minute} = 116\,rpm \qquad \text{[eq 3]}$$

The value of 116 rpm is then sped up by a worm gearbox by a factor of 1.5. Therefore, a value of 174 rpm is used for the calculations. Finally, the 13 gears must be measured for their number of teeth, N and their outside diameter Do. From there all other values can be calculated. First the diametric pitch, Pd, is found. The diametric pitch is the ratio of the number of teeth by the amount of distance around the outside of the gear, and which the value is determined from equation 4.

$$P_d = \frac{N}{Circumference} = \frac{N}{(\pi * D_o)} \qquad \text{[eq 4]}$$

Next the 'a' and 'b' values are determined which will be used in subsequent equations. They are functions of the diametric pitch and are found as follows:

$$a = \frac{1}{P_d} \qquad b = \frac{2.5}{P_d} \qquad \text{[eq 5]}$$

A number of new parameters can now be found dependent on the 'a' and 'b' values. The first is the diameter, D, which is the diameter that the gears act upon. This means that it is diameter of the circles formed which pairing gears touch. The diameter can be found from equation 6.

$$D = D_o - 2a \qquad \text{[eq 6]}$$

The root diameter, D_r, is next found. The root diameter is twice the distance from the center of the gear to the start of the gear from the middle. This value can be found using equation 7, which is also a function 'b'.

$$D_r = D - 2b \qquad \text{[eq 7]}$$

Next is the whole diameter, h_t, which is the total distance from the root of the tooth to the top. The whole diameter is a function of 'a' and 'b' and found from equation 8.

$$h_t = a + b \qquad \text{[eq 8]}$$

Finally, the speed at the pitch, V_t, is found in the units inches per minute. V_t is calculated using equation 9.

$$V_t = 0.5 * D_o * n \qquad \text{[eq 9]}$$

A sample of the summary of the data obtained is presented in Table III. The complete table can be found in Appendix C.

TABLE III - SUMMARY OF TEETH NUMBER, DIAMETERS AND SUBSEQUENT CALCULATIONS FOR GEARS A TO C

Gear Calculations				
Description	**Variable**	**A**	**B**	**C**
Outside Diameter	D_o	2.75	4	5.1875
# Teeth	**N**	20	30	40
Circumference	**C**	8.6394	12.5664	16.2970
Diametric Pitch	P_d	2.3150	2.3873	2.4544
-	**a**	0.4320	0.4189	0.4074
-	**b**	0.5400	0.5236	0.5093
Diameter	**D**	1.8861	3.1622	4.3726
Root Diameter	**Dr**	1.0221	2.3245	3.5578
Whole Depth	h_t[in]	0.9719	0.9425	0.9167
Speed @ Pitch	V_t[in/min]	239.25	348	451.3125

A pressure angle, Φ, must be known, which is based on the geometry of the tooth. Though there are three angles that are manufactured, 20° was chosen as it is the most common tooth type [12]. Now that all values for the gear are known, the transmittal load is determined from equation 2. The transmittal load is the force transmitted through the gear teeth.

Next, a new face width, F, must be chosen, and the current one must be measured. Measurement with a caliper found F to be 5/8”. Through trial and error as well as backwards calculations of the stress, 5/16” is chosen to minimize size without gear failure. Whole fraction values were assumed as they require less machining, which again increases costs. The geometry factor, J, can now be determined. J is a function of F, the input gear’s tooth count and the output gear’s tooth count, and is solved using figure 8.

Figure 8 - Geometry Factor, J, as a Function of N_{input} and N_{output} [12]

With all four values known, the ideal stress can now be calculated from equation 1. A sample summary of the results for gears A-C is shown in table IV.

TABLE IV- SUMMARY OF IDEAL STRESS AND THE VARIABLES NEEDED FOR CALCULATION

Ideal Stress Variables				
Description	**Variable**	**A**	**B**	**C**
pressure angle	**Φ [°]**	20	20	20
transmittal load	**Wt [lb]**	1 919.71	1 144.98	828.03
Torque	**T [lb in]**	2 715.52	2 715.52	2 715.52
geometry factor	**J [in]**	0.325	0.365	0.37
face width original	**F_1 [in]**	0.625	0.625	0.625
face width proposed	**F_2 [in]**	0.3125	0.3125	0.3125
ideal stress original	**σ_t [psi]**	21 879	11 982	8 789
ideal stress proposed	**σ_t [psi]**	43 757	23 964	17 577

The ideal stress does not account for any kind of correction factor, which is dependent on the size, the overloads that occur and other possible correction factors listed. The correction factors must now be accounted for using equation 10. The actual stress is a function of the ideal stress, the overload factor K_o, the size factor, K_s, the load distribution factor, K_m, the rim thickness factor, K_b and the dynamic factor K_v.

$$\sigma_t = \frac{W_t P_d}{FJ} * K_o * K_s * K_m * K_B * K_V \qquad \text{[eq 10]}$$

The overload factor, K_O is given in the problem description as 1.50. The size factor, K_s is 1.00 for a P_d value of 5 or less [12]. The load distribution factor, K_m, is equal to the sum of the pinion proportion C_{pf}, the mesh alignment C_{ma} and 1. To find the C_{pf} and C_{ma} the ratio of F/D_p is first found and the values are then looked up using figure 9 and figure 10.

Figure 9 - C_{pf} as a function of F and F/D_p [12]

Figure 10 - C_{ma} as a function of F [12]

The rim thickness factor, K_b, is a function of m_B, which in turn is a function of parameters calculated in Appendix C. Using equation 7 and figure 11, the rim thickness factor of 1.00 is determined for all cases.

$$m_B = \frac{t_R}{h_t} \qquad \text{[eq 7]}$$

Figure 11 - KB as a Function of mB [12]

The dynamic factor K_v is found based on the pitch line velocity calculated in Appendix C. As well a quality value, Q_v was shown to be unneeded as the values converge to 1.0, at the speed of 174 rpm. Figure 12 shows how these values were determined.

Figure 12 - Kv as a function of pitch line velocity [12]

With all correction factors now known, the actual stress can be calculated as shown in table V using equation 10. The proposed stress can then be compared to the allowable stress of 80 ksi. If the difference between the values is positive, the design is valid. Provided is a sample summary for gears A-C.

TABLE V - SUMMARY OF CORRECTION FACTORS AND THE RESULTING ACTUAL STRESS

Resulting Stresses				
Description	**Variable**	**A**	**B**	**C**
ideal stress original	**σ_t [psi]**	21 879	11 982	8 789
ideal stress proposed	**σ_t [psi]**	43 757	23 964	17 577
overload factor	k_o	1.4	1.4	1.4
size factor	k_s	1	1	1
ratio original	**F_1/D**	0.3314	0.1976	0.1429
ratio proposed	**F_2/D**	0.1657	0.0988	0.0715
pinion proportion	C_{pf}	0.1	0.1	0.1
mesh alignment	C_{ma}	0.13	0.13	0.13
load distribution	k_m	1.23	1.23	1.23
backup ratio	m_b	0.5258	1.2332	1.9405
rim thickness	k_b	1	1	1
quality number	Q_v	10	10	10
dynamic factor	k_v	1	1	1
actual stress original	s_t[psi]	37 675	20 633	15134
actual stress proposed	s_t[psi]	75 350	41 267	30 268
allowable stress	st [psi]	80000	80000	80000
difference	Δst [psi]	4 650	38 733	49 732

As shown above, all differences are positive for each gear and therefore all gears are valid. Finally the horsepower range was determined by finding the point where the proposed design stress was equal to the allowable stress. This limit was found to be 5.038 hp and thus the valid range for this stress analysis is between 0 hp and 5.038 hp. Figure 13 shows the final gear design proposed change from 5/8" gears shown on the left to the 5/16" gear shown on the right. With the stress analysis complete, and the gears verified to work, a weight analysis is conducted.

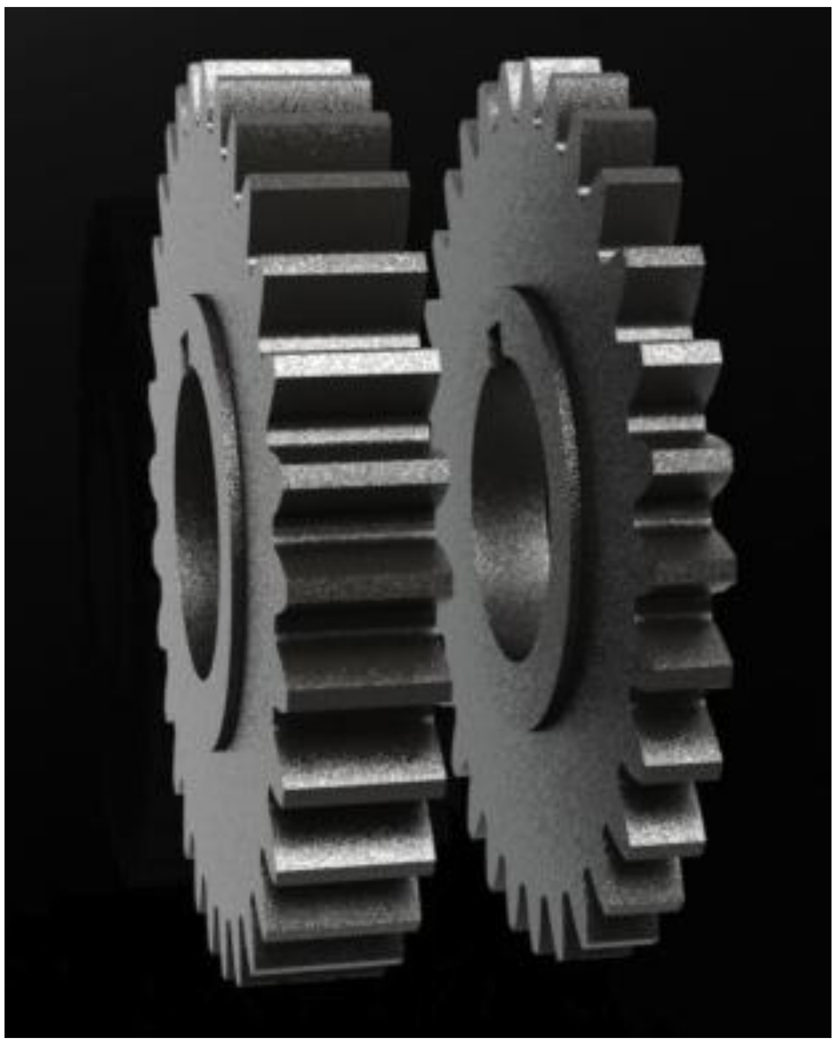


Figure 13 - Gear Size Difference From Current Gear Design (left) to Improved Design (right).

3.1.3. Overall Weight Analysis

The total reduction of weight of the wet gearbox for the 7600 Pull Type air boom fertilizer spreader is calculated. The weight was reduced in a total of three ways. The first method of weight reduction was the aforementioned reduction in gear thickness, a term defined as 'F'. The second method of weight reduction was the subsequent shortening of the gearbox length. Finally, a reduction in size in the internal bearing wall was prescribed.

The first method of weight reduction involved reducing the size of the gears. As the main component of a gearbox, reducing the gears' size was an obvious way to make the gearbox weigh less. Since the gearbox's configuration had to remain constant, as the gearbox is attaching to other components, changes in the diameter of the gears would be problematic. Therefore gears were reduced in their thickness, F. Through the stress analysis already shown, the gears were shown to still have a reasonable amount of shock sensitivity with a reduction in F of precisely 50%. Originally 10/16" thick, the proposed gears are to be 5/10" reducing their weight, through symmetry by exactly 50%. A summary of these values is shown in table VI.

TABLE VI - SUMMARY OF CURRENT AND PROPOSED GEAR WEIGHTS AND THEIR TOTAL CONTRIBUTION TO WEIGHT OF THE GEARBOX

Gear Name	Current Design Weight [lb]	Proposed Weight [lb]
A	0.59	0.30
B	1.66	0.83
C	3.16	1.58
1	1.07	0.54
2	1.19	0.60
3	1.30	0.65
4	1.41	0.71
5	1.53	0.77
6	1.66	0.83
7	1.80	0.90
8	1.93	0.97
9	2.08	1.04
10	2.22	1.11
Total	21.60	10.80
Total Both Sets	43.20	21.60

As this table shows, one set of gears contributes 10.80 lbs to the total gearbox weight. Since there are two sets of gears there will therefore be a total 21.60 lb reduction in weight from the gears alone.

The second method of weight reduction was to subsequently shorten the gearbox length by the sum of the gearbox's changes in width. Since the widths of the gears are parallel to the length of the gearbox, the total difference in length will be the change in width multiplied by the number of gears. Therefore the total difference in contribution to the gearbox length from the proposed gearbox, ΔL, is 4 1/16". Since the gears are made of the metal AISI 1010, the density of the case material is known to be 491.4 lb/ft^3 [H4]. With all dimensions of the gearbox known, the volume can be calculated and multiplied by the density. The results of these calculations are shown in table 7.

TABLE VII - SUMMARY OF TOTAL MASS REMOVED FROM THE PROPOSED GEARBOX DESIGN

Mass Savings Total		
Description	**Variable**	**Value**
Proposed gear size	**F [in]**	0.3125
Number of gears	**N**	13
Contribution to gearbox length from gears	**ΔL [in]**	4.0625
Density of walls	**ρ [lb/ft^3]**	491.4
Box wall thickness	**t [in]**	0.25
Box wall height	**h [in]**	6
Box length	**L [in]**	11.625
Box width	**w [in]**	10
Volume change	**V [in^3]**	22.344
Volume change	**V [ft^3]**	0.013
Mass change	**m [lb]**	6.354

As the table shows, the total volume difference is 0.013 ft^3 from using the proposed case design. The total mass change is 6.354 lb reduced. Figure 14 shows the final gearbox length change from the current case length of 11 5/8", shown on the right to proposed redesign of 6 1/16" shown on the left.

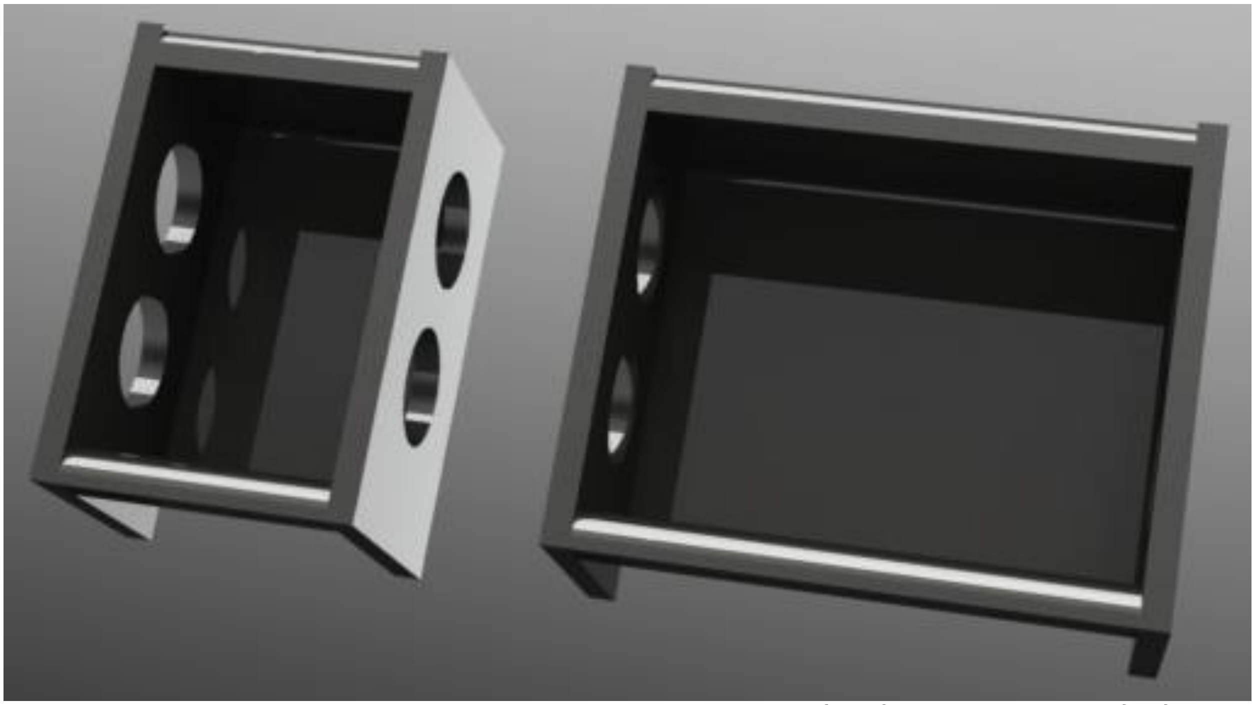


Figure 14 - Gear Case Length Difference From Current Gear Design (right) to Improved Design (left)

The final method of weight reduction is a decrease in size in the internal bearing wall. As shown in figure 15, the wall separating gears A-C with gears 1-10 is currently one piece of AISI 1010 steel 1" thick.

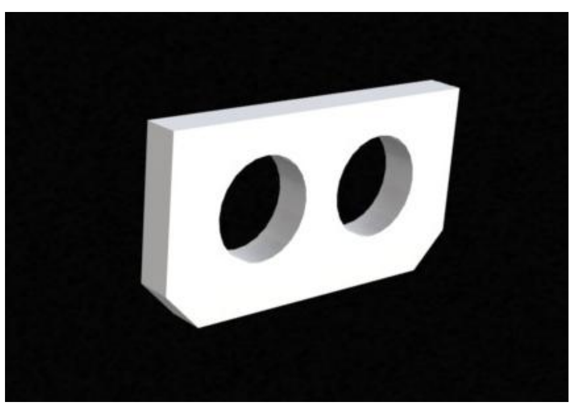


Figure 15 - Original Center Wall.

The proposed solution is to use two walls welded together as shown in 16. One wall will remain the same thickness of 1", as it still has to carry the two 2/3" bearings. The other half will be reduced to a ½" wall, as it only houses one 2/3" bearing.

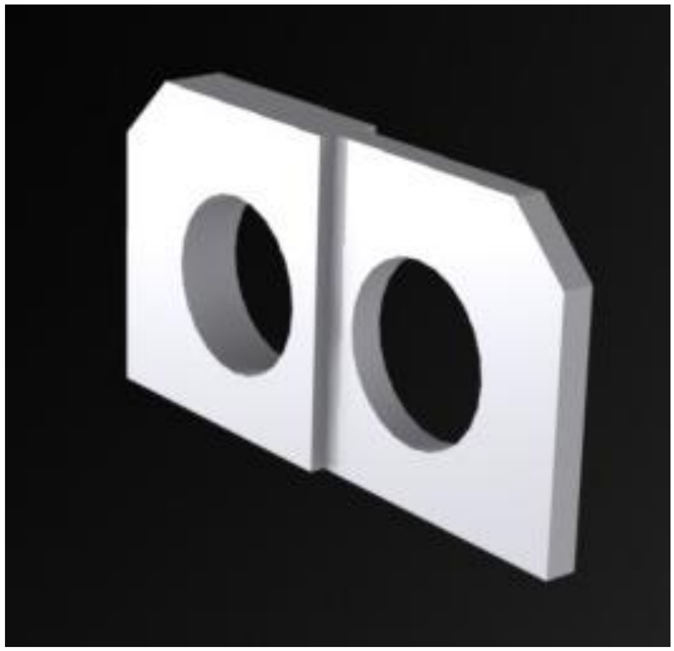


Figure 16 - Proposed Design for Bearing Wall.

As a result of this reduction, there is a decrease in weight of the bearing wall. Overall, the original mass of the bearing wall was found to be 15.53 pounds and the improved design is 11.65 pounds. This is a difference of 3.88 pounds or exactly a 25% change. A summary of these calculations is shown in table 8.

TABLE VIII - SUMMARY OF TOTAL MASS REMOVED FROM BEARING WALL

Mass Removal of Bearing Wall		
Description	**Variable**	**Value**
Density of walls	**ρ [lb/ft^3]**	491.4
length of bearing wall	**L [in]**	9.5
thickness of wall	**t [in]**	1
height of wall	**h [in]**	5.75
removed volume	**V [in^3]**	13.6563
removed volume	**V [ft^3]**	0.0079
mass removed	**m [lb]**	3.883

4.1.4. Case Weight Summary

The total weight reduction is the sum of these three changes in weight. These total changes are summarized in table 9.

TABLE IX - SUMMARY OF TOTAL WEIGHT REDUCTIONS

Weight Reduction Totals		
mass change gears	**m [lb]**	21.600
mass change length	**m [lb]**	6.354
masss change bearing wall	**m [lb]**	3.883
total change	**m [lb]**	31.84

Therefore, a total weight change of 31.84 lb will occur from the proposed design change. The current design weight of the wet gearbox is measured to be 148.2 lb. This is therefore an overall change of 22%, which, though less than the design goal, is the best possible design given the restraints set in place.

4.2. Oil Leak

The oil bath in the Gearbox Case is used to keep the gears constantly lubricated. There are problems with the gasket not bonding with the silicone sealant and oil leaking out of the case. The lid of the case has a plug welded to it for visual cue of the oil level, but leads to warping on the lid and unevenness when mounting to the case of the gearbox. This is another source of problems when bolting down the lid as it is uneven and becomes more difficult to mount flush to the lid.

4.2.1. Grease Substitute

The oil bath the gears sit in could be changed to grease. The gears spin is reported to peak at around 174 revolutions per minute (see Section 3.1.2) so their need for constraint oil bathing might not be required. Grease is also less prone to leakage and handles stop and start movement of the machine better. Grease churning in gears does result in much more heat production than oil. This can lead to heat induced oxidation and the need for lubrication changes more frequently. The lid would no longer require an oil plug to be welded thus removing the warping of the lid that will cause mating problems.

Proper maintenance and lubrication is required for the gearbox. Grease can be packed and left alone for longer periods compared to oil which has to be drained and replaced. Oil is easily monitored via a see-through cap on the hub. Of course, there may be condensation forming during cold to hot transition in temperatures, ultimately leading to gear damage and failure.

Oil baths can be expected to go much longer distances than grease, but only if they are used on a regular basis. Let them sit for a few weeks and condensation can form. But since the system experiences so little torque (as calculated in Gear Redesign), there is little in room of the condensation and storage for months between seasons causing damage when the gearbox is started up again.

The primary issue comes down to customer care. The customer will typically run the gearbox until the gearbox is broken. If the gearbox runs low on oil they put more in. There is no maintenance done outside of putting more oil in the gearbox. This makes any suggestion of packing grease in place of oil a hesitant one, as with the benefit of grease would be the requirement of a proper re-greasing every few years.

4.2.2. Bolts Along Case Lid

The case lid mounts onto the case head with bolts around the edge of the lid. The spacing of the bolts varies on each side and the tension at each point is not enough when combined with the gasket to prevent leaks. The case lid is warped due to the welding of the plug and even when flattened with a press it still has some unevenness. More bolts along the lid can help create a more even application of force applied to the gasket for a greater seal to counter this warping. There is no gain in putting too many bolts as there would be no benefit and it could impact the structural integrity of the case walls. The lid only has 8 bolts holding it down with a distance of 5 to 6 inches between each bolt depending on the side (see Figure 17).

When coupling both the case size reduction due to the slimmer gears and adding more bolts along the lip of the case, we ended up with a 16 point mount (see Figure 17).

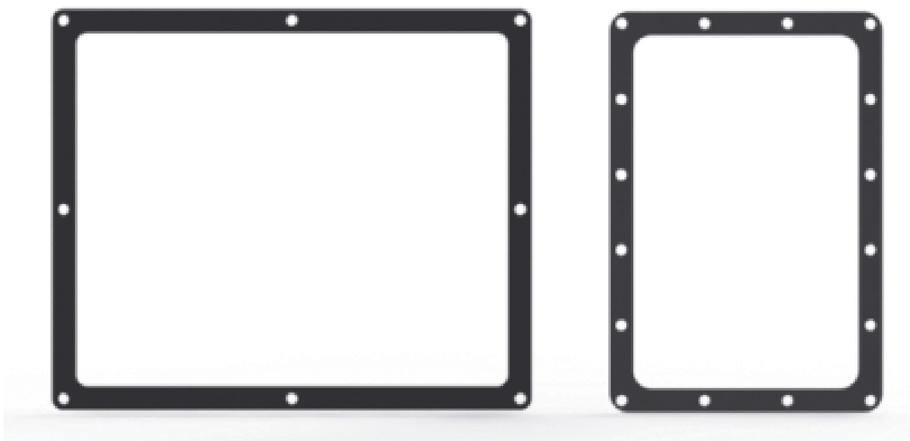


Figure 17- Gasket of Old Case (left) versus New Case and More Points (right).

This will help prevent any possible leaks down the road as the gasket now has enough pressure on it to create a tight seal for the case. It counteracts any warping of the lid.

4.2.3. Case Lid Groove

A lip could be manufactured in the case lid to have a tighter seal on the gasket. There would be less surface area in contact with the case edge but more force making contact between the surfaces and the gasket. The lid would need to be as flat as possible, which is a problem right now due to the warping when welding on the oil viewer. A quick pass in machining could be done along the walls of the case lid before it is welded to have an even cut. When the lid is attached with multiple bolts the forces will cause the lid to sit evenly and the previous groove cut will remain identical along the edge.

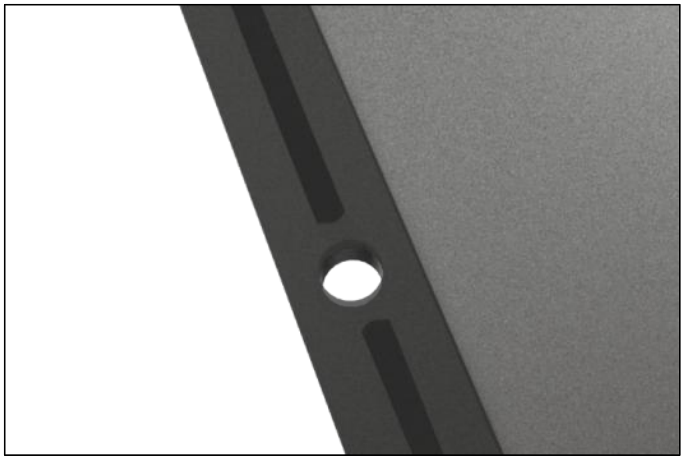


Figure 18 - Groove Along Case Lid

4.2.4. Crush Washers for Case Lid Bolts

Install a crush washer on the outside for each bolt head to tighten down on the lid to prevent any fluid leaking out along the threads. It is recommended that nylon is used so as to not cause any galvanic corrosion that could happen with copper or other metal washers.

4.2.5. Gasket Material

The gasket is not bonding well with the silicon for the gearbox case to seal properly. Changing the gasket material could help with less leaks and a better bonding result with the silicone. The gasket would still need to have a long endurance life and be able to handle the oil inside the gearbox without deteriorating.

When reviewing the material and manufacturing procedure of the gearbox, it becomes apparent that the flaws in the case lid and how it's mounted is the biggest culprit to the leak. The case lid isn't smooth due to its warping when welded. The silicone is not bonding properly as the gaps where the oil leaks out are due to inadequate force from the lid. Previous suggestions such as more bolts and a groove will solve this problem. After applying the previous measures then use the bonding agent as a sealant.

4.2.6. Oil Leak Summary

The gearbox's leaking problems stem from the key components that were addressed above. Proper mounting points, grooves for a tighter fit, and crush washers for sealing in oil that escapes along the threads are the main design proponents that will reduce the oil leaks for the gearbox. It would not be advisable to go with a grease packing option as the maintenance handled by the customer would have to change. There is no evidence that grease would result in better performance if the main issue with the leak is that more oil has to be added. The oil bath has provided the protection and support required even with the fluid loss.

The case lid warping from welding can be reduced with additional passes through a press if required. The extra mounting points and the groove will give a stronger barrier for the oil to rest against. A crush washer would provide a preventative measure down the road of any other possible sites for leaks. The sealant used for the rubber gasket is the proper type for that application. Its ineffectiveness stems from the previously mentioned issues of the case lid. The sealant could not make up for the inadequate mounting points. When used in conjunction with the upgrades addressing the oil leak, the gasket will have a much more cohesive bonding to the case.

4.3. Shifting Mechanism

The gearbox shifting upgrades represent an important part of the project because they affect either the usability of the machine or the ease of gearbox assembly. One complaint voiced by Valmar about the shifting is the tendency for the shift shaft to over-extend from within the shift tube while being used in the field. If this happens, the gearbox is essentially useless. A second shifting related complaint is the difficulty experienced while assembling the shift key into the shift tube. Valmar assembly technicians have requested a better way to assemble these components, without the high probability of needing to complete the process more than once to ensure a proper assembly.

4.3.1. Shift Shaft Dislocation

Curing the shift shaft dislocation was accomplished by redesigning the shift key. By raising the wall of the key to match the height of the shift tube, the key will disallow the shaft to be accidently removed from the tube. The current key design does not prevent the shaft from dislocation. To design the key, measurements of the existing assembly were gathered using a precision caliper gauge. The maximum vertical height of the key is equal to that of the outer diameter of the shift tube. Key tip geometry has been updated to match the new thickness of the gears. The new design was modeled using SolidWorks CAD software thereby allowing Valmar to easily implement this solution. Altered key dimensions are illustrated in Figure 19.

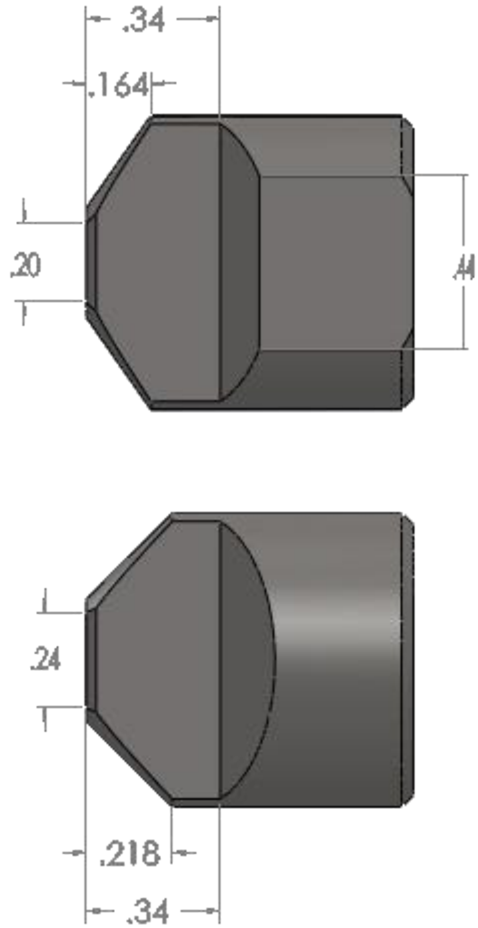


Figure 19 - New Shift Key Design (top) Compared to Old Design (Bottom)

The new shift key design requires no additional tooling from Valmar to implement. Manufacturing of the key is not performed by Valmar and the dimensional changes will not affect the assembly process. Costs to manufacture will be similar to the old configuration because raw materials remain the same, and difference in production time will be negligible.

Maintenance and system reliability will be unaffected by these design changes. Any gearbox repairs will differ slightly. Clearly, the shaft cannot be removed from the tube without more invasive disassembly. Rarely is this type of repair required, but to remove the shaft, the case cover must now be removed to allow access to the shift key. Then the key can be depressed using appropriate tooling to allow removal of the shaft.

The new design is expected to leave field operation of the gearbox exactly the same as the old configuration with the exception of elimination of failure due to shaft dislocation. No additional customer training is required to use the new product.

4.3.2. Shift Shaft Key Assembly Process

During gearbox assembly at Valmar, the shift key can easily rotate within the shift shaft. The assembly process must be repeated until the key tip remains as positioned by the assembly technician. Each attempt to install these parts increases risk of parts loss as these are small, easily misplaced items. In addition, the key and its retainer are under spring tension during the assembly. The technician is required to compress the spring using their fingers while carefully forcing the key into the shift tube. This is quite a precarious situation that can rarely be accomplished in one attempt. Valmar is currently using a much stiffer spring versus their original design, to help prevent the shaft dislocation on over-extension. Since our designed key eliminates the shaft dislocation, the original spring stiffness can again be implemented. The benefits to using a lesser spring constant is easier assembly, and also less force required by the field operator to change gearing selection.

The current shaft has a drilled interface with the key. The solution is to mill a non-circular hole in place of the drilled one. This will prevent accidental rotation of the key during assembly. A comparison of the old and new key is shown in figure 20.

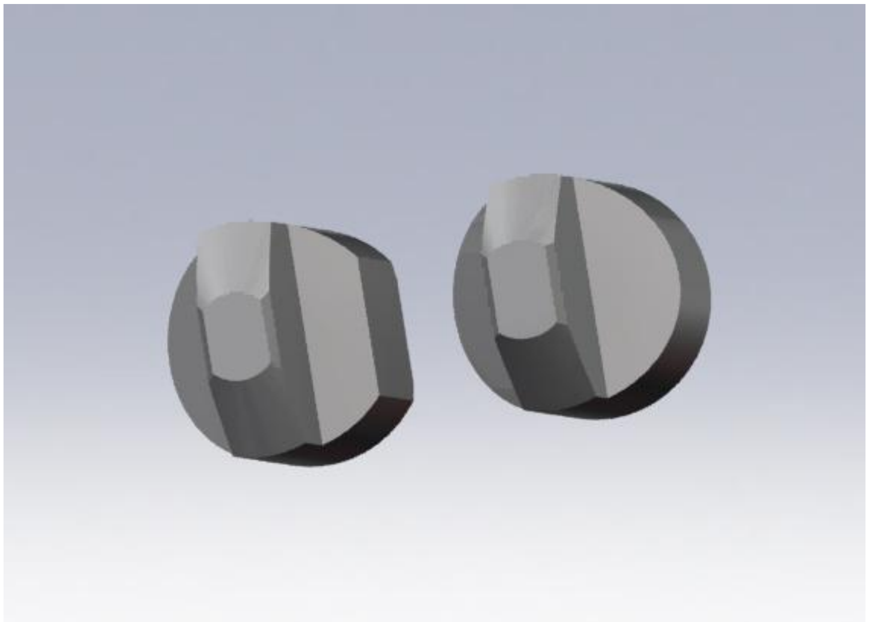


Figure 20 - Comparison of Non-Circular Key (left) to Old Circular Design (right).

These parts are not manufactured at Valmar. Changing from drilling to milling is a more costly, time-consuming process so a small cost increase is expected with these changes, which will be offset by the reduction in assembly time. The new design will not affect maintenance, reliability, or operation of the gearbox. It will only rectify the time consuming assembly process.

4.3.3. Summary of Shifting Analysis

The gearbox shifting mechanism requires significant effort from the user to perform gear changes, in addition to excessive assembly time from the technician. Easier gear shifting is available from the reduced spring stiffness, which is made available from the redesigned shift key. The shift key features revised wall geometry to prevent accidental shaft dislocation by the operator. As well, the shift key features a non-circular base allowing easier, faster assembly for the technician.

4.4. Summary of the Gearbox Redesign

A summary of the all the design changes made is shown in table X. A render of the completed redesign is shown in figure 21, as well as a quick schematic in figure 22.

TABLE X - SUMMARY OF DESIGN CHANGES FOR GEARBOX

Improvement	Current Design	Improved Design
gear size - thickness	5/8”	5/16”
gearbox case length	11 5/8”	6 1/16”
bearing wall mass	15.53 lb	11.65 lb
bolt count	8 bolts	14 bolts
case-lid interface	smooth	grooved
washers for bolts	none	nylon crush washers
key base	round	flattened on side
key tip geometry	allows shaft dislocation	prevents shaft dislocation
key spring constant	high	medium

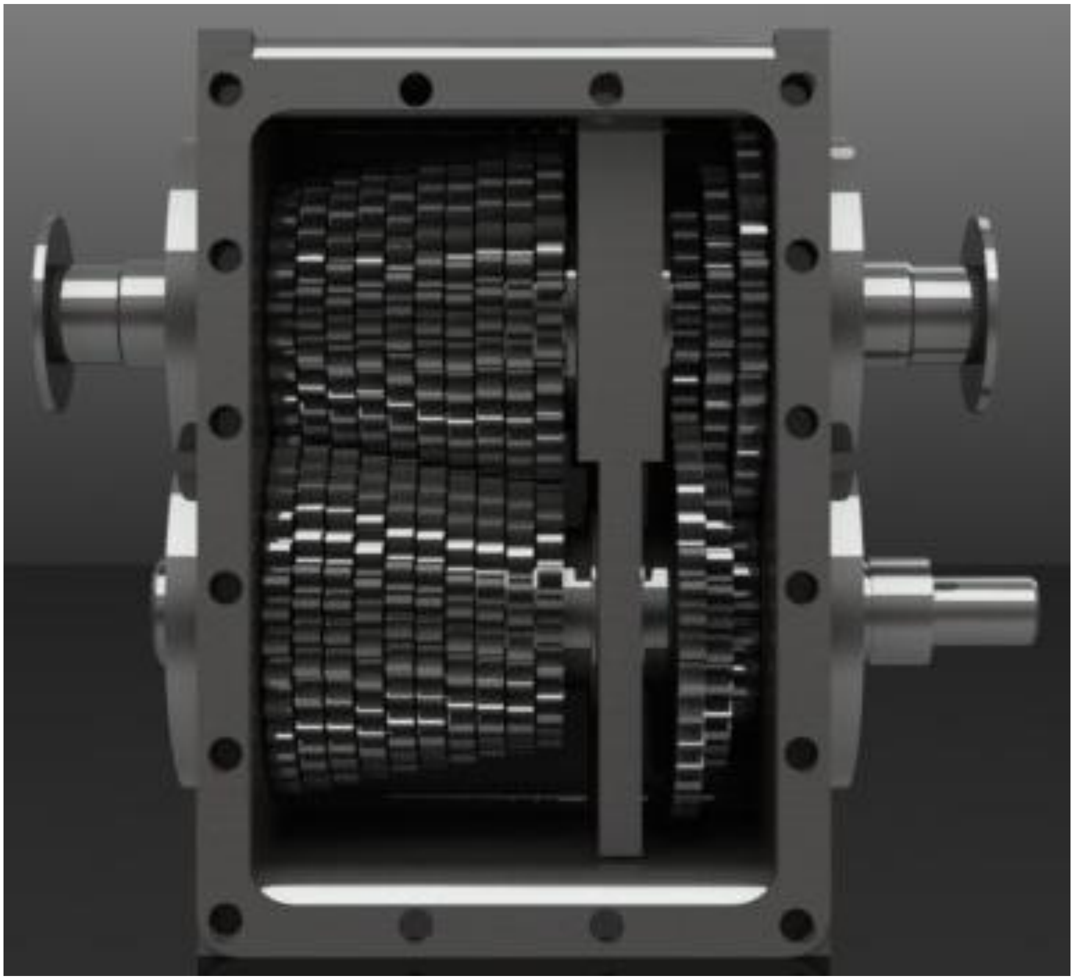
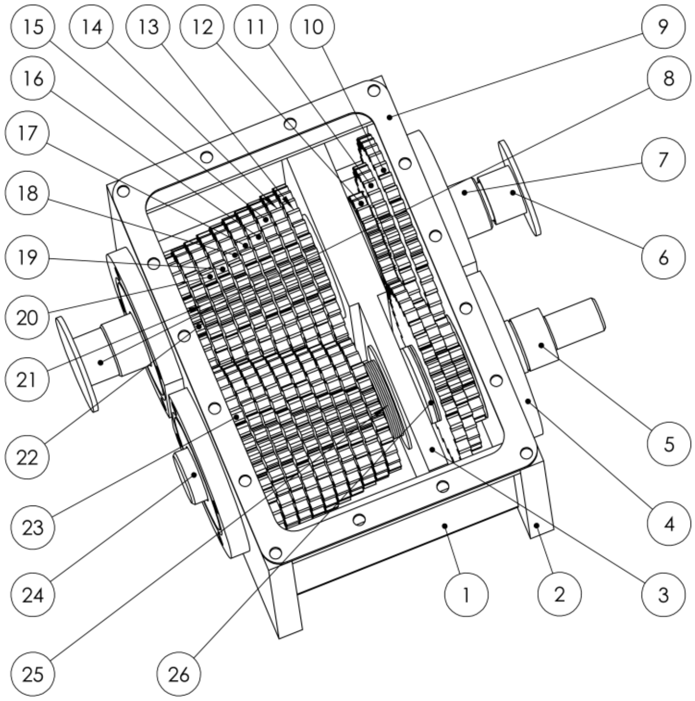


Figure 21 - Completed Redesign of the Gearbox



NO.	Quantity	Name
1	1	Case
2	2	Side wall
3	1	Centre wall
4	4	Seal holder
5	1	Input shaft
6	1	Short shaft
7	1	Tube shifer
8	1	Long shaft
9	1	Gasket
10	2	40T gear
11	4	30T gear
12	2	20T gear
13	2	34T gear
14	2	33T gear
15	2	32T gear
16	2	31T gear
17	2	30T gear
18	2	29T gear
19	2	28T gear
20	2	27 gear
21	2	26T gear
22	1	25T gear
23	1	35T gear
24	1	Output shaft
25	7	0.05 washers
26	2	0.134 washers

Figure 22 - Quick Schematic of the Redesigned Gearbox

5.0 Cost Analysis

A detailed Bill of Materials (BOM) is available in Appendix A. The BOM provides part number, quantity, material costs, and setup and labor costs. For convenience, the current gearbox and our proposed gearbox are included in the BOM.

Savings are realized from the reduction of material of certain items. The material cost of the new gearbox case is found by its known mass reduction:

$$New\ Gearbox\ Materials\ Cost\ (\$) = Old\ Materials\ Cost\ (\$) x \frac{New\ Gearbox\ Mass\ (kg)}{Old\ Gearbox\ Mass\ (kg)}$$

$$New\ Gearbox\ Materials\ Cost = \$71.89$$

The savings arising from material reduction in the gears is calculated similarly, although a "New Part Penalty Multiplier" is assumed as these items are sourced from Ringball Corporation:

$$New\ 30T\ Gear\ Cost\ (\$) = \ Old\ 30T\ Gear\ Cost\ (\$)\ x\ \frac{New\ Gear\ Mass\ (kg)}{Old\ Gear\ Mass\ (kg)}\ x\ 1.50$$

$$New\ 30T\ Gear\ Cost = \$6.74$$

The complete gearbox parts and materials cost is now $681.21, compared to the original $725.69. This represents a 6.13% overall material savings. The greatest source of cost savings results from the redesigned gear set, providing a cost reduction of $58.91.

Some redesigned items such as the gear engagement key, feature the same cost as the currently implemented design. This is because the mass of raw materials needed has not been increased, nor has its cost to manufacture and process.

The redesigned shift shaft has increased in cost as a result of more complicated processing. This cost is manifested in the raw materials – component column.

The cost analysis neglects the time-value of money, and any reduction in shipping and handling as a result of less mass.

6.0 Conclusion

The three objectives of the gearbox redesign are to reduce its mass, eliminate lubrication leakage, and improve aspects of the shifting mechanics. Brainstorming yielded many solutions that were generated, analyzed and presented in a concept matrix. Additional knowledge was gained from competitors' products, patent research, and internal and external searches. The highest scoring concepts during the screening process are the gearbox improvement solutions presented to Valmar.

A total gearbox mass reduction of 22% was accomplished from several design changes. The majority of weight savings came from reducing the thickness of each of the 26 gears. Stress analysis was performed on the gears to find an ideal thickness that would not compromise gear loading. The original gear thickness of 5/8" resulted in very low operating stress. Our calculations prove that 5/16" thick gearing will safely perform within typical material stress levels. The original gear set weighed in at 43.20lbs, while the new set will be 21.60lbs.

Reducing the gear thickness enables the gear set to be more compact, in turn allowing a dimensionally changed gear case. The case is redesigned with less overall length, thereby reducing its mass by 6.35 lbs. The shafts and mounting interface of the gearbox will be unchanged, so installing the unit to the fertilizer spreader will be unaffected.

A third feature for weight reduction is the redesigned internal bearing wall. The original wall provides inefficient use of material, as there is a 1.0in steel plate supporting a 2/3in bearing. The proposed bearing wall is a two piece structure which steps from 1.0in to 0.5in, sufficient for the single bearing on that side. This change reduces mass by 3.88lbs. No increase in structural stress will result from changing the wall, as the single bearing remains sufficiently supported with the new thickness.

All of the methods to reduce mass are economically feasible for Valmar to implement, due to reduced material costs. Combining all of our design changes will yield a 9.18% parts and materials cost savings.

Several options are presented to Valmar that will eliminate loss of lubrication. Our design for revised bolting includes adding eight additional bolts to the case. We recommend this as the initial action to curing the oil loss. The additional bolts yield even pressure on the seal, and add only marginal cost to the product.

Further measures for better oil containment would be to implement a grooved sealing surface. The groove creates high pressure on the sealing forcing the oil to remain internal. This will require modification to the existing case and lid assembly.

There was a possibility of oil leakage past the case lid retaining bolts. We recommend installing a crush washer with every retaining bolt to prevent the possibility of leaks. This is a very inexpensive way to ensure no leaking results at the bolt and lid interface.

A final proposition to Valmar for the oil bath was replacing their oil lubrication with farm machinery grease. If this option is pursued, we recommend that the client evaluate the temperature rise that may result from using grease.

Valmar stated the requirement for design changes that will eliminate shift shaft dislocation from within the shift tube. The redesigned engagement key features revised side walls that prevent shaft dislocation. As the shaft nears the end of its travel, the wall of the key contacts the tube, stopping it from withdrawing. Included models of the new key illustrate the required change. These changes will not affect gearbox operation in any other way.

To rectify the difficulty experienced by the assembly technician during the key install, the engagement key features anti-rotation geometry. Also, the key spring constant is reduced to simplify key install and require less force when the operator changes the gearing selection.

We recommend that Valmar considers pursuing a cast gearbox case in the future. The cast case will introduce significant start-up cost although that cost would be dissipated as more units are produced. A cast case would offer advantages such as reduced assembly time and further weight savings.

Valmar's needs for improved gearbox shifting, oil leakage, and mass reduction were met by our gearbox redesign project. Gearbox shifting improvements include easier gear changes, reduced assembly time, and eliminated shaft dislocation. Our engagement key allow gear changes to be performed by one hand, while reducing time to assemble the key, and eliminating the tendency of shaft dislocation. Oil leakage is eliminated from the redesigned case and lid bolting interface. The chance of oil leakage past the bolt heads is fixed through the use of crush washers. Further recommendations include the possibility of using grease in place of oil for lubrication. Gearbox mass was significantly reduced by optimizing gear thickness. Furthermore, this allowed reducing the case dimensions to save material, and therefore mass. As a result of the significant material reduction, the completed gearbox can be manufactured at a reduced cost compared to the original.

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Appendix

Appendix A: Bill of Materials

TABLE 11 - BILL OF MATERIALS

BILL OF MATERIALS						
	Existing Gearbox				Redesigned Gearbox	
	Item	Quantity	Material Cost ($)	Labour (H) & Setup	Material Cost ($)	Labour (H) & Setup
1	Shifter, Front	1	22.43	0.67	33.65	0.67
2	Shifter, Rear	1	16.51	0.67	24.77	0.67
3	Case, Gearbox	1	80.35	14.63	71.89	14.63
4	Cover, Gearbox	1	11.71	0.13	11.71	0.13
5	Shift Indicator, Front	1	0.94	0.92	0.94	0.92
6	Shift Indicator, Rear	1	0.47	0.92	0.47	0.92
7	Holder, Seal	4	63.04	0.00	63.04	0.00
8	Tube, Shifter	1	70.59	0.00	70.59	0.00
9	Shaft, Output	1	22.45	0.00	22.45	0.00
10	Key, Shifter	2	32.76	0.00	32.76	0.00
11	Shaft, Input	1	25.34	0.00	25.34	0.00
12	Pin, Key	2	8.30	0.00	8.30	0.00
13	Spring, Key	2	5.31	0.00	5.31	0.00
14	Gear, 20T	2	10.08	0.00	7.56	0.00
15	Gear, 25T	1	7.36	0.00	5.52	0.00
16	Gear, 26T	2	13.94	0.00	10.46	0.00
17	Gear, 27T	2	14.89	0.00	11.17	0.00
18	Gear, 28T	2	15.94	0.00	11.96	0.00
19	Gear, 29T	2	16.88	0.00	12.66	0.00
20	Gear, 30T	4	35.95	0.00	26.96	0.00
21	Gear, 31T	2	18.92	0.00	14.19	0.00
22	Gear, 32T	2	20.12	0.00	15.09	0.00
23	Gear, 33T	2	21.08	0.00	15.81	0.00
24	Gear, 34T	2	21.90	0.00	16.43	0.00
25	Gear, 35T	1	11.36	0.00	8.52	0.00
26	Gear, 40T	2	27.28	0.00	20.46	0.00
27	Vent Plug	1	4.31	0.00	4.31	0.00
28	Bushing, Hex 1/2 X 1/8 NPT	1	2.49	0.00	2.49	0.00

BILL OF MATERIALS						
	Existing Gearbox				Redesigned Gearbox	
	Item	Quantity	Material Cost ($)	Labour (H) & Setup	Material Cost ($)	Labour (H) & Setup
29	Sight Glass 1/2" NPT	1	4.87	0.00	4.87	0.00
30	Plug, Pipe 1/4 Hex Head Plated	1	0.17	0.00	0.17	0.00
31	Bearing, Ball 6207	7	23.89	0.00	23.89	0.00
32	Seal, 1V3/8 X 2.839 X 0.312	4	25.20	0.00	25.20	0.00
33	Gasket, Gearbox	1	12.60	0.00	12.60	0.00
34	O-Ring, 3.00ID X 3.25OD	4	0.69	0.00	0.69	0.00
35	O-Ring, 1.00ID X 1.25OD	2	0.09	0.00	0.09	0.00
36	Silicone, RTV Black 10.3Oz	0.194	0.86	0.00	0.86	0.00
37	Shim, 1 3/8ID X 0.062	13	14.34	0.00	14.34	0.00
38	Shim, 1 3/8ID X 0.062 W/Keyway	13	14.34	2.86	14.34	2.86
39	Key, 1/4 x 1/4 x 2.0	1	0.07	0.03	0.07	0.03
40	Snap Ring, #5100-137	6	1.27	0.00	1.27	0.00
41	Key, 1/4 X 1/4 X 7.25	1	0.26	0.12	0.26	0.12
42	Shim, 1 3/8ID X 0.134	9	5.67	0.00	5.67	0.00
43	Shim, 1 3/8ID X 0.020	4	3.57	0.00	3.57	0.00
44	Shim, 1 3/8ID x 0.050	8	1.26	0.00	1.26	0.00
45	Decal, ABC 6600	1	0.76	0.00	0.76	0.00

BILL OF MATERIALS						
	Existing Gearbox				**Redesigned Gearbox**	
	Item	**Quantity**	**Material Cost ($)**	**Labour (H) & Setup**	**Material Cost ($)**	**Labour (H) & Setup**
46	Decal, 1-10 6600	1	1.06	0.00	1.06	0.00
47	Oil, Gear GL-5 80W90	2.5	9.61	0.00	9.61	0.00
48	Flatwasher, 1/4 STD 18-8 SS	8	0.13	0.00	0.13	0.00
49	HHCS 5/16 X 7/8 NC 18-8 SS	4	0.45	0.00	0.45	0.00
50	HHCS 5/16 X 3/4 NC 18-8 SS	12	1.16	0.00	1.93	0.00
51	HHCS 1/4 X 3/4 NC 18-8 SS	14	0.67	0.00	0.67	0.00
52	1/4 Nylon Crush Washer	14	N/A	N/A	2.66	0.00
		Total:	**$725.69**	**20.95**	**$681.21**	**20.95**

Appendix B: Gearbox Diagrams

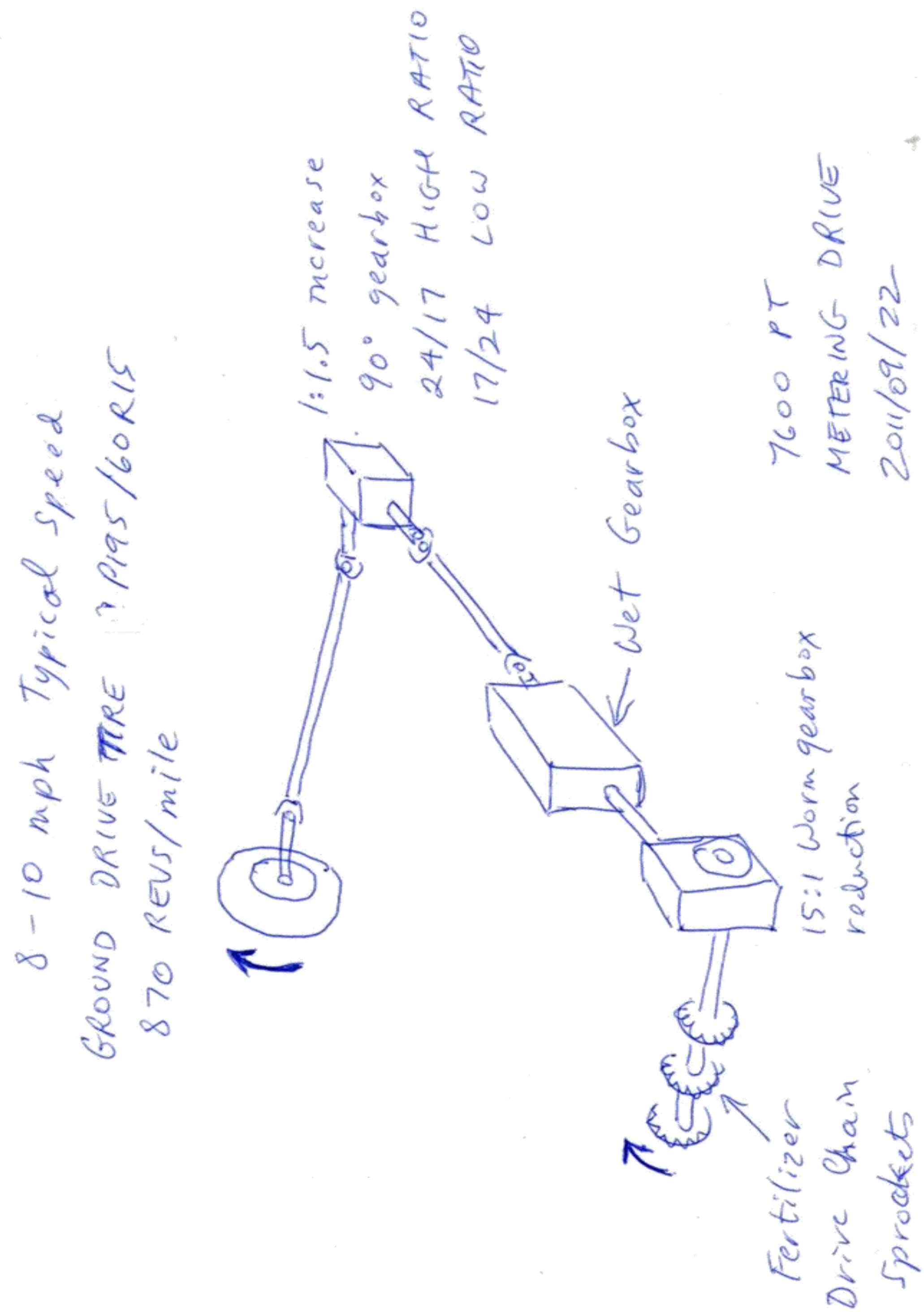


Figure 23 - Fertilizer Spreader Gearbox Layout

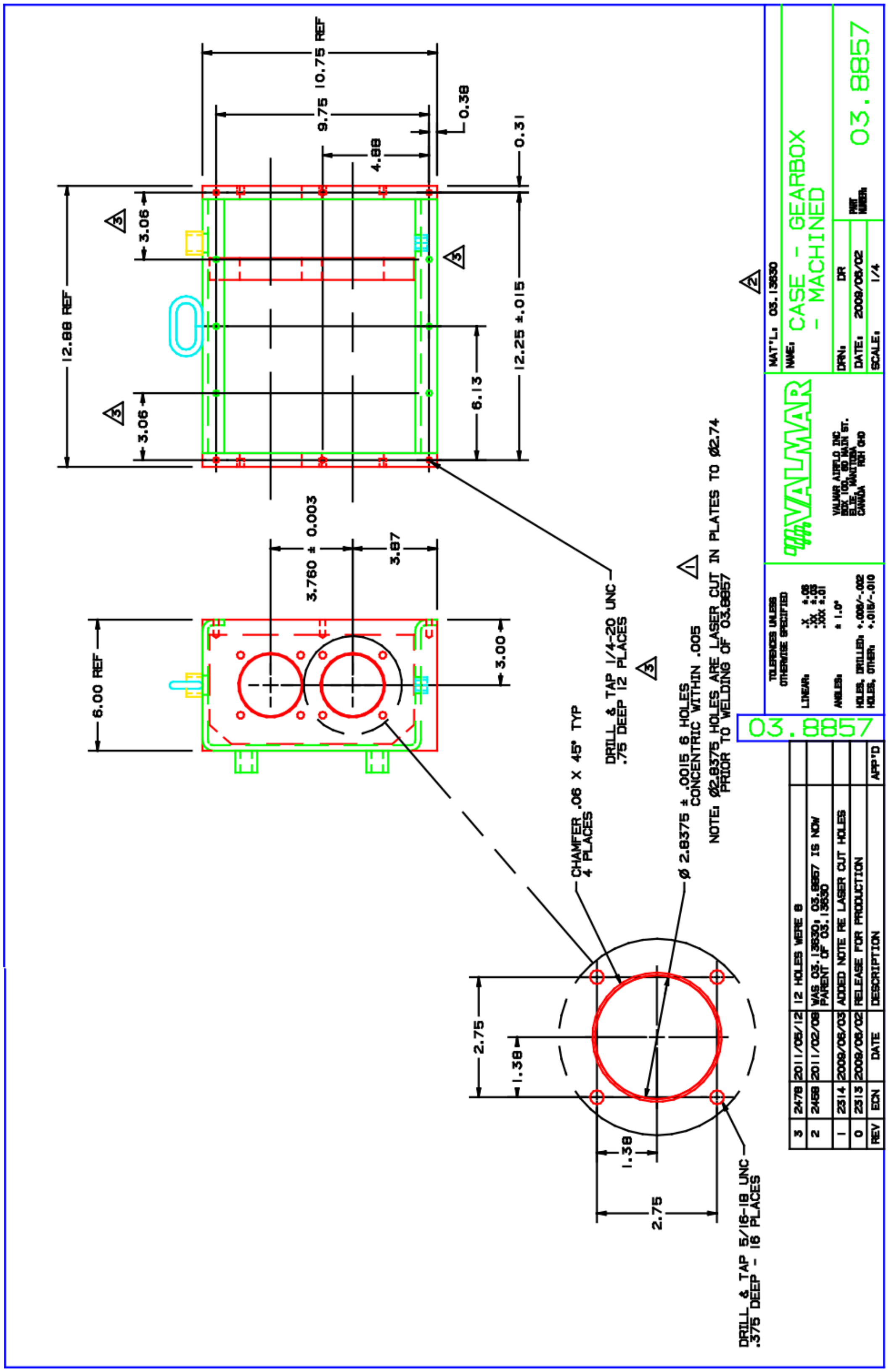


Figure 24 - Gearbox Dimensions

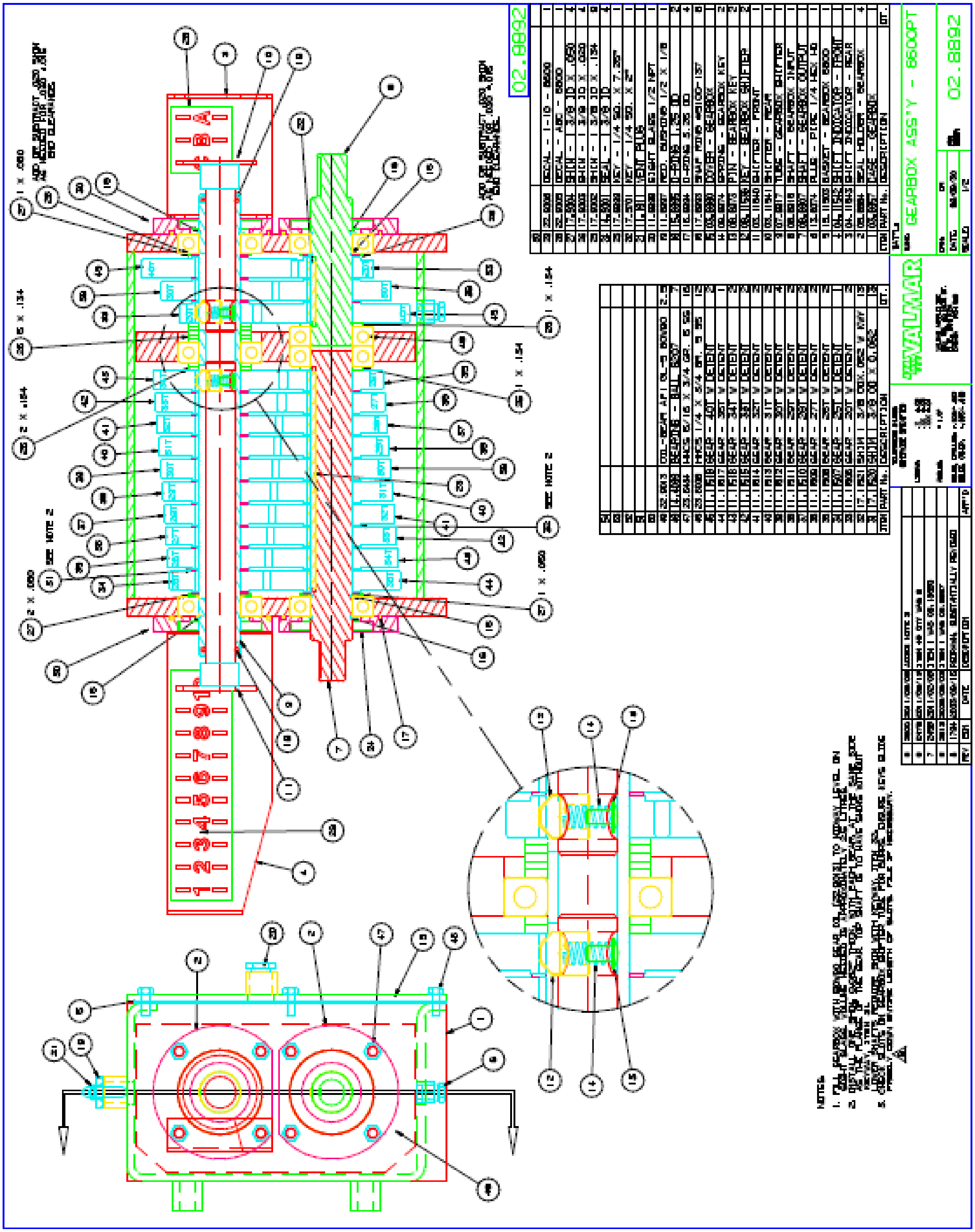


Figure 25 - Gearbox Internals Diagram

Appendix C: Gear Calculations

Appendix D: Sponsor Company Permission

E-mail conversation between team member James Kehler and Valmar representative Dennis Rice for permission to include Company material in final report.

From: drice@valmar.com
To: electricaljk@hotmail.com
Subject: Re: Gear Prices

Date: Fri, 25 Nov 2011 08:22:52

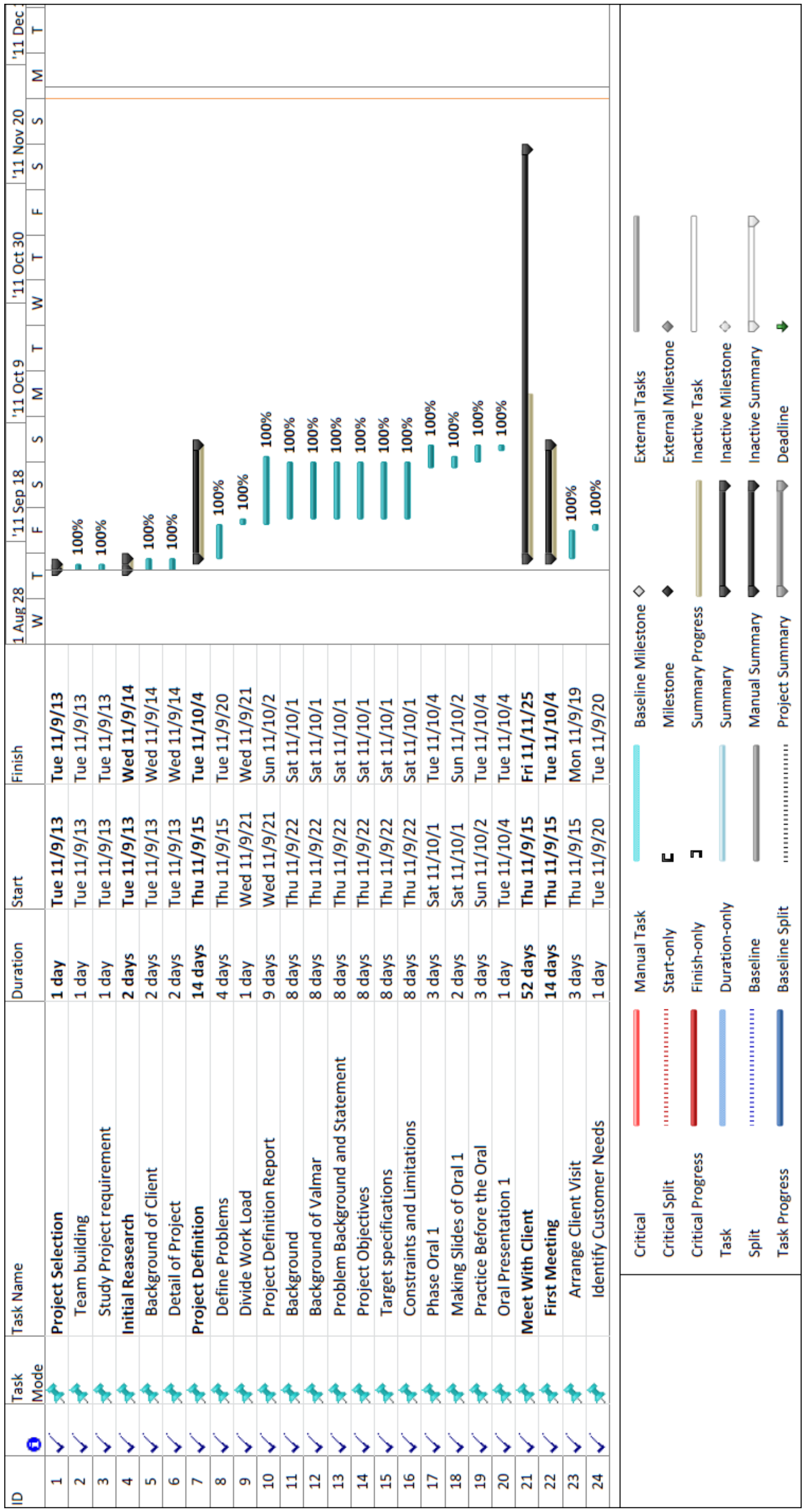
Hi James,

You have our permission to use whatever company material you need in order to complete your report.

Thanks,

Dennis

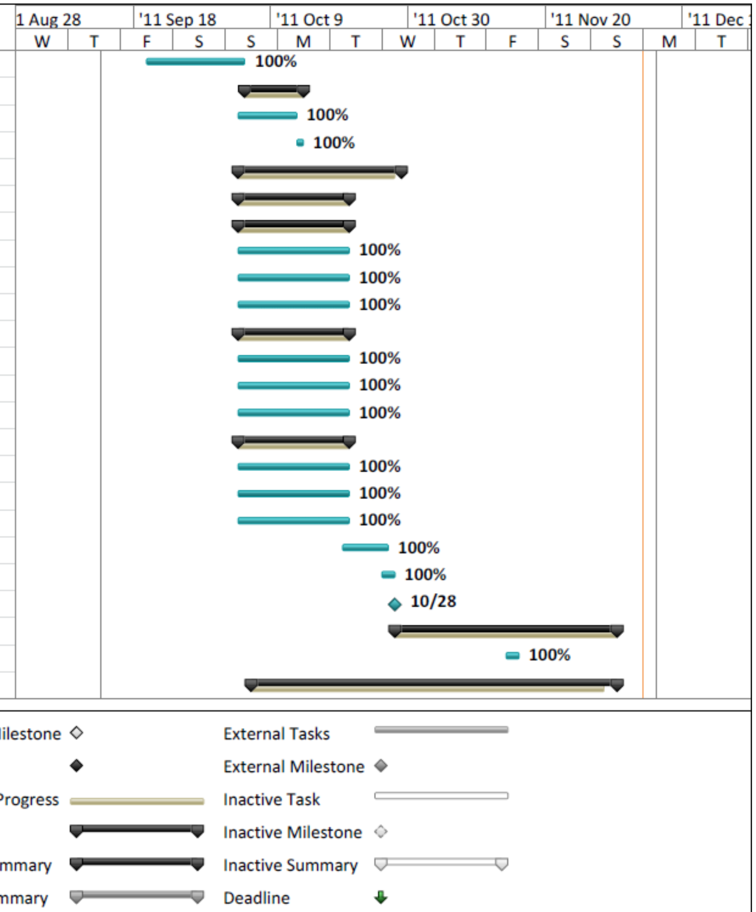
Appendix E: Gantt Chart



ID	Task Mode	Task Name	Duration	Start	Finish
1		**Project Selection**	**1 day**	**Tue 11/9/13**	**Tue 11/9/13**
2		Team building	1 day	Tue 11/9/13	Tue 11/9/13
3		Study Project requirement	1 day	Tue 11/9/13	Tue 11/9/13
4		**Initial Reasearch**	**2 days**	**Tue 11/9/13**	**Wed 11/9/14**
5		Background of Client	2 days	Tue 11/9/13	Wed 11/9/14
6		Detail of Project	2 days	Tue 11/9/13	Wed 11/9/14
7		**Project Definition**	**14 days**	**Thu 11/9/15**	**Tue 11/10/4**
8		Define Problems	4 days	Thu 11/9/15	Tue 11/9/20
9		Divide Work Load	1 day	Wed 11/9/21	Wed 11/9/21
10		Project Definition Report	9 days	Wed 11/9/21	Sun 11/10/2
11		Background	8 days	Thu 11/9/22	Sat 11/10/1
12		Background of Valmar	8 days	Thu 11/9/22	Sat 11/10/1
13		Problem Background and Statement	8 days	Thu 11/9/22	Sat 11/10/1
14		Project Objectives	8 days	Thu 11/9/22	Sat 11/10/1
15		Target specifications	8 days	Thu 11/9/22	Sat 11/10/1
16		Constraints and Limitations	8 days	Thu 11/9/22	Sat 11/10/1
17		Phase Oral 1	3 days	Sat 11/10/1	Tue 11/10/4
18		Making Slides of Oral 1	2 days	Sat 11/10/1	Sun 11/10/2
19		Practice Before the Oral	3 days	Sun 11/10/2	Tue 11/10/4
20		Oral Presentation 1	1 day	Tue 11/10/4	Tue 11/10/4
21		**Meet With Client**	**52 days**	**Thu 11/9/15**	**Fri 11/11/25**
22		**First Meeting**	**14 days**	**Thu 11/9/15**	**Tue 11/10/4**
23		Arrange Client Visit	3 days	Thu 11/9/15	Mon 11/9/19
24		Identify Customer Needs	1 day	Tue 11/9/20	Tue 11/9/20

Figure 26 - Gantt Chart

ID	Task Mode	Task Name	Duration	Start	Finish
25		Technical Detail From Client	11 days	Tue 11/9/20	Tue 11/10/4
26		**Second Meeting**	**7 days**	**Wed 11/10/5**	**Thu 11/10/13**
27		Arrange Client Visit	7 days	Tue 11/10/4	Wed 11/10/12
28		Process of Assembly Study	1 day	Thu 11/10/13	Thu 11/10/13
29		**Concept Design Report**	**19 days**	**Tue 11/10/4**	**Fri 11/10/28**
30		**Brainstroming**	**13 days**	**Tue 11/10/4**	**Thu 11/10/20**
31		**Gear Shaft and Key**	**13 days**	**Tue 11/10/4**	**Thu 11/10/20**
32		Shaft Slot Contains Burrs	13 days	Tue 11/10/4	Thu 11/10/20
33		Precision Fitting for Key	13 days	Tue 11/10/4	Thu 11/10/20
34		Cost	13 days	Tue 11/10/4	Thu 11/10/20
35		**Gasket**	**13 days**	**Tue 11/10/4**	**Thu 11/10/20**
36		Manufacturing	13 days	Tue 11/10/4	Thu 11/10/20
37		Material of Gasket	13 days	Tue 11/10/4	Thu 11/10/20
38		Cost	13 days	Tue 11/10/4	Thu 11/10/20
39		**Gearbox Case**	**13 days**	**Tue 11/10/4**	**Thu 11/10/20**
40		Method of Construction	13 days	Tue 11/10/4	Thu 11/10/20
41		Material of Case	13 days	Tue 11/10/4	Thu 11/10/20
42		Cost	13 days	Tue 11/10/4	Thu 11/10/20
43		Draft of Concept Report	5 days	Thu 11/10/20	Wed 11/10/26
44		Final Concept Report	2 days	Wed 11/10/26	Thu 11/10/27
45		Concept Report	0 days	Fri 11/10/28	Fri 11/10/28
46		**Technical Design**	**24 days**	**Fri 11/10/28**	**Wed 11/11/30**
47		Get Measurements	2 days	Mon 11/11/14	Tue 11/11/15
48		**Build up Model with SW**	**40 days**	**Thu 11/10/6**	**Wed 11/11/30**



ID		Task Mode	Task Name	Duration	Start	Finish
49			Build up the Original Model	16 days	Thu 11/10/6	Thu 11/10/27
50			Build up the Redesign Model	23 days	Thu 11/10/27	Mon 11/11/28
51			Numerical Analysis	5 days	Tue 11/11/15	Mon 11/11/21
52			Comparison with the Original Design	8 days	Mon 11/11/21	Wed 11/11/30
53			**Draft of Final Report**	**15 days**	**Thu 11/11/10**	**Wed 11/11/30**
54			Divide Work Load	1 day	Thu 11/11/10	Thu 11/11/10
55			Draft Writing	12 days	Tue 11/11/15	Wed 11/11/30
56			Draft Report Due	0 days	Wed 11/11/30	Wed 11/11/30
57			**Final Report**	**5 days**	**Wed 11/11/30**	**Tue 11/12/6**
58			Combine the Drafts	1 day	Wed 11/11/30	Wed 11/11/30
59			Improvement 1	3 days	Wed 11/11/30	Fri 11/12/2
60			Improvement 2	2 days	Sat 11/12/3	Mon 11/12/5
61			**Poster Making**	**10 days**	**Wed 11/11/23**	**Tue 11/12/6**
62			Poster Designing	3 days	Wed 11/11/23	Fri 11/11/25
63			Poster Formating	7 days	Sat 11/11/26	Mon 11/12/5
64			Poster Publishing	0 days	Tue 11/12/6	Tue 11/12/6
65			**Final Presentation**	**4 days**	**Wed 11/11/30**	**Mon 11/12/5**
66			Divide Work Load	1 day	Wed 11/11/30	Wed 11/11/30
67			Making Slides of Final Presentation	2 days	Thu 11/12/1	Fri 11/12/2
68			Practice Before the Presentation	2 days	Fri 11/12/2	Mon 11/12/5
69			Final Report	0 days	Mon 11/12/5	Mon 11/12/5