Valmar Metering Drive Redesign Final Report

Sponsoring Company: Valmar Airflo Inc.

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Team Name: Adamu and Associates

Team Number: 8

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Letter of Transmittal

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E2-229 Engineering Bldg
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December 5th, 2011

Dr. P. Labossiere
E1-190 Engineering Bldg.
Winnipeg, MB, R3T 5V6

Dear Dr. Labossiere:

Please find enclosed the report, due December 5th, 2011, that you requested. The title of the report is *Valmar Metering Drive Redesign Final Report*. The purpose of this report is to detail the design solutions Team Adamu and Associates have created in order to improve the current metering drive system used by Valmar Airflo Inc.. The content of the report focuses on the final designs, details and analysis that Team Adamu and Associates believe are the best solutions that satisfies the client's needs. We'd like to thank-you in advance for your time and encourage you to contact Dallas Gade if you have any questions or concerns regarding our project or report.

Sincerely,

Dallas Gade
Team Manager

Dallas_Gade@hotmail.com

Letter of Transmittal

Team Adamu & Associates
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December 5th, 2011

Mr. D. Rice
E1-190 Engineering Bldg.
Winnipeg, MB, R3T 5V6

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Sincerely,

Dallas Gade
Team Manager

Dallas_Gade@hotmail.com

Abstract

Valmar Airflo Inc. is a Manitoba based company that makes granular applicators for various agricultural applications. Valmar's implement-mounted applicators make use of a ground driven gear and chain system to meter product accurately into their pneumatic distribution system. The variation in designs of each implement on which Valmar's products must be capable of mounting to causes difficulty in making the gear and chain connection from the ground to the metering gearbox. Team Adamu & Associates (TAA) has designed two alternate metering drive systems, one using a flexible driveshaft, and the other utilizing a hydraulic drive. The flexible driveshaft system can be installed with minimal changes to the current system and represents the most cost-effective, simple solution. TAA is skeptical as to whether the hydraulic drive can remain accurate over a wide range of ambient temperatures without using expensive components; for this reason, TAA has developed a test procedure for the hydraulic drive to evaluate its accuracy.

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1 Introduction & Problem Description

Valmar Airflo Inc. (Valmar) is a Canadian based company that designs and manufactures agricultural equipment used worldwide, specializing in the production of equipment used for the application of granular products such as fertilizer, seed, herbicide, insecticide, inoculants and forage preservatives. There are two categories of granular dispensers made by Valmar – pull type and implement mount. Pull-type units , such as the one seen in Figure 1, are stand-alone units that connect directly to the back of a tractor. Implement mount units like the one seen in Figure 2 must be mounted to and used in conjunction with an existing piece of equipment such as a harrow bar or cultivator. Although there are other companies that manufacture granular applicators, Valmar is well known in the agricultural world for providing the highest level of accuracy in terms of even product dispersal over the field [1]. This level of accuracy is achieved through a metering system developed by Valmar which controls the amount of product applied to the field at any given ground speed. This project is aimed at redesigning the metering drive system to address some of its current problems.

Figure 1. Pull-type granular applicator [2]. Photo used with permission from Valmar.

Figure 2. Implement mounted applicator [3]. Photo used with permission from Valmar.

Currently, Valmar uses a universal metering drive system that attaches to both their pull-type and implement mounted applicators. This metering drive system, as seen Figure 3, consists of a series of interconnecting gears and chains. The conveyor, which deposits the product into the venturis from the hopper, connects to the metering gearbox. The gearbox is attached to a ground driven wheel via the aforementioned system of gears and chains. As a result of this direct connection, the rate at which the product is dispensed is directly proportional to the ground speed of the agricultural implement. This ensures even distribution of the product regardless of the implement's speed.

Figure 3. Current metering drive system [4]. Photo used with permission from Valmar.

Valmar encounters difficulties when installing their current metering drive system on implement mount units due to physical space limitations as well as the wide variation in implement designs. A cultivator, for example, is designed quite differently than a harrow bar, but Valmar's products must be easily mounted on both. Even the variation in designs between harrow bars from different manufacturers can be problematic. To account for this, Valmar must tailor the system to meet the needs of individual customers, which results in additional cost, labor, and installation time.

The primary objective of this project is to redesign the metering drive system to eliminate the problems described above, while working within the constraints and limitations imposed by the client. Team Adamu and Associates has worked towards designing a universal system that is compatible with all Valmar products without additional custom work and stands behind the ideas presented as being the best possible solutions.

2 Design Details

TAA has come up with two different designs that can be used to replace the existing gear and chain setup. The first design involves utilizing a hydraulic drive system to replace the connection between the ground drive and the gear box. The second design will also replace this connection, but will do so with a flexible driveshaft. Both designs will significantly add to the level of flexibility in locating the ground driven wheel, and should facilitate installations in tight spaces.

2.1 Re-used Components

Several major components from Valmar's current ground driven system will be re-used in both new designs. These components, as labeled in Figure 4 below, include the spring loaded arm, the wheel and hub assembly, and the metering gear box.

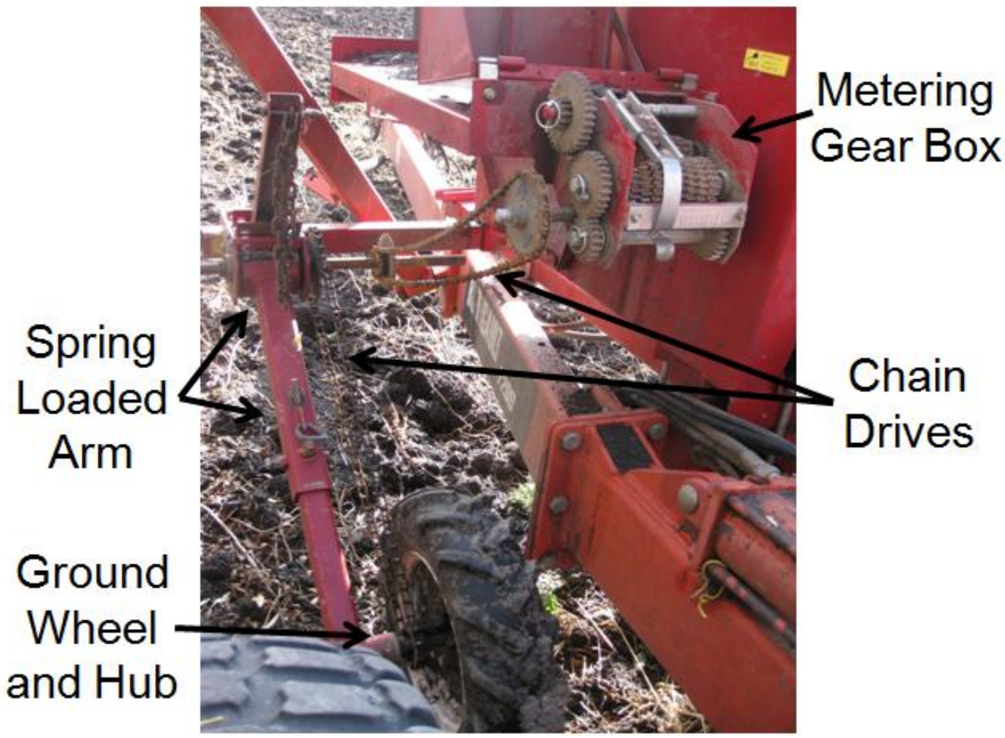


Figure 4. Current metering chain drive design. Photo credit: Rob McDougall.

A final chain reduction system will have to remain in the new designs. This is done to ensure that the applicators maintain their flexibility with respect to their seed distribution capabilities. This final reduction ratio depends on the width of the implement the applicator is installed on and the number of application outlets. Outlets are nozzles where the product is applied to the field along the length of the implement. A wider implement must have a faster-turning conveyor to achieve the necessary product application density. Valmar accomplishes this reduction by making the third drive sprocket, called the "X-Sprocket" (see Figure 5 below), application-specific with respect to its size.

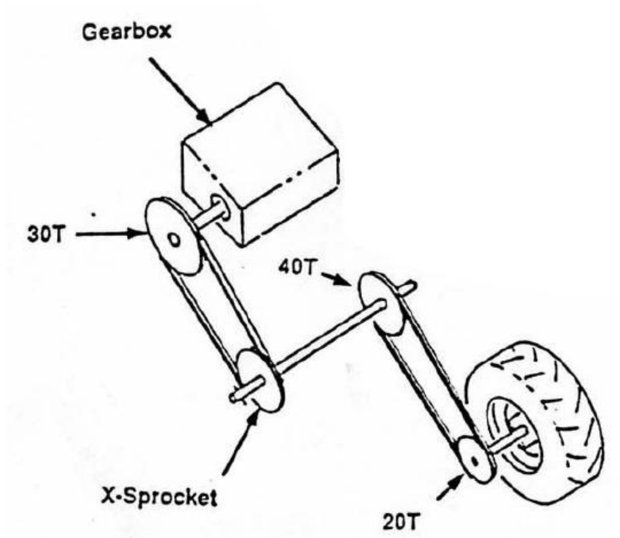


Figure 5. Existing chain drive diagram [5]. Used with permission from Valmar.

The following equation is used to calculate the number of teeth on the X-Sprocket. [5]

$$No.\ of\ Teeth\ of\ X - Sprocket\ = \frac{Implement\ Width\ (inches)}{Number\ of\ Outlets}$$

The X-Sprocket tooth number is then rounded to the nearest available sprocket size. The following example is for a 44 foot wide implement with 20 outlets. [5]

$$\frac{44' \cdot 12"}{20} = 26.4\ \approx 26\ Tooth\ Sprocket$$

The use of the final sprocket and chain reduction does not pose any problems with respect to physical space limitations given the proposed mounting location (see subsequent assembly drawings). It should be noted that further application rate fine tuning is still available to the customer via the 60-speed metering gearbox. The following sections will describe and show both the hydraulic drive and flex shaft drive system. A bill of material will also be included in this section listing the parts in both designs and their cost.

2.2 Hydraulic Drive Design

The hydraulic drive design is very simple in constructions as far as hydraulic systems go. Here, the working medium is pressurized hydraulic fluid. A hydraulic pump is driven by the rotation of the ground wheel and oil is pulled from the reservoir and pushed through the lines to a hydraulic motor. The fluid is forced through the motor, turning the motor's output shaft in the process, before it returns to the reservoir. The pump and motor have the same displacement and should theoretically give a one-to-one drive ratio.

The proposed hydraulic drive design can be seen in Figure 6. Complex fabrication methods are not used to build this system, and only a basic level of mechanical aptitude is required to facilitate installation. The major components of the hydraulic drive system are the oil reservoir, the two Char Lynn 101-1700 hydraulic motors, hydraulic hoses and the mounting brackets to affix the motors, tank and other parts to the agricultural equipment and the applicator's hopper.

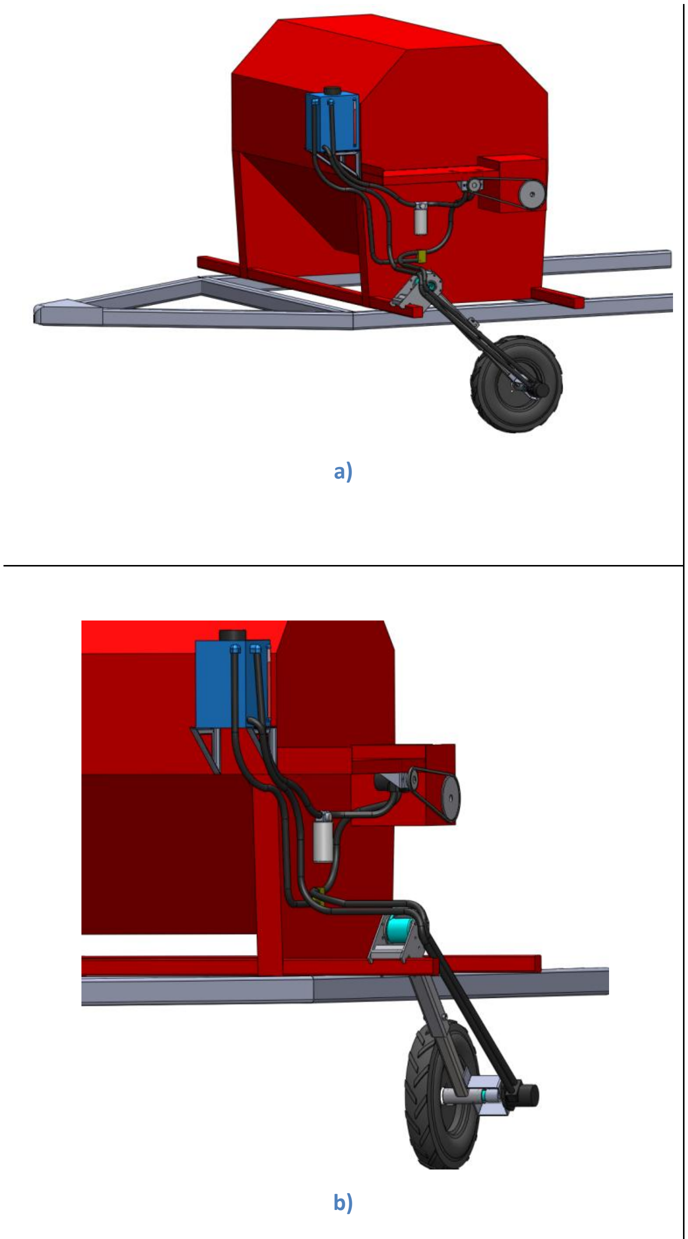


Figure 6. a) Hydraulic drive assembly on the Valmar implement, b) Hydraulic drive assembly close-up on the Valmar implement, c) Hydraulic drive front view

All the parts except for the mounting brackets are readily available off-the-shelf components. Figure 7 and TABLE 1 constitute the bill of material for the system.

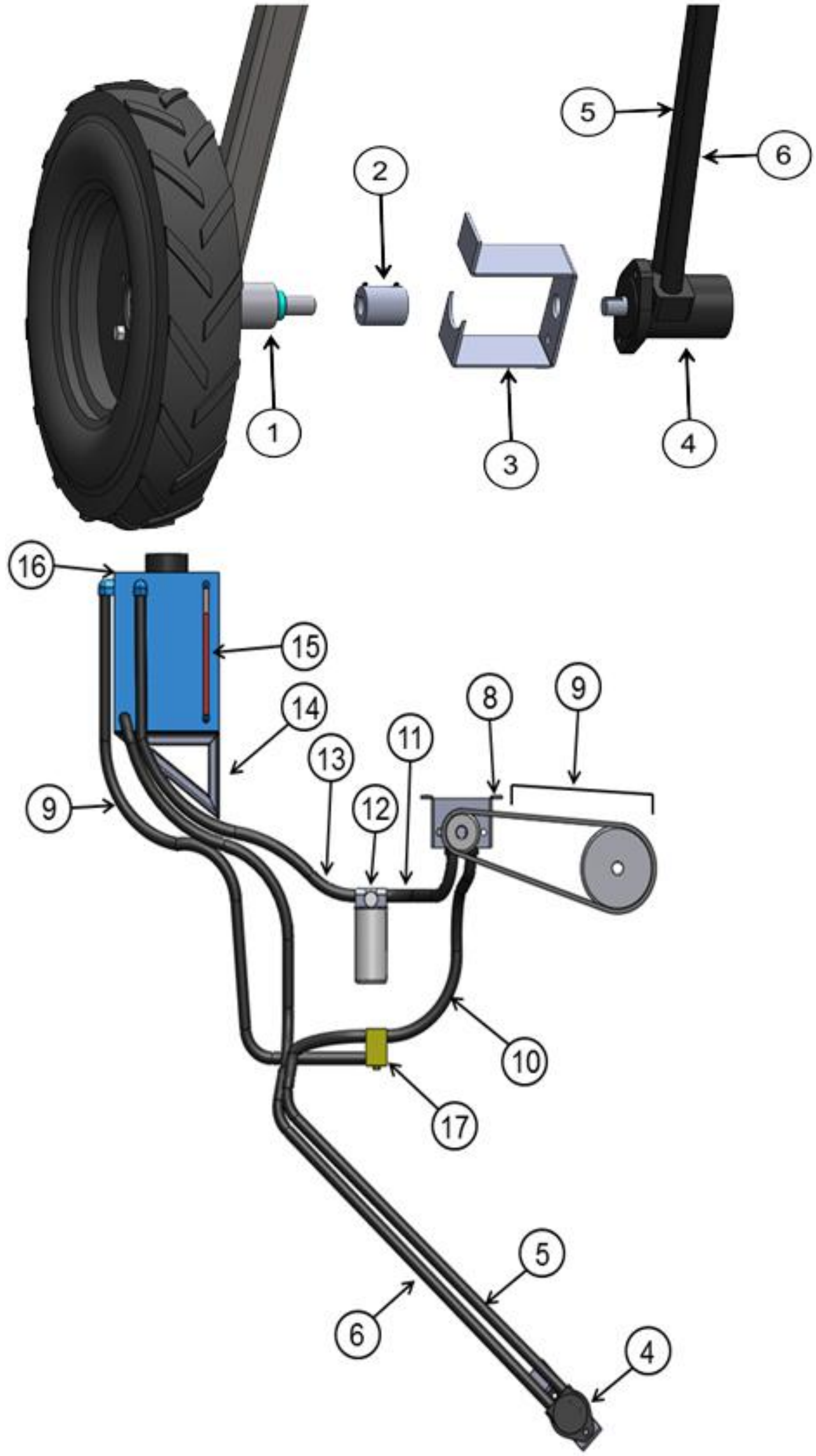


Figure 7. Hydraulic drive itemized assembly. Image credit: Patrick Lessard

TABLE 1. BILL OF MATERIAL, HYDRAULIC DRIVE

Part #	Description
1	Wheel/arm assembly
2	Spider coupling
3	Bracket, pump
4	Pump, Char Lynn model 101-1700
5	Hose, pressure
6	Hose, suction
7	Motor, Char Lynn model 101-1700
8	Bracket, motor
9	Chain assembly with adjustable sprocket size
10	Hose, pressure
11	Hose, return
12	Filter
13	Hose, return
14	Bracket, reservoir
15	Sight gage
16	Reservoir
17	Valve, relief, 2000psi

Note: blue shading indicates custom-made component

The hoses will need to be purchased in bulk and cut to the required lengths based on fitting a specific implement. Unlisted components in the bill of materials include hydraulic fittings and the necessary mounting hardware (nuts and bolts).

Parts 3, 8, and 14 are the mounting brackets that will need to be fabricated for this design. Part 3 and 8 are the pump and motor mounting brackets respectively and are to be fabricated from a thick gauge mild steel sheet. Part 3 is to be welded to the spring loaded arm in front of the bearing hub and part 8 is to be bolted to the underside of the hopper access step. The oil resevoir bracket, part 14, will be constructed from a 1" X 0.125" mild steel equal leg angle stock. The oil reservoir will then be bolted to the top of the bracket. It is important that the reservoir (part 14) be located at a position higher than the pump in order to help pre-charge it.

Part 9 represents a modified version Valmar’s existing chain reduction assembly (as discussed previously). The elimination of the first stage of the chain drive system also eliminates the 2:1 drive ratio it achieved. Replacing this 2:1 ratio is the 1:1 drive ratio between the pump and motor. In order to maintain the same overall ratio as the original chain drive a 15 tooth sprocket will be installed on the hydraulic motor, part 7, and the new final chain drive reduction will feature a Y-Sprocket connected to the gearbox. The number of teeth on the Y-Sprocket can be changed to customize the drive ratio and suit the implement's width and number of outlets. Figure 8 below depicts the motor and its mounting bracket along with the Y-sprocket reduction.

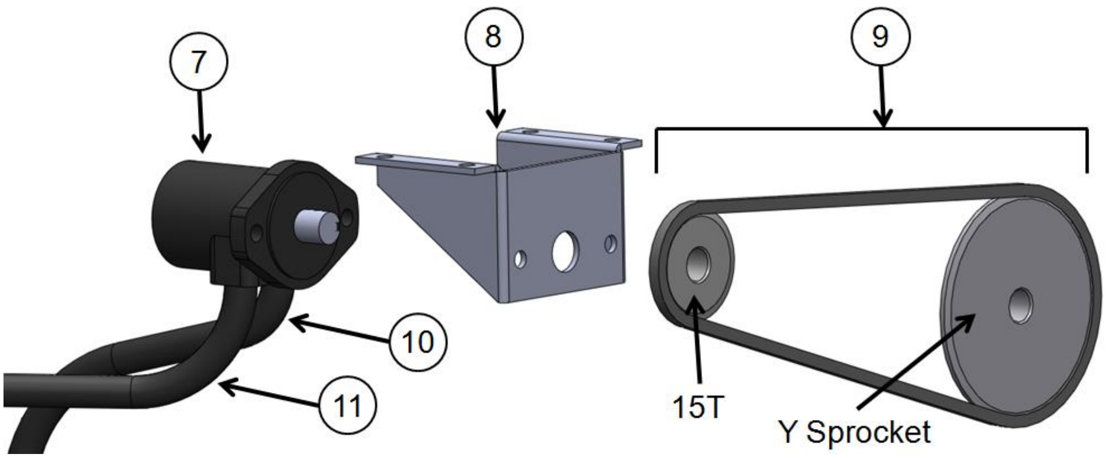


Figure 8. Hydraulic motor gearbox drive assembly.

To calculate the correct amount of teeth neccessary for the Y-Sprocket, the relasionship between the original sprocket speeds and the X-Sprocket must be know.

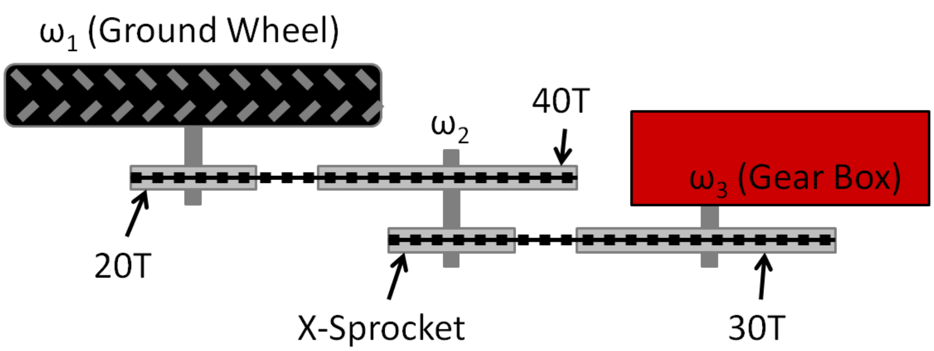


Figure 9. Original Valmar chain drive speeds diagram.

Figure 9 displays the different speeds of every sprocket. The ground wheel speed is denoted by ω_1 and the metering gearbox speed in denoted by ω_3. Note that we are simplifying the use of two nested chain drives to one. In other words the middle shaft speed, ω_2, will not be used. The shaft speed relationships can be represented by:

$$\frac{\omega_1}{20} = \frac{\omega_2}{40} \qquad and \qquad \frac{\omega_2}{X} = \frac{\omega_3}{30}$$

Thus the input, ω_1, and output, ω_3, speed ratio can be represented by:

$$\frac{\omega_1}{\omega_3} = \frac{X}{60}$$

The new chain drive can be represented by the following diagram:

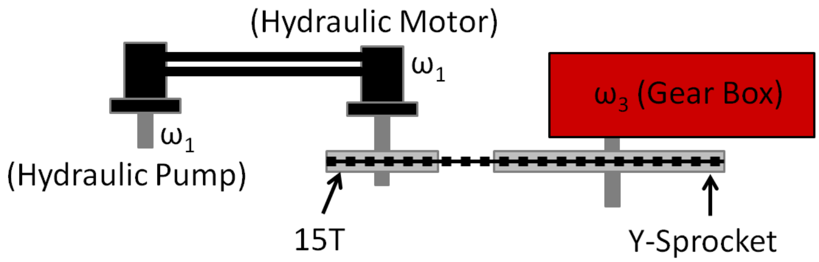


Figure 10. New final chain drive speeds diagram.

The new input and output speed ratio can be represented by:

$$\frac{\omega_1}{\omega_3} = \frac{15}{Y}$$

By equating the input and output speed ratio equations of the new and old system, a relationship can be found between the X-Sprocket and Y-Sprocket drive system.

$$X = \frac{900}{Y}$$

The above relationship can be substituted in the prior X-Sprocket teeth number equation. After rearranging this equation the number of teeth for the Y-sprocket can be calculated, given a 15-tooth input sprocket, via the following equation:

$$\frac{900 \cdot Number\ of\ Outlets}{Implement\ Width\ (inches)} = No.Of\ Teeth\ of\ Y - Sprocket$$

From the original example of an implement of 44' width having 20 outlets, an equivalent drive ratio can be attained by calculating the new Y-sprocket size as:

$$\frac{900 \cdot 20}{44' \cdot 12''} = 34.09 \ \approx 34\, Teeth$$

The advantage of using this type of drive system is the flexibility of where the drive ground wheel can be placed. Hydraulic hoses can be routed in any direction, with only small limitations placed on the permissible bend radii, and can be cut in any length.

The disadvantage of this system is that the oil properties, namely viscosity and density, can change dramatically with temperature. This could lead to the system losing accuracy outside a very small range of ambient temperatures. Valmar's applicators are used from late spring until early fall and large ambient temperature fluctuations can be seen during these months. Valmar claims to have had metering accuracy issues when using a hydraulic drive over 20 years ago, crediting this problem to oil temperature fluctuations in the system. TAA's recent correspondence with French agricultural company Gregoire Besson revealed that the company has a prototype hydraulic metering drive system in use on their own products and have been pleased with its performance. This would seemingly come as evidence that such a system *could* work for Valmar. However, Gregoire Besson's applications do not require the level of metering accuracy required by Valmar. The accuracy of the hydraulic system cannot be determined analytically due to the complexity of the heat transfer modeling involved. A full analytical background on the potential metering accuracy problems is discussed in Appendix A. A test procedure will be presented subsequently in order to determine the feasibility of the hydraulic system based on metering speed consistency with variation in ambient temperature.

2.2.1 Selection of Components

Every hydraulic system contains a few main components – a pump, reservoir, filter, and actuator. Even in designing simple hydraulic systems such as the hydraulic metering drive under consideration, the selection of suitable components is important. A brief discussion of the major system components will be carried out in this section, along with justification of the component selection.

2.2.1.1 Pump Selection

Typical hydraulic systems employ the use of gear or piston pumps, with vane pumps being less popular due to their generally lower-pressure capabilities. The choice of pump depends on many factors including system complexity, required flow rate, system pressure, fluid contamination limitations, and perhaps most importantly, cost. Typically, gear pumps are more tolerant of contamination before wear becomes an issue as compared to piston pumps as a result of the larger clearances characteristic of gear pump construction. These larger clearances, however, come at the expense of volumetric efficiency [6]. Piston pumps are generally more expensive than gear pumps due to the number of moving parts as well as tighter clearances. Given that the system in question will be operated in a farm field, it can be assumed that it will be subject to significant contamination (airborne dust and dirt), and thus a gear pump is preferable on that front. The lower cost of gear pumps comes as further justification, and thus a gerotor-type internal gear pump was chosen. Figure 11 depicts the typical gerotor pump or motor construction.

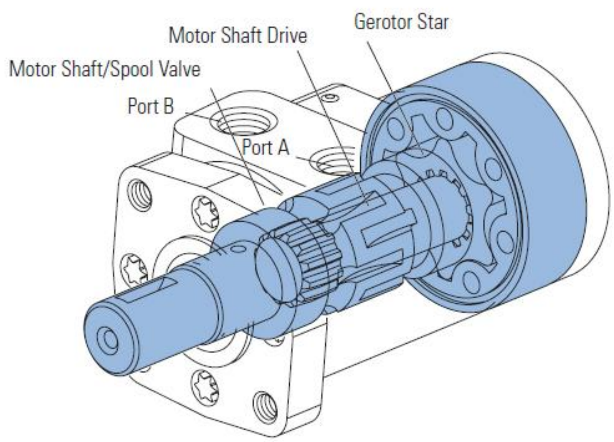


Figure 11. Typical gerotor pump/motor construction [7]. Permission pending.

Performance curves supplied from the pump manufacturer must be consulted to determine whether a given pump is well suited for a given application. Figure 12 depicts a typical gerotor-pump performance curve. It should be noted that this curve was taken from a different pump manufacturer versus the pump to be used. While the data is for a pump of the same displacement, it might differ *slightly* from the pump that was selected; however, for the purposes of this analysis it will be used interchangeably.

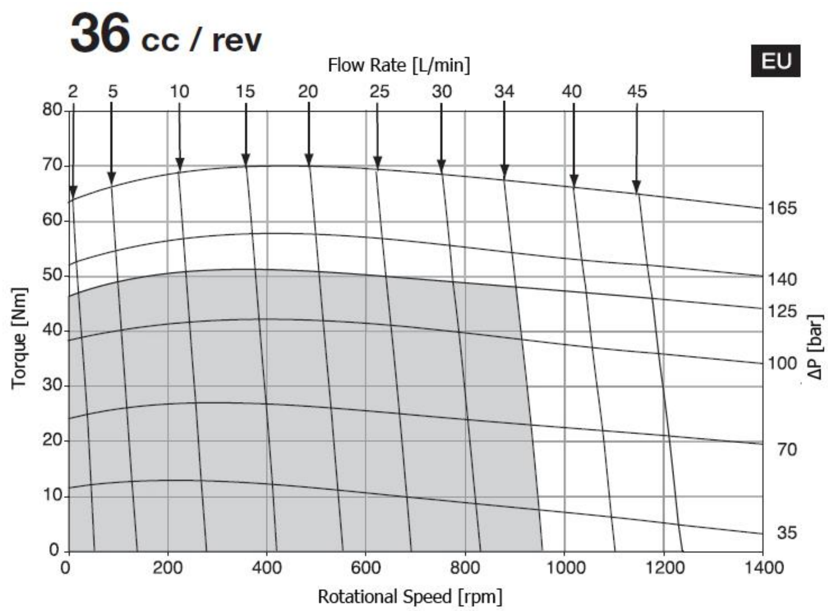


Figure 12. Gerotor motor performance curves[8]. Permission pending.

2.2.1.2 Flow Rate & Hose Sizing

One important design factor to consider is the system's volumetric flow rate. In a positive displacement pump such as a gear pump, it is a simple extrapolation to determine the volumetric flow rate based on the pump displacement. The theoretical flow rate can be determined using the following formula.

$$Q_t = V_g N 2\pi/60$$

Where:

$$Q_t = Theoretical\ flow\ rate$$
$$V_g = displacement$$
$$N = rotational\ speed\ of\ the\ pump$$

Using pumps that Valmar currently stocks, having 36 cm^3/rev displacement, the flow rate can be determined to be 1.2 gpm, given a 10 mph tractor ground speed, corresponding to a pump speed of 150 rpm. While there is no flow rate requirement in this application as there is in most hydraulic systems, the flow rate will determine the required hose sizing. Large flow velocities should be avoided to reduce the pressure loss in the line. Fluid velocity can be found by:

$$v = \frac{Q}{A}$$

Where:

$$v = fluid\ velocity$$

Standard recommended upper-limits for line velocities exist for aiding in determining hose sizing. Figure 13 provides a graphic means of determining proper hose sizing. Drawing a line from the flow rate on the left to the maximum recommended line velocities on the right yields an intersection with the hose inner diameter (ID) line in the middle of the chart; the point of intersection on the hose ID line corresponds to the hose size required. From Figure 13 it can be seen that suction hoses require a lower line velocity; it is good design practice to make suction hoses large to further reduce pressure drop that can cause pump cavitation. From the lines drawn on Figure 13 it can be seen that the suction hoses will have to be 3/8” ID. The ID of the pressure hose, according to the chart, could be safely chosen as 3/16”, but will also be chosen to be 3/8” given that reducing pressure loss is important in this design (see Appendix A).

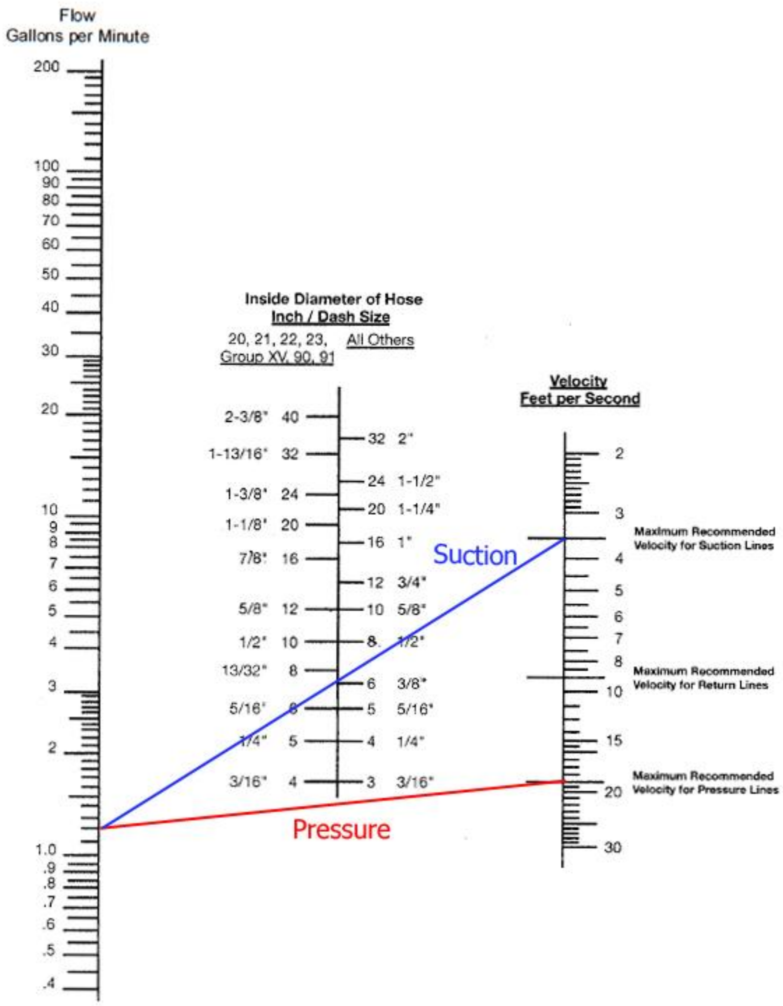


Figure 13. Hose sizing chart [9]. Permission pending.

2.2.1.3 Fluid Selection & Viscosity

Figure 12 also shows some other behaviors that must be accounted for. Theoretically, the flow rate at any given rotational speed should remain constant regardless of load in the form of pressure; however the flow rate lines are not vertical as would be expected. Instead, they have a slightly negative slope; this is the effect of internal leakage that is accompanied by a pressure increase. Viscosity, defined as the resistance of a fluid to flow, is assumed to be constant in Figure 12, but will also affect the amount of leakage. Leakage is described as the amount of fluid that gets recirculated through the pump's clearances that does not contribute to the work done by the pump. Leakage is undesirable, but unavoidable in any style of pump due to the operating clearances present. Leakage flow rate can be quantified using the following equation [10]:

$$Q_L = \frac{k * \Delta P}{\mu}$$

Where:

$Q_L = Pump\ leakage\ volumetric\ flow\ rate$

$\Delta P = Pressure\ rise\ across\ the\ pump$

$k = Pump\ geometry\ constant$

$\mu = dynamic\ viscosity\ of\ the\ working\ fluid$

The actual volumetric flow rate, Q, can be found by subtracting the leakage flow rate from the theoretical flow rate. It can be inferred from the equation that leakage increases with both an increase in pressure rise across the pump, as well as a decrease in viscosity. Fluid viscosity will be a major concern in the design of the hydraulic system. As the system runs, the temperature of the oil will increase due to line friction and leakage through the pump, as well as heat transfer with the external environment (see appendix B.1). The rate of heat generation due to leakage, q_L, can be expressed by the following formula.

$$q_L = Q_L * \Delta P$$

Figure 14 shows temperature-viscosity curves for three grades of automatic transmission fluid, but can be assumed to be representative of the behavior of all hydraulic fluids. As the fluid temperature rises, the viscosity of the fluid decreases. This will cause the leakage through the pump to increase, and will further increase the temperature of the fluid. Due to the instability of this cycle, it is important to keep the oil viscosity within the pump manufacturer’s predefined limits by ensuring the operating temperature of the system remains within corresponding limits.

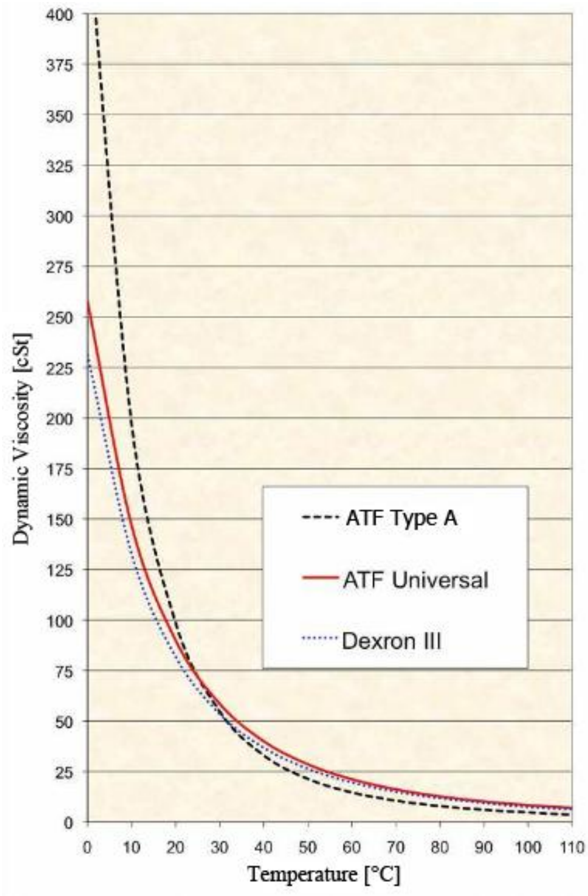


Figure 14. ATF temperature vs. viscosity [4]. Permission pending.

In order to minimize the leakage through the pump, it is important to choose the correct working fluid to be used. Oils are classified on the basis of viscosity grade (VG) and viscosity index (VI). The viscosity grade of the oil is a measure of the oil's resistance to flow (viscosity); the higher the grade, the higher the viscosity. Viscosity index is a relative measure of the oil's change in viscosity as the working temperature changes. A high viscosity index means that a change in temperature will have a small change in viscosity, and is desirable in most applications to minimize leakage and thus reduce power losses [11]. The power loss is of little consequence here, given that the metering drive wheel harnesses virtually "free power" from the forward motion of the tractor; however, the metering accuracy will be directly affected by a significant change in fluid viscosity. For these reasons, oil with high viscosity grade and index such as Automatic Transmission Fluid (ATF) will be selected [12].

2.2.1.4 Motor Selection

While hydraulic pumps create flow via rotation of the pump shaft, hydraulic motors perform the opposite duty; that is, they receive pressurized fluid from the pump that is then used to rotate the output shaft and create torque on the shaft before the oil is expelled to the reservoir. In short, pumps *require* shaft work, while motors *create* shaft work. The construction of pumps and motors is very similar; gear motors and piston motors look nearly identical to similar sized pumps of the same type in many cases, where only subtle features differentiate the two. Most motors are designed to be bi-directional, meaning that they can be spun either clockwise or counterclockwise. Some motors can be driven as pumps, but most pumps cannot be used as motors. Figure 11 actually depicts a hydraulic motor, but since the motor depicted can be driven as a pump, it is not technically incorrect to refer to it as a pump.

The motor in Valmar's hydraulic drive system will be subjected to the same fluid as the pump, and thus, the same contamination compliance requirement applies. It is thus a simplifying decision to select the same two motors, both gerotor type and of the same displacement, to act as motor and pump. The Char Lynn model 101-1700 gerotor motor Valmar stocks is suitable.

2.2.1.5 Torque vs. Pressure

The load (torque) on the output shaft of the hydraulic motor results in pressurization of the fluid within the motor. The amount of pressure present is directly proportional to the torque required to turn the shaft – a larger torque requirement creates more pressure. This relationship can be verified from Figure 12, the motor's performance curve. The ratio between output torque and internal pressure varies from motor to motor; that is, motors of larger displacement can achieve a given output torque at lower pressures. In general, motors with larger displacements are capable of greater output torques. Figure 15, below, demonstrates the effect of displacement; in the 36 cc/rev motor on the left, a

50 Nm torque requires 140bar of pressure differential, whereas in the 130 cc/rev motor, a 50 Nm torque is achieved at a mere 50bar pressure differential.

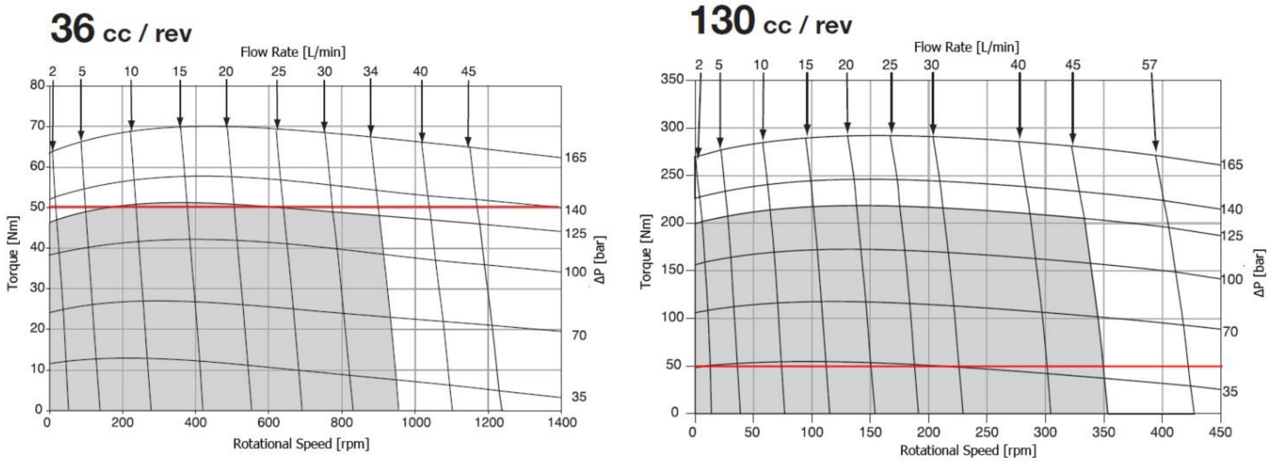


Figure 15. Gerotor motor performance curve comparison [8]. Permission pending.

Given the estimated torque required to turn the metering conveyor of 14 lb ft (19 Nm), the estimated system pressure can be found to be approximately 53 bar, or 770 psi. Any pressure under 1000 psi is low in hydraulic systems, and thus there should be no issue using the small 36 cc/rev motor.

2.2.1.6 Reservoir Considerations

The size and location of the oil reservoir can have significant effects on the system performance. The main functions of a reservoir are to dissipate heat, remove contaminants and fundamentally provide oil storage. The hydraulic drive system's oil-holding capacity, as determined by summing the volume of fluid stored in all hoses, and the pump and motor, is small (much less than 1 gallon); as such, a 1 gallon reservoir would be sufficient. The location of the reservoir needs to be considered both in physical placement and system placement. The tank should be located physically higher than the pump to allow gravity to naturally prime the pump. The tank should also be placed directly before the oil pump in the system with no other components in between. This will further reduce the pressure drop in the suction line and thus decrease the chance of cavitation occurring in the pump. Even though cavitation is nearly impossible at such low design speeds (150 rpm), it is still good practice to adhere to the preceding methodology. Suction lines should draw from slightly above the bottom of the tank to prevent sucking fluid in which a high concentration of contaminants may have settled. The return line(s) should be above the highest oil level of the tank to prevent a siphon effect from occurring when the pump is stopped. The return line should also be as far removed from the suction line as possible. The reservoir should feature an integral filter in the filler/breather assembly such that airborne contaminants (dust, dirt etc.) cannot enter via the filler cap.

2.2.1.7 Filtration

A properly selected filter is crucial to prevent pump and motor wear. Pump manufacturers typically provide minimum filtration standards which must be adhered to in order to ensure that mechanical wear remains minimal. The filter is to be placed in the return line upstream of (and external to) the tank; in this way it will be easily accessible for maintenance purposes. The filter element mesh size should be selected based on the oil cleanliness code requirements outlined by the pump manufacturer; in this case, a 10 micron mesh size is acceptable.

2.2.1.8 Relief Valve Selection

A relief valve, while not a primary system component per say, is crucial in any hydraulic system. Acting only to ensure the wellbeing of the pump and motor, a relief valve's function is to divert flow back to the tank at low pressure in the event that system overpressure occurs for any reason. It is difficult to conceive of any scenario involving the metering drive system in which the pressure would build to the point of needing a safety valve given the low service pressures. However, overpressure is still possible, and thus it is still good practice to include a relief valve; failure to do so could result in pump or motor damage, personal injury, and crop contamination, all of which are highly undesirable. A non-adjustable, 2000psi poppet-style in-line relief valve is recommended for its low cost and ease of installation.

2.2.1.9 Hose Selection

Hydraulic hose is rated according to how much pressure it can withstand. Though the system pressure is expected to be quite low at ~750psi, a medium pressure hose is recommended. At minimum, the pressure rating of the hose must be equal to the relief valve setting, or hose failure is possible before the relief valve pressure setting is reached. Pressure, return, and suction hose should all be 3/8" as determined in section 2.2.1.2. In routing the hose, attention must be paid to the minimum allowable bend radius provided by the manufacturer; failure to do so could lead to premature hose failure.

Figure 16 shows a hydraulic schematic of the proposed system.

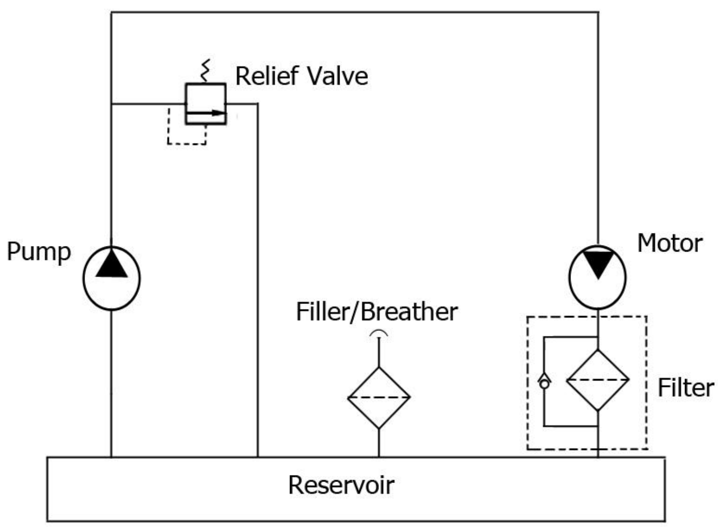


Figure 16. Hydraulic Drive Schematic. Image Credit: Rob McDougall

2.2.2 Accuracy Testing

Given the uncertainty associated with the heat transfer modeling (presented in appendix A-1.2), a physical test that would simulate the system in operation would be desirable. The goal of such a test would be to determine how much the hydraulic motor's shaft speed could be expected to vary with ambient temperature. A test procedure must be carefully devised in order to ensure the results are representative of those to be expected under actual operating conditions.

2.2.2.1 Apparatus

The recommended test is not one that can be run indoors. It was initially suggested that system fluid temperature could be varied through heating of a fluid reservoir on a hot plate and monitored with a thermometer. While this would allow for control of fluid temperature, the heat transfer from the system taking place indoors at room temperature will vary greatly from that occurring outdoors, as would be experienced in operation. This method would be perfectly acceptable if the range of fluid operating temperature was known; however, this is unknown and difficult to estimate, and testing system performance at a fluid temperature higher than the maximum temperature that would be expected in the field would yield misleading test results. By running the test in still outdoor air, the heat transfer taking place should theoretically mimic the worst-case scenario in the field. The resulting system fluid temperature in the test scenario should thus mimic the temperature that could be expected in the field – this is crucial to obtain meaningful test results.

The hydraulic drive system, mounted on a hopper assembly, can be used as-is for the test with one change; given that the test is to take place in a stationary setting, the metering drive wheel cannot be used to provide power to the pump. Thus, an electric motor should be used to spin the pump at a speed equal to the speed of metering drive wheel at the maximum tractor ground speed (approximately 150rpm – corresponding to a 10mph ground speed). This can be achieved via a simple gear/chain or belt/sheave speed reduction if a variable speed motor cannot be obtained for testing. It is important that the pump speed (and thus system flow rate) be equal to the pump speed that can be expected in practice. The reason is that flow rate effects pressure drop and thus heat generation. A simple optical laser tachometer could be used here to measure pump and motor shaft rotational speeds [13]. The test can be run with an empty hopper (i.e. not dispensing product). This should not significantly change the torque and thus system pressure required to turn the metering gearbox. Figure 17 depict the experimental test setup.

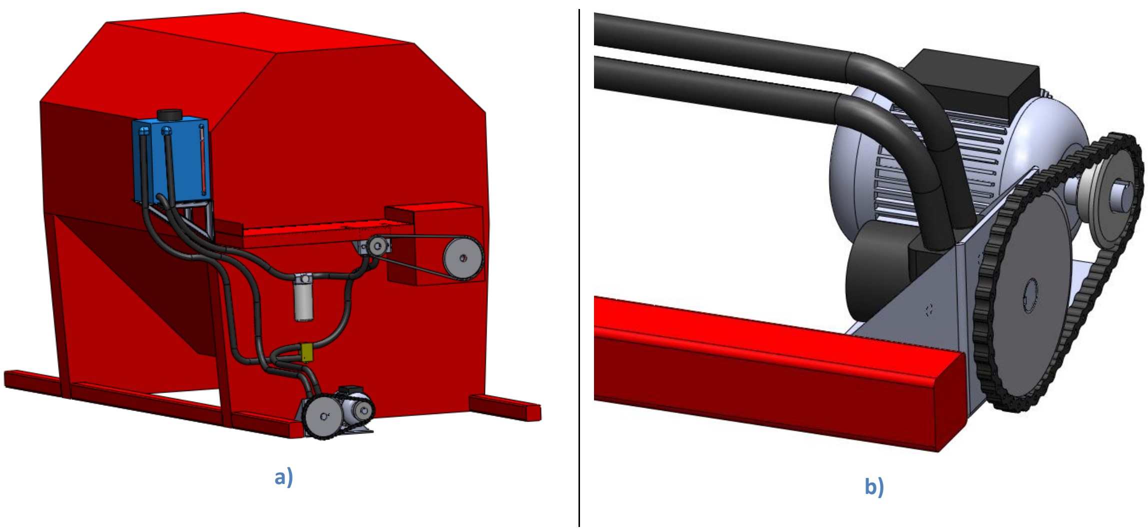


Figure 17. a) Experimental test setup. b) Electric motor close-up showing gear reduction.

2.2.2.2 Procedure

The test should be run in all seasons that the applicator can be expected to be used. These include the seasons of spring through fall, with ambient temperatures ranging from 0 to 35°C. Multiple tests should be run at each range of ambient temperature to ensure repeatability of data. Ambient air temperature should be recorded. The electric motor should be started and the equipment should be allowed to run for 1-2 hours, or the time it takes for the time it takes for system temperature to stabilize (as indicated by the thermometer built into the reservoir’s sight gage). It is crucial that the system temperature is allowed to stabilize to maintain validity of the resulting experimental data. After system temperature has stabilized, shaft speed measurements should be taken using the laser tachometer. TABLE 2, below, depicts the form of the table that should be used to record results.

TABLE 2. EXAMPLE TEST RESULTS FORMAT

Date [mm/dd/yyyy]	Ambient Temp [°C]	Elapsed Time [hrs]	Fluid Temp [°C]	Pump Speed [rpm]	Motor speed [rpm]	Speed difference [%]
04/28/2012	3	1.5	65	148	135	8.8

The speed difference should be calculated as follows:

$$Speed\ Difference\ [\%] = \left(\frac{Pump\ Speed - Motor\ Speed}{Pump\ Speed}\right) x100$$

2.2.2.3 Interpretation of Results

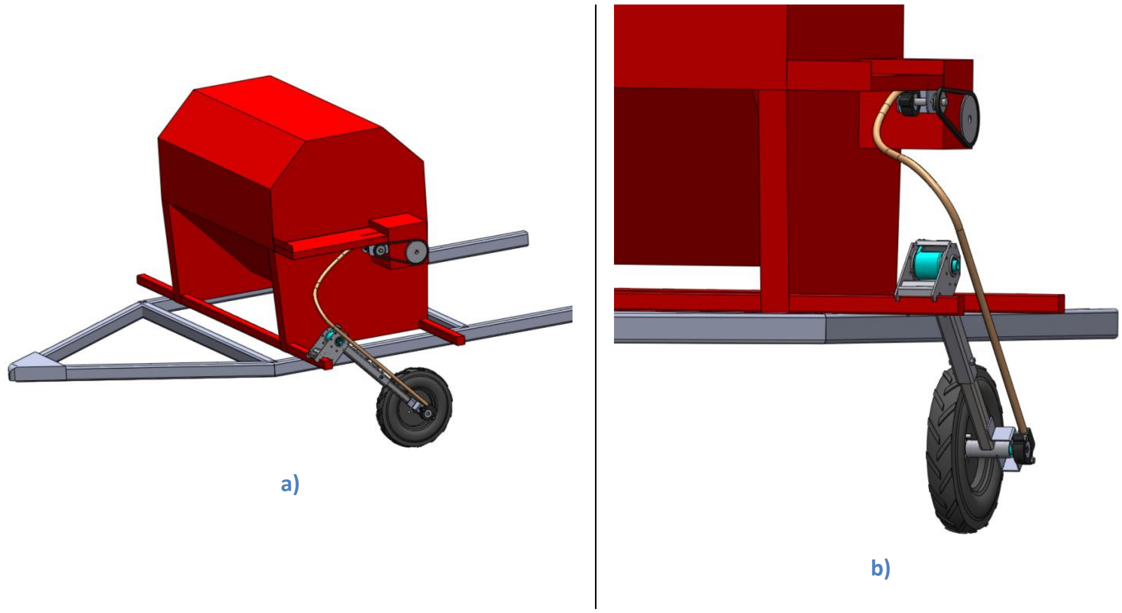
The results of the test should provide a very straightforward means of verifying whether the system can maintain its accuracy to within 5%. Since the hydraulic motor shaft speed is directly proportional to the conveyor speed, a given percentage change in hydraulic motor speed will result in the same percentage change in conveyor speed. It is possible that the system may remain accurate at low ambient temperatures and become inaccurate above a given ambient temperature, and analysis of test results should verify this.

While not shown in the hydraulic schematic, it is possible to enlist in the use of temperature compensation to ensure that a consistent flow rate is maintained. Temperature compensation could come in the form of either a cooler or radiator setup, but either would require an alternate power source and is thus undesirable. An alternative means of compensating for fluid temperature changes would be through the use of a temperature-compensated flow control valve. Such a valve, as the name implies, can be pre-set to maintain a given flow rate regardless of the temperature of the fluid. These valves are specially designed to compensate for viscosity changes. However, they are also extremely expensive, and would significantly increase the system's cost (see section 3 for cost analysis).

2.3 Flex Shaft Drive Design

The flex shaft drive design, shown in Figure 18, offers features similar to the hydraulic drive system in that it can be routed in awkward areas where a rigid drive shaft cannot. During the design research and brainstorming, a product from Elliot Manufacturing® called the FlexSeeder System [14] was found. This flexible drive shaft is fully enclosed and ready for implementation in any system. The FlexSeeder System is designed to replace chain or belt drives in agricultural seeding applications. Power is transferred through an enclosed flexible cable from the ground wheel to the metering gearbox. The major components of Valmar's granular applicator are being used as a base for this drive system.

A special feature of the Elliot Manufacturing® FlexSeeder System [14] is the use of gearboxes at both the input and output ends of the flex shaft. These gearboxes are mounted tangentially to the flex shaft and deliver power perpendicularly to the driving shaft. This eliminates the need for the flex shaft to be in-line with the ground wheel's axle to facilitate coupling, eliminating the issue of protrusion of the flex shaft and allowing the shaft to be run along the spring arm.



a)

b)

Figure 18. a) Flexshaft design on Valmar 55 series applicator, b) Flexshaft design on Valmar 55 series applicator close-up view. Permission pending

There are far less parts in this design compared to the hydraulic system as can be seen from the bill of material (Figure 19, Figure 20, Figure 21). The only items that need purchasing are a 1" I.D. flange bearing (part 2) and the FlexSeeder System which include parts 1, 8, and 10. In reality these parts come as one unit assembly. Fasteners are not shown in this assembly.

The parts that need to be fabricated are similar to the ones in the hydraulic drive design, with the exceptions of the input and output shafts. The parts that need fabricating are listed below:

- flex shaft output bracket
- flex shaft output bracket
- flex shaft hex input shaft
- flex shaft input bracket

The input and output mounting brackets for the flex shaft are similar to the hydraulic drive system. They are constructed of sheet metal which is bent into the desired shape. The two input and output shafts are fabricated out of 1 inch round bar stock which is keyed on one end to receive either the ground wheel or the input sprocket for the output chain drive and the other end needs to be hexed to be able to be inserted into the flex shaft carrier. A more detailed description of the fabrication methods will be included in Appendix B. The Bill of Materials for the flex shaft design can be found in TABLE 3.

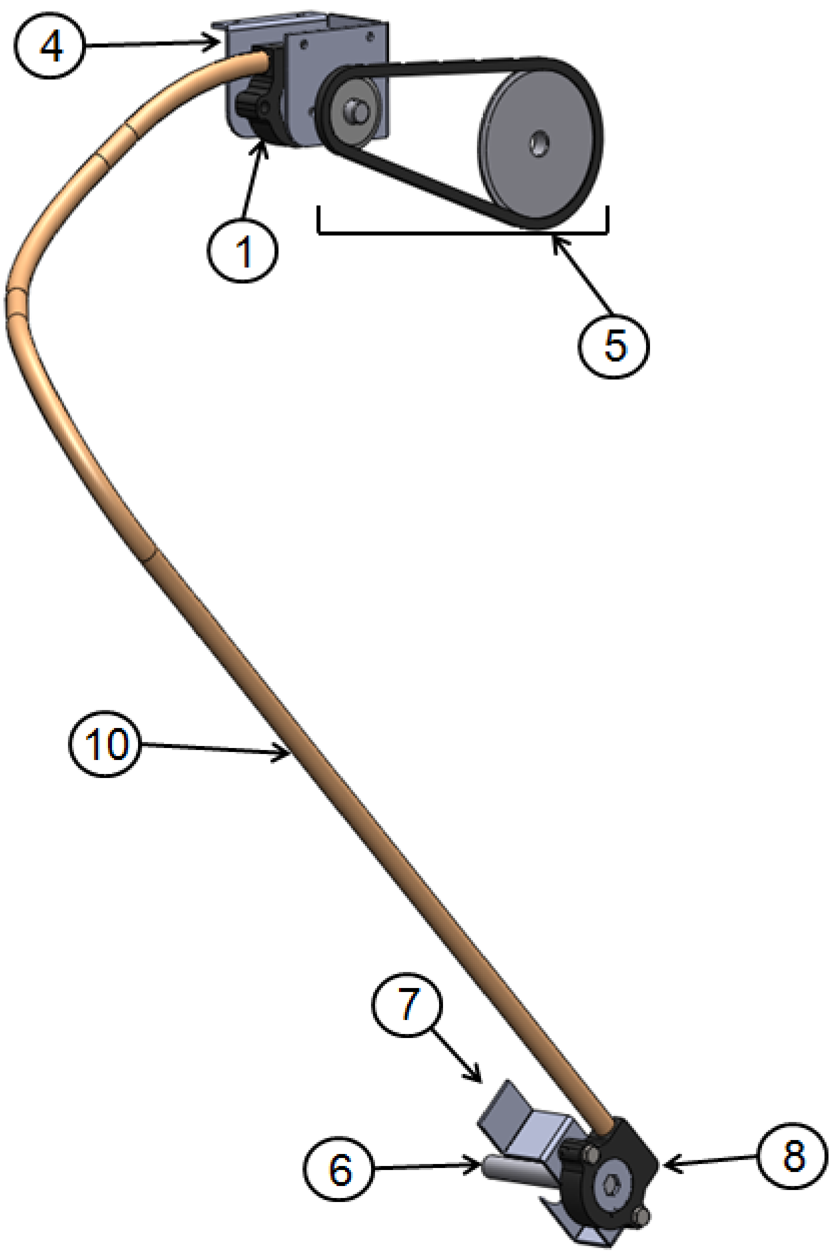


Figure 19. Flex shaft design. Permission pending

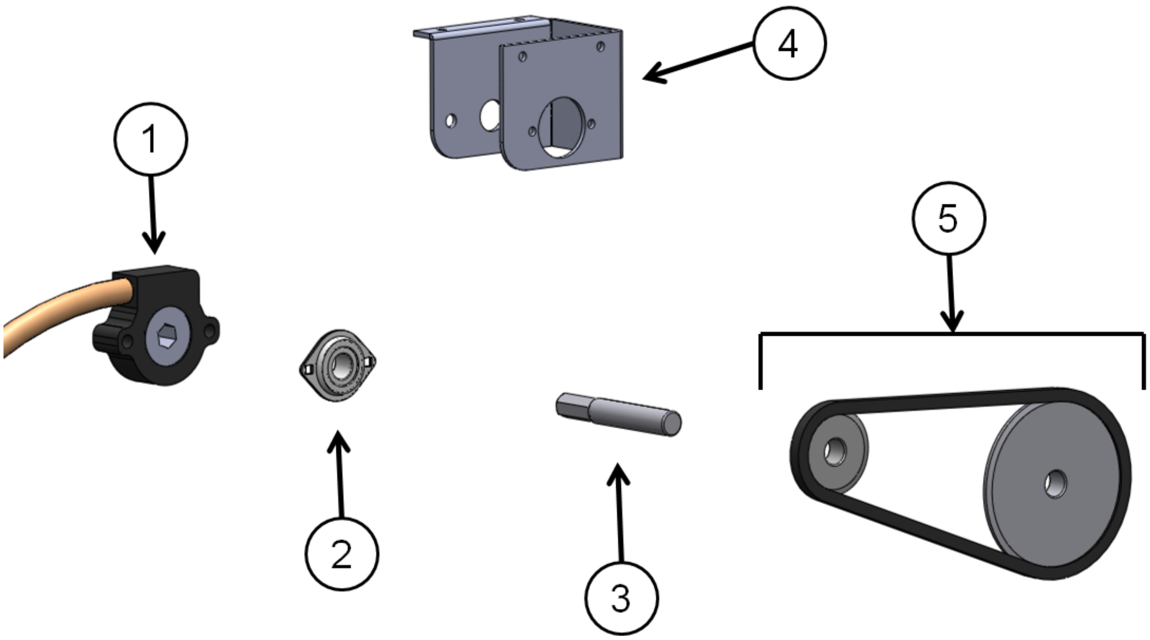


Figure 20. Exploded flex shaft drive design output assembly. Permission pending

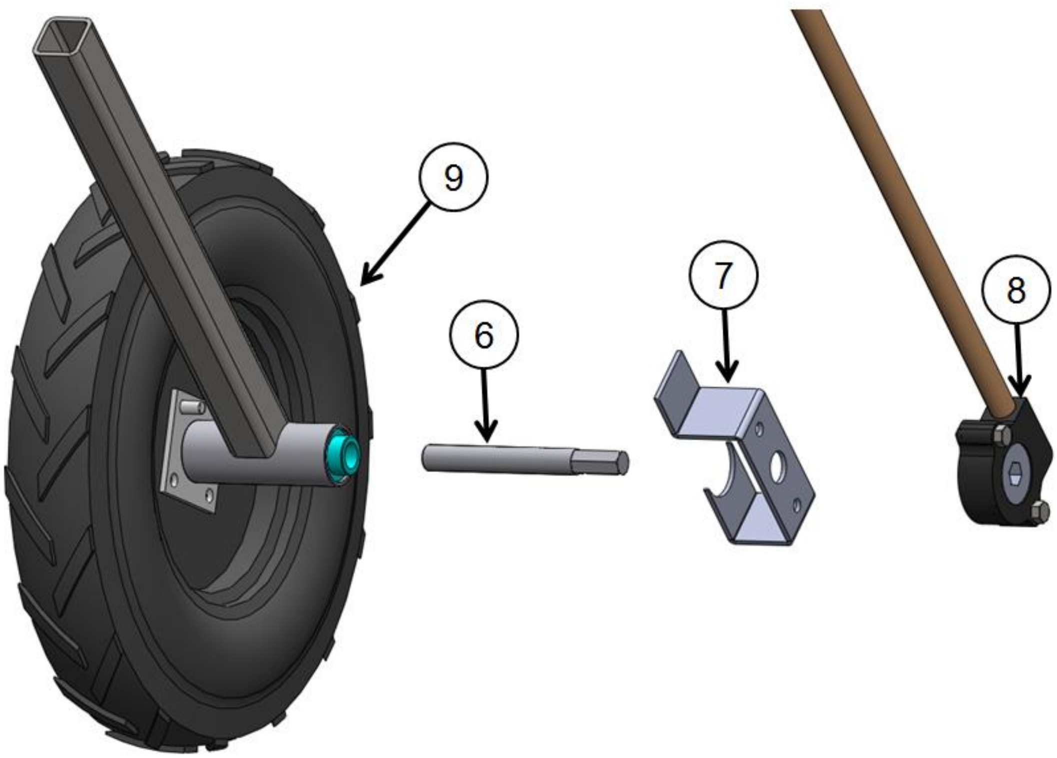


Figure 21. Exploded flexshaft drive design input assembly. Permission pending

TABLE 3. BILL OF MATERIAL, FLEX SHAFT DRIVE.

Part #	Description
1	Output FlexSeeder flex shaft gearbox
2	1 in I.D. Flange Bearing
3	Flex shaft hex output shaft
4	Flex shaft output bracket
5	Valmar variable chain assembly (*X-Sprocket* assembly) [5]
6	Flex shaft hex input shaft
7	Flex shaft input bracket
8	Input FlexSeeder flex shaft gearbox
9	Valmar tire and wheel bearing assembly
10	FlexSeeder flex shaft

Note: Blue shading indicates custom-made component.

Orange shading indicate the FlexSeeder assembly purchased pre-assembled.

A final chain reduction again needs to be used as in the hydraulic drive design in order to allow for ratio flexibility. A unique feature of the Elliot Manufacturing® FlexSeeder System [14] is that the input and output gearboxes can be chosen by the customer to have a certain drive ratio. That is, the first 2:1 reduction, which can be seen in Figure 5, can be matched via gearbox options from Elliot Manufacturing®. Thus the original X-Sprocket system can still be used in this design. The original X-Sprocket equation can also be used as well as the same stock sprockets and chains for the flex shaft design. Figure 20 shows that the FlexSeeder System bolts onto one side of the output bracket and a 1" I.D. flange bearing sits on the other side to support the output shaft. Assembly 5 in Figure 20 represents the X-Sprocket final chain reduction of the current Valmar system shown in Figure 5.

The advantage of using the flex shaft is the smaller number of components as compared to the hydraulic drive system. It is a direct mechanical drive and thus its accuracy is much more predictable than the hydraulic drive system which is affected by ambient temperature. The disadvantage compared to the hydraulic lines is that the FlexSeeder

System can only be supplied stock by the manufacturer at lengths up to 15ft. Hydraulic lines can be virtually limitless in length.

2.3.1 Flex Shaft Considerations

The main component for solving Valmar's problem in this design is the flex shaft itself. Its unique design makes it exceptionally perfect for Valmar's application. One design feature that makes the shaft ideal is its core [15]. A major requirement for the design is that the drive must be able to transmit enough power to rotate the metering gearbox. Elliot manufacturing's flexible shaft has core sizes ranging from ⅛ to 1⅝ inches in diameter that can transmit up to 10HP over a speed range of between 0-50,000 rpm, which makes it adequate as our requirement falls within range. Care must be taken however to ensure that the right direction of rotation is applied. If the shaft is operated in the wrong direction, as much as 30% of its power can be lost. Cores are available in right hand wind for clockwise operation and left hand wind for left hand operation. Bi-directional cores are also available in right hand wind for both directions of operation.

One major concern Valmar had with the flexible drive is the impact varying environmental conditions will have on the shaft, and corrosion as a result of fertilizer deposition. Elliot Flexible shafts are designed with casings [15] specifically to eliminate these issues. Elliot Manufacturing has made it clear their flex shafts can overcome pretty much all weather conditions and environmental obstacles as they are currently mounted on units that sit outdoors all year round [16]. The casing [15] is also designed and manufactured to exhibit a minimum amount of stretch and twist providing a uniform bearing surface for the core. A case ferrule [15] is attached to the casing as well to prevent it from rotating and to support it in bending. Ideally, a large radius of operation should be maintained and sharp bends or kinks avoided.

Other design features include the shaft fitting and motor connection [15]. The shaft fitting mates the shaft with the drive wheel and is available in a wide variety of designs to fit customer needs. The motor connection, which connects the shaft to the drive wheel on one end and the chain assembly on the other, provides the required gear ratio of 2 to 1 between the two sections. This also is available in a variety of designs to suit customer requirements.

One of the most important features of the flex drive is its flexibility. This tackles one of Valmar's major issues of limited space, as it provides Valmar with the opportunity to mount the drive wheel virtually anywhere on the implement and still achieve the same level of metering accuracy.

Another very attractive quality of the Elliot flex drive is that it requires very little maintenance. The core should be re-lubricated after every 200 hours of as a result of the screw action of the core winding which moves the lubricant to one end of the casing. A good petroleum base grease of temperature range between -40° to +250° F should be used. It should be ensured that the casing is never packed with grease, as it will increase torque load on the core [15].

Other features of the flex drive include:

- Ability of the flex shaft to overcome problems with misalignment
- Ability to absorb and isolate transmission needs
- Ability to withstand the shock of sudden changes in load due to starting and stopping
- Ability to overcome adverse environmental and weather conditions
- Ease of mounting
- Low maintenance cost
- It is very safe to operate

3 Material Cost Breakdown

Valmar's existing chain drive for their granular applicators is not as costly as the proposed designs. However, the goal of the project was not necessarily to design a more cost effective solution, but improve the flexibility in the installation of the system. The current drive system costs $248.18 for fabrication and materials. Almost all of the old ground drive system is being kept. TAA’s designs are additions to the current drive system for the added flexibility of the system. The following TABLE 4 and TABLE 5 depict the general cost of components and materials for both designs. A breakeven analysis will be shown in Appendix C.

TABLE 4. HYDRAULIC DRIVE BOM COST.

Hydraulic Drive				
Part	BOM #	QTY	Cost	Sum
Existing Ground Drive [12]	unlisted	1	$248.18	$248.18
Motor/Pump	4, 7	2	$200.00	$400.00
Hydraulic Reservoir	6	1	$200.00	$200.00
Hoses [17]	5, 6, 9, 10, 11, 13	12	$5.30	$63.60
Hydraulic Fittings [18]	unlisted	24	$15.00	$360.00
Motor Brackets	3, 8	2	$75.00	$150.00
Reservoir Bracket	14	1	$75.00	$75.00
Oil Filter [19]	12	1	$35.00	$35.00
Relief Valve [19]	17	1	$75.00	$75.00
Hardware	unlisted	1	$30.00	$30.00
Pump to Wheel Coupler	2	1	$50.00	$50.00
			Total =	$1,686.78
(Optional Addition)				
Temp. Comp. Flow Control	unlisted	1	$650.00	$650.00
			Total	$2336.78

The BOM cost of TABLE 4 includes two extra rows at the bottom. This is the cost of the system if it included the use of a temperature compensated flow controller. This device would automatically change the oil flow in the system depending on the system temperature. Details on the temperature compensated flow controller can be found in section 2.2.2.3.

TABLE 5. FLEXSHAFT DRIVE BOM COST.

Flex Shaft Cost Breakdown				
Part	BOM #	QTY	Cost	Sum
Existing Ground Drive [12]	unlisted	1	$248.18	$248.18
FlexSeeder Shaft [16]	1, 8, 10	1	$275.00	$275.00
1" Bearing	2	1	$13.00	$13.00
Mounting Brackets	4, 7.	2	$75.00	$150.00
Custom Hex Shafts	3, 6	2	$80.00	$160.00
Hardware	unlisted	1	$15.00	$15.00
			Total =	$861.18

4 Conclusion

Team Adamu & Associates designed and discussed two alternative designs to address the issues inherent in the current metering drive system used by Valmar Airflo Inc.

The first design used a hydraulic drive to replace the sprocket and chain drive system. The ground wheel would rotate the input shaft on the hydraulic pump. The pump would then force oil through the hydraulic lines to rotate the motor. A gear mounted to the motor shaft would then turn a gear mounted to the gearbox input shaft via a chain. This design will allow Valmar to place the ground driven wheel virtually anywhere on the implement provided hose lengths are sufficiently long. In the absence of a conclusive heat transfer analysis, the testing procedure outlined in section 2.2.2 should help determine whether the system can maintain 5% accuracy in metering conveyor speed over a wide range of ambient temperatures.

The second design involved replacing part of the current sprocket and chain assembly with a flexible drive shaft. This shaft would allow Valmar to change the location of the ground driven wheel without having to ensure sprocket alignment. The flex shaft is a commercially available off the shelf product build by Elliot Manufacturing.

Both designs are capable of increasing the degree of flexibility of Valmar's applicators, but TAA recommends the flex shaft system because it is considerably more cost effective, is available off-the-shelf and represents a much simpler system in terms of the number of components. Above all, the flexible driveshaft is guaranteed to meet the accuracy requirements imposed on the design, and the same cannot be said of the hydraulic drive.

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Appendix A: In-depth Design Analysis

A-1 Hydraulic Drive Accuracy Concerns

The analytical background presented subsequently was originally composed with the goal of determining whether or not the system could be accurate to within 5% given a wide range of operating temperatures. However, the required analysis would result in an inaccurate estimate at best given the uncertainty in the numerous design variables. From the length of hose used, to the leakage through a motor or pump, to the system operating pressure used, there are too many variables for which estimates would have to be used which would compound and create an inaccurate model of the system's behavior. Thus, the theory will still be presented and should serve as background information, but focus will be placed on development of a test procedure that will determine whether a hydraulic drive system could work (and remain accurate). The background and theory will be subsequently split between the fluid mechanics and heat transfer aspects of the design.

A-1.1 Fluid Mechanics – Important Principles

A-1.1.1 Liquid Compressibility

In fluid mechanics, the fact that liquids are often treated as being incompressible is a misnomer. In liquids, pressure is transmitted in the same way as in gases, and that is by physical compression of the molecules. Furthermore, all liquids contains *some* amount of dissolved gases, namely air, at any given time. Gases are highly compressible, and thus, this amount of dissolved air is also subject to compression when the liquid is compressed. Normal and acceptable amounts of dissolved air in mineral oil can account for upwards of 10% of the oil's total volume [10]. One need only look at a density vs. temperature plot for any given fluid to see that fluid density does not remain constant with changing temperature. Figure 22, below, demonstrates this trend.

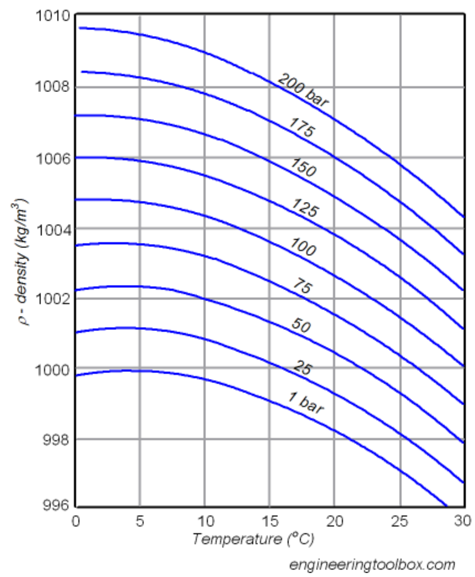

Figure 22. Water density vs. temp & pressure [20]. Permission pending.

However, comparison of the relative changes in density for a gas (for example, air) versus a fluid reveals that the density change with temperature, expressed as a percentage, is *far* greater in gases than in liquids. TABLE 6 summarizes the relative difference in density at atmospheric pressure for atmospheric air vs. water.

TABLE 6 DENSITY VS. TEMPERATURE, AIR & OIL.

Fluid	T_1[°C]	$\rho_1[kg/m^3]$	T_2[°C]	$\rho_2[kg/m^3]$	$\Delta\rho$
Air	0	1.276	57	1.056	16.1 %
Engine Oil		899.1		865.5	3.7 %

Data taken from *Fundamentals of Heat and Mass Transfer, 7th Ed* [21]

As was to be expected, the results from TABLE 6 show that the magnitude of the density change is far smaller in engine oil, thus the justification for at times considering this liquid (like all others) as incompressible. This is done to simplify calculations in fluid mechanics, but in some applications where a precise flow rate is required, liquid density cannot be considered constant.

A-1.1.2 Volumetric Flow Rate vs. Mass Flow Rate

The negative implications of treating liquids as compressible are observable here. Using the principle of continuity we can infer that the mass flow rate in any pipe flow (control volume) situation must remain constant. Volumetric flow rate, on the other hand, can vary throughout a control volume when density is known to fluctuate - this can be applied to both gases and liquids. Figure 23 depicts a representative pipe flow situation in which heat is added to a control volume that features a flow of motor oil.

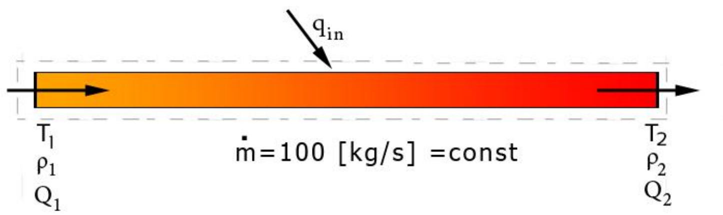

Figure 23. Example control volume analysis for pipe flow.

TABLE 7. ENGINE OIL VOLUMETRIC FLOW RATE VS. TEMPERATURE

	State 1	State 2
$\boldsymbol{T}$[°C]	0	57
$\boldsymbol{\rho}\ [\boldsymbol{kg/m^3}]$	899.1	865.8
$\boldsymbol{Q}\ [\boldsymbol{m^3/s}]$	0.1112	0.1155

Data taken from *Fundamentals of Heat and Mass Transfer*[21]

The volume flow rate, Q, is related to the mass flow rate, $\dot{m}$, as follows:

$$Q = \frac{\dot{m}}{\rho}\ [m^3/s] \qquad (1)$$

Thus, with varying values of fluid density, ρ, at state 1 and 2, it is obvious that the volumetric flow rate must vary between state 1 and state 2. TABLE 7 summarizes these exemplary results. The increase in flow rate for a 57 degree temperature increase can be found as follows:

$$\Delta Q = \left(\frac{Q_2 - Q_1}{Q_1}\right) \cdot 100 = \left(\frac{0.1155 - 0.1112}{0.1112}\right) \cdot 100 = 3.9\%$$

From the above, it can be seen that while mass flow rate must remain constant in any control volume model, it is possible for the volumetric flow rate to increase within a pipe flow due to the change in density that accompanies heat addition.

A-1.2 Heat Transfer Analysis

The modeling of the heat transfer involved with the hydraulic drive system must be separated between the transient stage (i.e. before the fluid has reached operating temperature) and the steady-state stage (fluid at operating temperature). Both methods are strongly influenced by the ambient air temperature, and thus a worst-case ambient temperature must be employed for each.

A-1.2.1 Heat Generation

In any hydraulic system, heat is generated as a direct result of pressure losses and mechanical inefficiencies. Every flow-through component of the system generates some heat, including any valves, plumbing, motors, pumps, etc. Pressure losses can be minimized through good design practices, including proper sizing and selection of components. There are limits on the maximum operating temperature of any given hydraulic fluid as determined by the manufacturer; these are based mainly on the reduction in viscosity with increased temperature (promoting wear of mechanical parts, as well as leakage), as well as the break-down of beneficial additives. Thus, it is important that a hydraulic fluid is operated at or below its maximum rated temperature. The rate at which heat is generated within a hydraulic system, q_g, can be expressed as follows:

$$q_g = \rho Q g \Delta H \; [W] \tag{2}$$

where ρ is the fluid density expressed in units of $\left[\frac{kg}{m^3}\right]$, Q is the volumetric flow rate in units of $\left[\frac{m^3}{s}\right]$, g is the gravitational constant, and ΔH is the total head loss incurred by the fluid in the system, expressed in units of metres of fluid. This head loss can also be expressed in terms of a pressure loss as:

$$\Delta H = \frac{\Delta P}{\rho g} \tag{3}$$

where ΔP is measured in units of Pascals. Substituting equation *(3)* into equation *(2)* yields a simpler method of calculating the rate of heat generation:

$$q_g = \rho Q g \left[\frac{\Delta P}{\rho g}\right] = Q\Delta P \; [W] \tag{4}$$

The example from the fluid mechanics background demonstrated that heat addition, leading to a fluid density change, can affect volumetric flow rate through a control volume. However, the magnitude of the heat transfer rate, q_{in}, was not quantified in efforts to simplify the example. Determination of the incoming heat transfer rate, q_{in}, is possible using the following equation:

$$q_{in} = \dot{m}c_p(T_2 - T_1)[kW]$$

where c_p is the specific heat of the fluid (treated here as a constant, ie. no temperature dependence). Substituting the known values for the previous example with the flow of motor oil, we have:

$$q_{in} = 100(1.900)(57 - 0) = 190[kW]$$

Note that this approach is very limited in its application; it deals with a very simplistic case and does not deal with the transient behavior of the system.

A-1.2.2 Heat Dissipation and Absorption

Finding the rate of heat generation provides a starting point to estimate the fluid temperature change over a finite time interval. However, in order to accurately estimate this temperature change, one must also factor in the system's interaction with the external environment in the forms of convection, radiation, and conduction; this becomes a far more complicated task. For a system with no heat exchanger, the majority of the heat dissipation can typically be assumed to occur from the reservoir. However, when a worst-case (high) solar heat flux is factored in, the metal reservoir can actually act as a heater. Similarly, the heat absorbed through a length of black rubber hose in direct sunlight cannot be ignored given that it has significant surface area. Accurate estimation of the convection heat transfer taking place becomes critical, but as a worst-case scenario, a no-wind condition (free convection) could be employed. In this case, conduction from the pump, motor, and reservoir through the other metal parts in contact (hopper, pump/motor mounting brackets) becomes a significant mode of heat transfer, but is extremely hard to model.

A-1.2.3 Transient Behavior

The transient behavior of the system refers to time period where the system temperature has not yet reached its operating value. It is assumed that on start-up of the system, initial fluid temperature, T_i, will be equal to that of the ambient air temperature, T_∞ . As the system operating temperature is expected to be higher than ambient, any convective heat transfer will be from the system to the surroundings, while all components will continue to experience solar absorption gains. This model is complicated by the fact that ambient temperature does not necessarily remain constant throughout the operation; outside air temperature fluctuates throughout the day, being low in the morning, reaching a peak in late afternoon, and again falling in the evening. A quick glimpse at a temperature vs. viscosity curve for automatic transmission fluid from

Figure 14 shows that viscosity will vary *greatly* in the transient stage; ie, oil temperature ranging from 0°C to 50°C. As was previously shown in the fluid mechanics section, this variation in viscosity is unacceptable and will result in significantly less flow from pump output to motor output. Thus, the hydraulic drive system must have a means of pre-heating the oil prior to use (eg. a pre-heat valve). That is, the hydraulic motor should be disengaged (or the hopper closed off) until the system has reached operating temperature.

A-1.2.4 Steady-State Behavior

Once the oil has reached a steady-state operating temperature, this temperature is assumed to remain constant throughout the system. In reality, small local temperature changes will be observed due to local heat addition, but these changes will be assumed to be negligible. The operating temperature will, however, be a function of ambient temperature, and it is thus desirable to approximate this relationship over the range of ambient temperatures that can be expected in use. A significant difference in operating temperature could potentially prevent the system from being accurate to within 5% as required by Valmar, given that viscosity varies with temperature, and motor leakage varies with viscosity (as discussed in the hydraulics section). To further complicate the problem, the solar irradiation, G, that will contribute to a system temperature rise varies with latitude; thus, an average value for the southern United States [22] (in summer) could be used to be representative of a worst-case scenario. Figure 24, below, schematically depicts the heat transfer taking place, neglecting that to and from small components such as the relief valve and filter. Here, it is easily seen that accounting for all modes of heat transfer for each component becomes a daunting task.

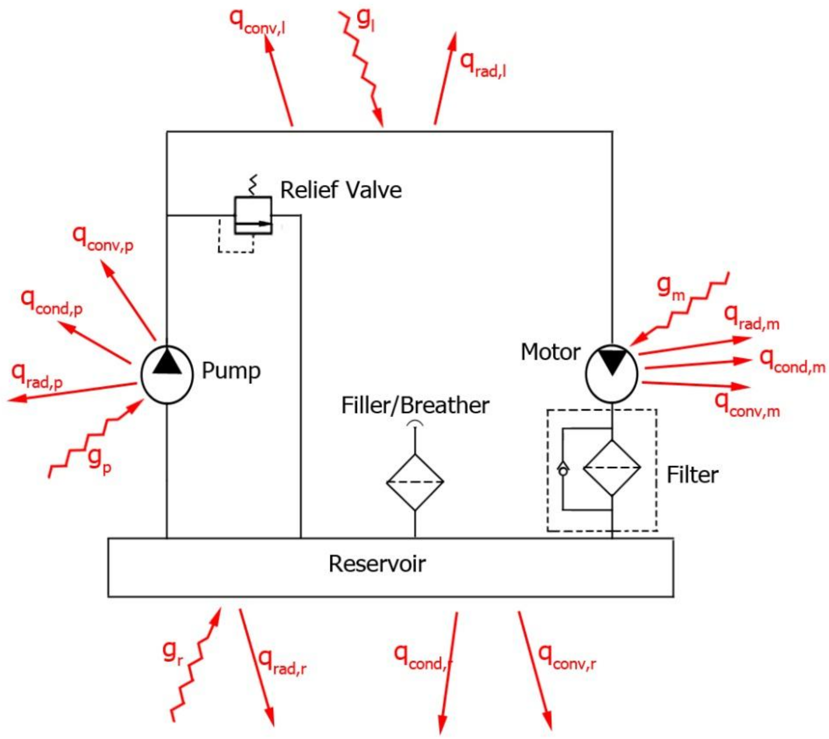

Figure 24. Heat transfer schematic.

This background should serve as concrete support for the importance of the accuracy test described in the body of the report.

Appendix B: Assembly & Manufacturing Principles

Manufacturing principles were carefully considered in the design of all parts for this project. The type of machining and assembly process for every part was selected based on accuracy and cost. Materials selected were also considered based on their strengths and other mechanical properties such as machinability, that best fit the design operational requirements.

Input and output as seen in Figures 28, 29, 30, 31 for both the hydraulic drive and the flex drive will be fabricated from a thick gauge metal sheet. As these parts have low tolerances and require a high level of accuracy, all holes on the brackets will be laser cut and bent corners made with a CNC machine. All input brackets will be welded to the wheel drive shaft at the hub while the output brackets will be bolted to the hopper. This is because more stability is required at the input as the shaft-to-shaft connection has a very low tolerance for misalignment as opposed to the shaft-to-gear connection at the output.

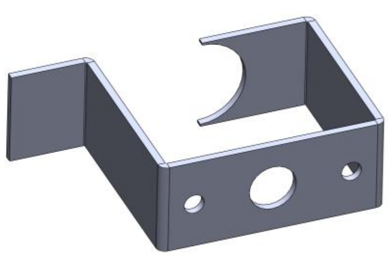

Figure 25. Hydraulic pump input bracket

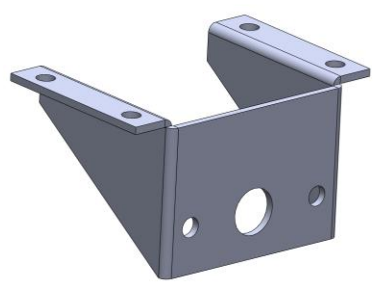

Figure 26. Hydraulic motor output bracket.

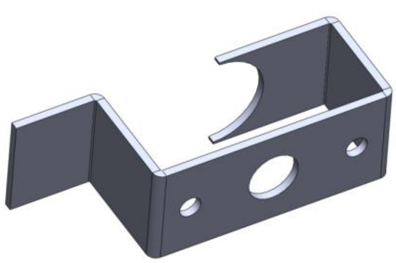

Figure 27. Flex shaft input mounting bracket.

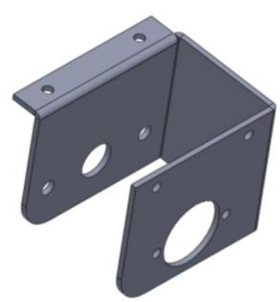

Figure 28. Flex shaft output mounting bracket.

The oil reservoir bracket (Figure 29) will be cut from a 1” angle iron stock. As accuracy is not of great importance for this part, CNC and laser cutting will not be necessary. The bracket will be welded to the side of the hopper and holes drilled on top of it to bolt the reservoir.

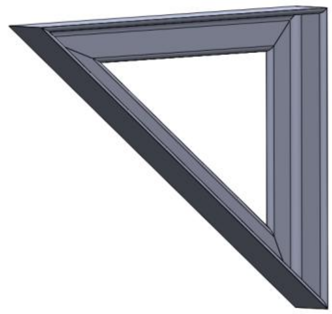

Figure 29. Oil reservoir bracket.

The input and output shafts (Figure 30) for the flex drive will be fabricated from 1” iron round bar stock. The bar stock will be machined to achieve the hexagonal pattern via a milling process. The shaft will be connected to the hub of the wheel through a bearing assembly and locked in place by a key to provide a tight rigid connection.

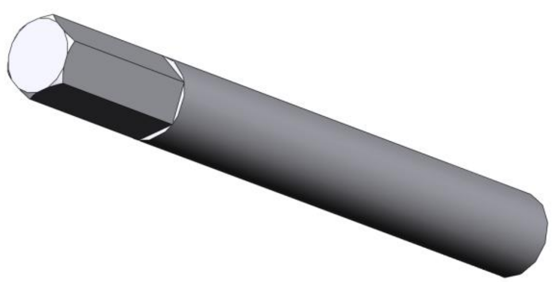

Figure 30. Flex shaft input and output shaft.

All standards applicable to this design were researched in the consideration of assembly and manufacturing processes. It is recommended that these standards be purchased and adhered to in the final design stages; however, budget constraints prevent TAA from purchasing copies of these standards. This responsibility will thus be left with the client. The International Standards Organization (ISO) [23] and the American Society of Agricultural & Biological Engineers (ASABE) [24] were the primary organizations through which standards searches were conducted. TABLE 8 below, summarizes the relevant standards that were found.

TABLE 8. APPLICABLE STANDARDS.

Organization	Standard # and Year	Description
ISO	4254-8:2009	Agricultural machinery -- Safety -- Part 8: Solid fertilizer distributors
ISO	3767-2:2008	Symbols for operator controls and other displays -- Part 2: Symbols for agricultural tractors and machinery
ISO	28924:2007	Agricultural machinery - Guards for moving parts of power transmission - Guard opening without tool

Appendix C: Breakeven Analysis

The breakeven analysis was conducted with respect to the break even cost, and not the breakeven point as is conventionally done. Valmar Airflow Inc. provided some economic data in order to proceed with this analysis. As a conservative approach, Valmar's labor, fixed overhead and variable overhead were scaled up by the percent difference between the existing chain drive cost and the two proposed designs, see TABLE 9.

TABLE 9. VALMAR AIRFLOW INC. ECONOMIC DATA AND SCALED DESIGN DATA.

	Chain Drive	Hydraulic	Flexshaft
Cost	\$248.18	\$1,686.78	\$861.18
Percent Difference	----------	679.66%	347.00%
Labour	\$16.50	\$112.14	\$57.25
Fixed Overhead	\$27.00	\$183.51	\$93.69
Variable Overhead	\$18.00	\$122.34	\$62.46
Units /year	100 to 150		

The breakeven point is typically calculated using the following equation:

$$X = \frac{FC}{P - VC}$$

where: X= Units Sold

P = Price per Unit

VC = Variable Cost

FC = Fixed Cost

Since the amount of units Valmar Airflow Inc. sells per year is already known, the equation will be rearranged in order to compare the price per unit:

$$P = \frac{FC}{X} + VC$$

In this arrangement the breakeven price per unit is calculated. The breakeven price of the proposed designs and the existing system can now be compared. *VC* included the material cost, the labor, and variable overhead. *FC* was set as the fixed overhead. P was calculated at 100, 110, 120, 130, 140, and 150 units to evaluate how much the breakeven cost would fluctuate during the year depending on how many units were sold.

TABLE 10. COMPUTED BREAKEVEN PRICE ITERATED FROM 100 TO 150 UNITS.

	Break Even Price		
Units Sold	Chain Drive	Hydraulic	Flexshaft
100	$291.86	$1,983.66	$1,012.75
110	$291.84	$1,983.54	$1,012.69
120	$291.83	$1,983.45	$1,012.64
130	$291.82	$1,983.37	$1,012.60
140	$291.81	$1,983.31	$1,012.57
150	$291.80	$1,983.25	$1,012.54

It can be noted that the fixed overhead barely affects the breakeven cost in all drive systems. In other words the price of the drive system will not fluctuate considerably in the range of units sold. This correlates to a stable company economic situation. If the client decides to pursue either design they can now use this information as a baseline and choose what profits they would like to see for their decision.

Appendix D: Initial Design Concepts

The following concepts were conceived at the Design Concept stage. The bulk of the selection process is included here to show how the flexible driveshaft and hydraulic drive systems were chosen as the final designs for consideration.

D-1 Hydraulic

The allure of a hydraulic metering drive system stems from its flexibility and simplicity. The ability to transmit fluid power through flexible hoses makes the system highly adaptable in implement-mount applications. Several variations of a hydraulic drive system have been considered, and will be discussed subsequently.

D-1.1 Metering Wheel-driven Pump with Hydraulic Motor

This scenario represents the simplest configuration of hydraulic drive system, and contains only the bare necessities of a hydraulic system – a small reservoir to store fluid, a pump to be powered by the rotation of the metering drive wheel, a hydraulic motor that will turn the gearbox input shaft, and hoses connecting all components. Gearing could be achieved, at least partially, by selection of a pump and motor of different displacements (i.e. one rotation of the pump input shaft does not equal one rotation of the motor's output shaft). Figure 2, below, depicts a sketch of such a system.

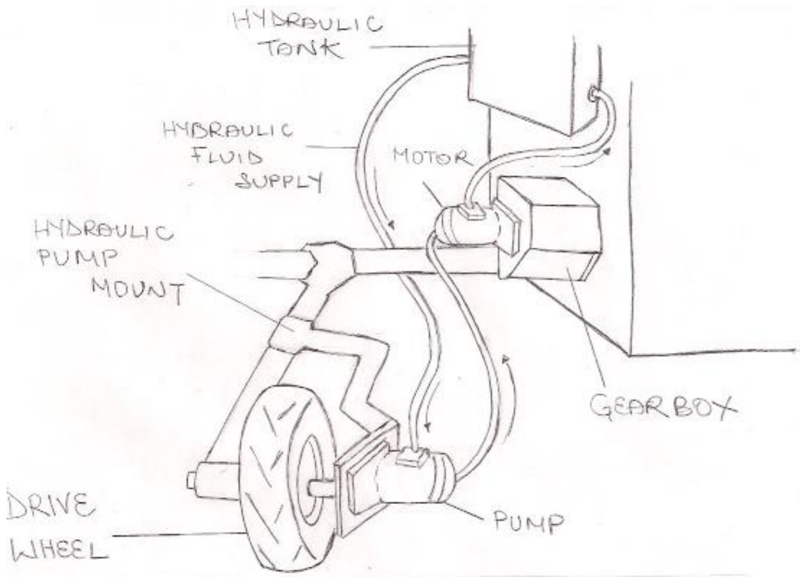

Figure 31. Metering wheel-driven pump with hydraulic motor design diagram.

In researching metering drive systems, it was found that a similar design is being used by French agricultural manufacturing company Gregoire Besson (GB), albeit for a slightly different application. Coincidentally, GB is using Valmar applicators mounted on their own implements. Correspondence with a representative from GB [25] provided positive affirmation that such a system could work well for Valmar's applications. No technical specifications on GB's drive system were obtained as this system is still in the early stages of development, and components and system parameters are not finalized.

D-1.2 Tractor-supplied Oil with Hydraulic Motor

This scenario involves utilization of the tractor's oil supply and pump, relying on an auxiliary hydraulic output line from the tractor to supply oil to the hydraulic motor which turns the gearbox input shaft. The advantage over design 1.1 is the lack of the need for a reservoir and pump, which further simplifies the installation of such a system and reduces its cost. The tradeoff here is that flow control, and thus metering speed control, becomes difficult since the flow rate of the tractor's pump depends on the tractor's engine speed, and not the ground speed. For example, an increase in grade (i.e. a hill) will result in a higher engine speed and thus higher flow rates to the hydraulic motor, resulting in denser product application than desired – this is unacceptable, and must be accounted for in design. Another potential problem is the availability of an auxiliary hydraulic output from the tractor; most modern tractors have several outputs, but depending on the tractor model (and vintage) as well as the application, it is possible that no additional hydraulic outputs will be available to power the hydraulic motor.

D-1.3 Electric Motor-driven Pump with Hydraulic Motor

This design attempts to address the space constraint issues inherent with use of the metering drive wheel by eliminating it altogether and powering the hydraulic pump via an electric motor. Such a system would benefit the implement-mount applications as the electric motor and pump could be located anywhere on the implement. While this solves one problem, it creates another – flow control becomes difficult when using an electric motor that is spinning the pump at a constant speed irrespective of tractor ground speed. It thus becomes more difficult to achieve proportionality between ground speed and metering rate. This option would also require electrical power from the tractor.

D-1.4 Gasoline Engine-driven Pump with Hydraulic Motor

Similar to design 1.3, this design eliminates the need to use the metering drive wheel. Pump speed could be pre-set using the engine's throttle, and once pre-set would remain constant. Thus, this system again would require a semi-elaborate method of flow control to achieve the desired proportionality between ground speed and metering rate. The gas engine is an improvement over the electric-motor design in the sense that it does not require electrical power from the tractor to operate; however, mounting a gas engine on an implement could be difficult due to space limitations as a gas engine is fairly bulky.

D-1.5 Servo Flow Control

All previously described hydraulic designs enlisted in a hydraulic motor to turn the applicator's gearbox input shaft. Here, the rotational speed of the gearbox input shaft remains constant, and gearing allows a variation in speed of the metering conveyor (and thus a variation in product application rate). The same function can be accomplished without a gearbox via the use of a servo-operated flow control valve in the hydraulic system. In this scenario, the hydraulic motor could be directly coupled to the conveyor drive wheel, eliminating the gearbox altogether. Such a design could employ closed-loop conveyor speed feedback to control the servo valve. The system would require a means of sensing tractor ground speed (i.e. a tachometer) and a signal converter to convert the input signal to a pulse width modulation (PWM) signal. A controller would compare conveyor speed with tractor ground speed and make adjustments to the servo valve position as necessary to ensure that flow rate to the motor (and thus metering rate) remains constant. While such a design promises to be extremely accurate, any closed-loop control system also promises to be more expensive and complicated.

Research reveals that similar systems are already available on the market today, some of which integrate with the tractor's on-board GPS system to provide speed feedback and automatic flow adjustment given the tractor's spatial coordinates on the field [26]. These systems are extremely accurate, and their high-end performance is accompanied by a correspondingly high price tag.

D-2 Flexible Drive Shaft

One of the major problems with the current system is in making the connection from the ground driven wheel to the gearbox. This is currently done with a system of gears, shafts and chains. This connection could be made easier if a flexible shaft was used to connect the power source to the gearbox. Designs involving a flex shaft are described below.

D-2.1 Direct from Metering Wheel to Gearbox

This design would work by replacing all the gears, shafts and chains with one flexible shaft that could be coupled directly to the drive wheel and the gearbox. This option is desirable because it will maintain the highly accurate sync between ground speed and application rate. It would also be very easy to install, and would not require any major redesigns of the current drive wheel configuration. Care will have to be taken to ensure that the flexible shaft does not interact with any nearby objects. This could be protected against by enclosing the flexible shaft in a protective sleeve. Figure 3, below, features a sketch of this design.

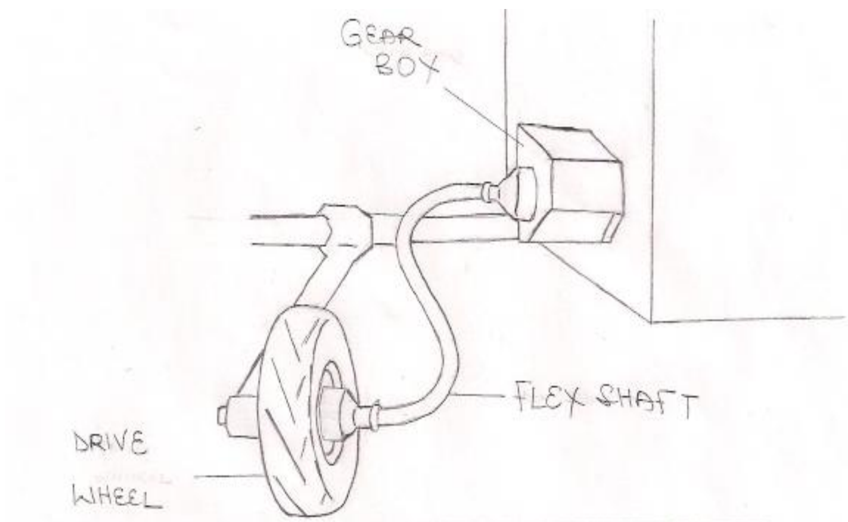

Figure 32. Flexible drive shaft design diagram. Image credit: Najib Adamu

D-2.2 Electric Motor Rotates Flex Shaft

The wheel drive system could be removed from the system by installing an electric motor and making a connection between the electric motor and the gearbox with a flexible shaft. The current system that uses a ground driven wheel is bulky and can cause problems when mounting on an implement. By replacing the drive wheel with an electric motor, the difficulty in mounting an applicator is greatly reduced. The use of a flexible shaft will also allow for the electric motor to be mounted in different places without the need to redesign the system. Due to the corrosive nature of some of the fertilizers used in Valmar's applicators, the electric motor may have a short lifespan before it needs replacing. The motor would also have to run off of the tractors electronic system.

D-3 Pneumatic

Like a hydraulic metering drive system, a pneumatic system uses fluid power, in the form of pressurized air, to drive the metering conveyor. Pneumatic systems have the advantage of being environmentally friendly compared to hydraulic systems given that atmospheric air is the working fluid. Several designs were considered and will be discussed individually.

D-3.1 Drive wheel-driven Compressor, Tank, and Pneumatic Motor

This system would be the pneumatic equivalent of the hydraulic system as outlined in design 1.1. However, the difference here would be that the air compressor (pump) would pressurize the tank, and the tank would supply the motor with pressurized air. This is a requirement since it is unlikely that any drive wheel-driven compressor would be able to keep up with the pressure and flow requirements of the motor given the drive wheel's low rate of rotational speed. This, in itself, is an inherent problem, unless the tank was pre-charged prior to use such that the compressor was able to maintain a charge pressure that was sufficient to drive the motor.

D-3.2 Electric Motor-driven Compressor, Tank, and Pneumatic Motor

This method would address the volumetric flow shortcomings of method 3.1 by assuring that the compressor could be rotated fast enough to maintain a given tank charge pressure. Here, electrical power would be required from the tractor in order to run the electric motor. Figure 33, below, depicts such a system.

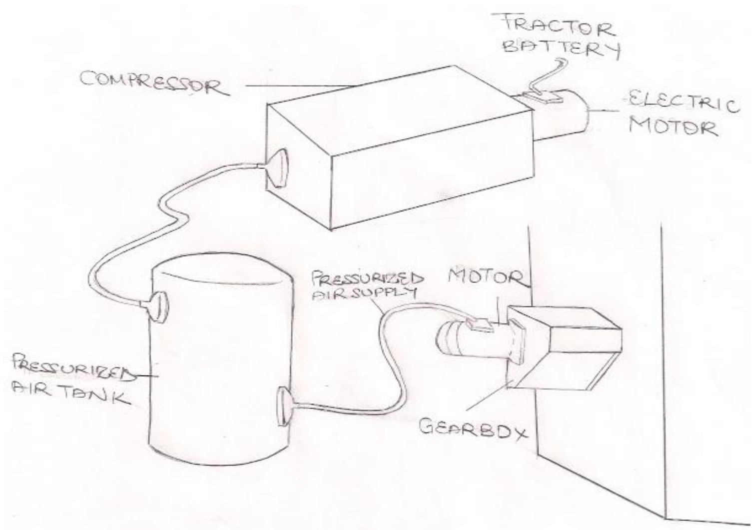

Figure 33. Electric motor-driven compressor, tank, and pneumatic motor diagram. Image credit: Najib Adamu

D-3.3 Gasoline Engine-driven Compressor, Tank, and Pneumatic Motor

Like design 1.4, this design utilizes a gasoline engine to power the compressor, assuring the compressor is spun fast enough to keep up with the motor's demand for flow. The lack of an electrical power requirement to rotate the compressor is an improvement over design 3.2, however the space limitations on some implements could translate into a problem in finding a suitable mounting location for the relatively bulky gasoline engine.

D-4 Variable Speed (VS) Electric Drive

The current trend in the agricultural world is to use variable speed application rates. This translates into direct savings to the farmer because they can use fewer products to achieve a maximum yield. This section will look at using an electric motor to rotate the gearbox directly.

D-4.1 Tractor Powered VS Electric Drive (GPS Control)

A variable speed electric motor powered by the tractor's electrical system can be attached to the gearbox. The speed of the electric motor will be controlled by a link to the tractors GPS system that will provide ground speed information. This will be capable of providing detailed information regarding the amount of product applied in various areas around a field. Such a system will dramatically increase component accessibility and decrease bulkiness. It will also be safe, as all moving parts will be enclosed inside the electric motor. This system will not work for all farmers if they do not have GPS. It will also be an expensive upgrade for a marginal increase in accuracy. The electronic components of the motor may also be damaged due to the corrosive nature of some fertilizers. Finally, this design will involve a complex control system that will be difficult to design. Figure 5, below, shows a sketch of such a system.

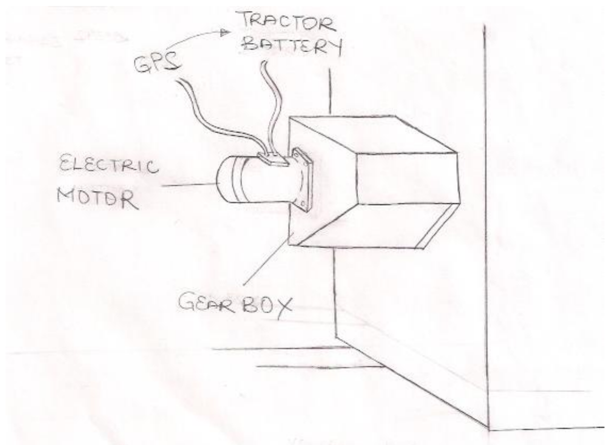

Figure 34. Tractor powered VS electric drive (GPS control) design diagram. Image credit: Najib Adamu

D-4.2 Tractor Powered VS Electric Drive (Tachometer Control)

This design is similar to the design discussed in 4.1. The major difference is that the application speed will be controlled by a tachometer relaying the ground speed to the electric motor through a control system instead of a connection to the tractors GPS system. This system would be very accurate, but would not provide information to farmer regarding application rates used in specific areas of the field.

D-4.3 Battery Powered VS Electric Drive

This is similar to design 4.2, but instead of using the tractor battery to power the electric motor, a battery cell could be mounted on the implement. The electric battery cell could be placed anywhere on the implement and power cables could be easily run to the driving motor. The problem with the battery cell is that they are limited in their charge capacity not to mention the heavy weight of the cell. The speed of the electric motor would be controlled by tachometer.

D-4.4 Wheel-driven Generator Powers VS Electric Drive

This system would have the same benefits as 4.3 but instead of a heavy battery cell, a generator could be attached to a ground driven wheel and directly power a motor at the metering gearbox. This would be similar to using hydraulics but the working medium is electricity.

D-5 Redesign Current System

One major issue with the current system is the limited space available for placing the metering drive wheel. In some situations, running the drive wheel off the ground becomes impossible and has to be mounted on the drive wheel of the implement. This leads to many complications such as reversing the direction of the metering system. TAA came up with several concepts that involving changes to the current system.

D-5.1 Forward-facing Metering Wheel

In the current system, the drive wheel is placed in a backward position. This can sometimes place the drive wheel too close to the wheel of the implement. Relocating the system drive wheel to face forward instead eliminates the issue of limited space therefore making the system more versatile and universal. The good thing about this concept is that it works exactly like the current system and therefore does not require any new parts and no added cost.

Valmar has tried this came design in the past. They chose to throw this design out when the square bar supporting the drive wheel began to bend and break do to collisions with rocks in the field. This problem is not encountered in the backward position because the wheel is pulled over obstruction instead of being pushed into them.

D-5.2 Forward Facing Metering Wheel with Shock

This system will have the same setup as that in 5.1. The only variation to the design is the addition of a shock absorber on the drive wheel shaft to prevent bending or fracture upon impact. This addition of a shock will result in increased cost, but this setup will allow greater installation flexibility. Figure 6, below, depicts such a system.

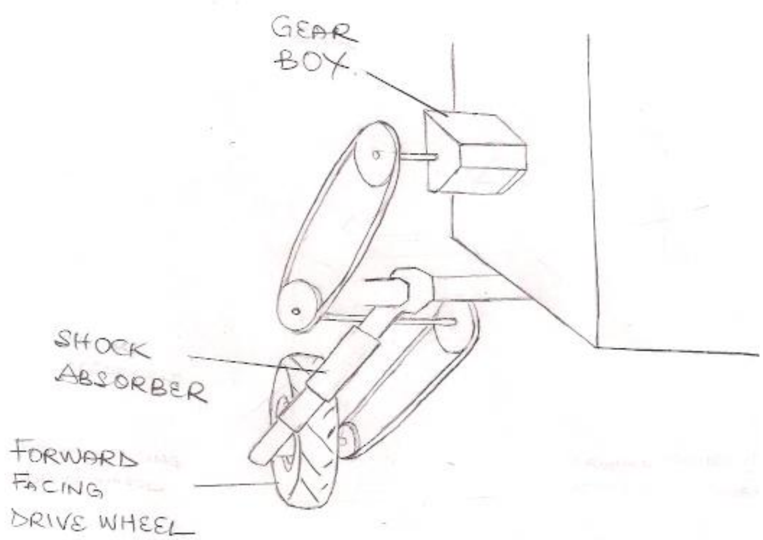

Figure 35. Forward facing metering wheel with shock design diagram. Image credit: Najib Adamu

D-5.3 Heavy Steel Wheel

This design is also similar to design 5.1. The rubber drive wheel will be replaced with a heavy steel wheel. The weight of the steel wheel eliminates the issue of bending and fractures, as the wheel is heavy and large enough to roll over most obstruction in the field. The weight of the wheel will allow it to stay firmly on the ground without the use of springs, as in the current system. While the heavy weight of the steel drive wheel appears to solve most of the issues, it has its disadvantages. Assembling this system can be a problem, as it will require a lot of manpower. Lifting the wheel for storage mode will also be difficult due to the weight of the wheel.

D-5.4 Height-Adjustable (Telescopic) Wheel

This design also addresses the issue of limited space in the current system. The drive wheel is placed vertically below the gearbox. The shaft of the wheel is allowed to telescope, making it possible to adjust the height. The wheel will be locked in place at the desired height by pins on both sides of the shaft. All connections between the drive wheel and the gearbox remain the same as the current system with minor adjustments as required. This design however might not be very productive as the probability of bending and fracture is quite high.

D-6 Multi-directional Drive Belts

Regular V-belts and timing belts are limited to planar motion. A multidirectional belt drive is a system that can transmit power via a belt in a 3-D space[27]. Many dentists use this type of drive system with their hand drills extended on a boom in a 3-D space. Applying this idea to Valmar's metering drive system would allow TAA to move the metering drive wheel to a different position and run a multi-directional drive belt through a series of pulleys to the metering gearbox. Unfortunately, these types of systems are generally only used in very low torque applications. The other downfall of this system is that Valmar would need a different length drive belt for every application. This is not very practical compared to using chains that can be quickly tailored to a custom length.

D-7 Reciprocating Wheel

The reciprocating wheel design idea consists of series of cranks and a push-pull cable to rotate the metering gearbox. A crank connected to the drive ground wheel would cyclically push and pull on a rigid cable. The cable would in turn push and pull on a crank, turning the metering gearbox. This motion would resemble the action of old steam locomotives. The light weight flexible push pull cable can be routed anywhere on the implement to drive the metering gear box. The problem with this system is it is not very practical. The function of the driven cranks could technically flip and run the gearbox in reverse as it moves forward. Also given the environment, the inside of the cable could get jammed with dirt and cease functioning. Push-Pull cables cannot take much load and can also easily break under non uniaxial pressure

D-8 Concept Analysis & Selection

Concept selection was performed via a fairly rigorous process involving screening and scoring matrices. Each stage will be discussed subsequently.

D-8.1 Screening Matrix

With all of the design concepts in place following the internal and external search processes, the designs were first subjected to a screening process with the intention of narrowing down the number of designs that would be considered. This process involved the use of a screening matrix which contained all of the designs as well as the design needs, which acted as the criteria on which each design was evaluated (refer to Appendix A). Given that the evaluation criteria was based on design needs, designs were ranked according to how well each design could meet a given design criteria relative to the other designs. The ranking system employed involved assigning a positive, negative, or neutral value (+, -, or 0 respectively) to all design criteria for each design. The resulting numerical rank for each design was determined by subtracting the number of negatives from the number of positives in a given column. This method successfully weeded out the designs that were not feasible for one reason or another.

D-8.2 Scoring Matrix

With screening complete, the five highest ranked designs were carried forth to the next stage of the design selection process – scoring. A scoring matrix was created in similar fashion to the screening matrix, featuring the remaining concepts and the design needs, again to be used as evaluation criteria (refer to Appendix B). As a reflection of the relative importance of each design criteria, a weight was assigned to each. Here, a finer ranking scale was used to rank each design on each evaluation criteria – a number from 1 to 5 was assigned, with 5 being the best, and 1 being the worst. The product of the need's weight multiplied by its assigned numerical rank was performed for each need, and summed. The designs were then ranked according to their total sums. This process was carried out by each team member individually in attempts to remove all bias from the end result. After each member ranked and summed their choices, the result of each member was then normalized to 25 points. The normalized results were summed again and ranked to find the most popular option.TABLE 11 summarizes each member's normalized picks and ranks the most popular option as 1 and the least popular option as 5.

TABLE 11. SCORING MATRIX SUMMARY

Designs -> / Team Member	1	2	3	4	5
1	25	22.10	18.94	21.84	18.43
2	25	22.25	21.34	19.50	17.54
3	25	23.42	16.63	22.33	14.56
4	25	23.95	19.61	25.00	18.68
Total Sums ->	100	91.72	76.51	88.68	69.22
Final Rank ->	1	2	4	3	5

Scoring Matrix Legend

1. Flex Shaft
2. Hydraulic Pump and Motor
3. Variable Speed Electric Drive (with gps)
4. Forward Facing Wheel with shock absorber
5. Pneumatic compressor electric motor drive

D-8.3 Technical Analysis

The most important design criteria will briefly be discussed here as applied to each design as a means of justifying the ranking found in the scoring matrixes.

D-8.3.1 Metering speed needs be in sync with the speed of the agricultural equipment

The flex shaft ranked the highest in this category due to the direct connection present between the drive wheel and gearbox. The hydraulic and pneumatic options both run the risk of experiencing a variation in flow rate between the supply (pump or tank) and the motor given the change in density of a fluid that accompanies a temperature change. The variable speed electric drive motor's speed would remain proportional to ground speed assuming the controller was accurate. The forward facing wheel should remain very accurate as it is just a modification of the current system, which is known to be very accurate.

D-8.3.2 Robust

The potential for breakage/damage to the flex shaft is relatively high given that it could be routed in areas that are potentially prone to catching things. Any hydraulic or pneumatic lines would also be subject to being caught and damaged while in motion, however there is more flexibility in the routing of these hoses (i.e. securing them along the frame out of the way). The forward-facing wheel with a shock is capable of impact absorption and thus is quite robust. The VSD is a rigid unit assumed to be mounted up out of the way of any potential obstructions that may be encountered in motion and thus ranked highly.

D-8.3.3 Mounted Quickly and with Minimal Parts

The flex shaft is the simplest and easiest to install, as it is a single component. The electrical wire routing of the VSD, along with the mounting and coupling of the motor with the gearbox input shaft, resulted in a lower scoring. The hydraulic system requires the pump, tank, and motor be mounted, as well as the hoses run and connected. Similarly, the hydraulic system requires hose routing, mounting the considerably larger air tank, an electric motor coupled to the compressor, as well as the running of wiring. The forward facing wheel design is subject to essentially the same installation time as the current system, which was a flaw of the current system; the drive wheel must be hard-mounted to the implement's frame, chains must be custom-made to the proper length, and subsequently installed around gears.

D-8.3.4 Accessible

The flex shaft should be adequately accessible in any situation. The hydraulic system could potentially see that drive wheel, and thus the pump, is positioned in a tight space. The pneumatic system's compressor and electric motor could be placed to allow accessibility (space-permitting). The forward-facing metering wheel promises to be an improvement over the current design in terms of accessibility, but some components are still likely to be positioned somewhat precariously. The VSD should be adequately accessible given that it will be directly mounted to the gearbox input shaft, located on the side of the hopper of most products.

D-8.3.5 Mechanically Simple (few moving parts as possible)

Here again, the flex shaft attains a high rank; a single rotating shaft is quite simple in nature. The use of a VSD or electric motor in the case of the pneumatic system adds a degree of complexity to the system, and the requirement of a coupling between shafts adds some complexity. The forward facing wheel design still employs several moving parts (chains and gears). The hydraulic system is fairly simplistic in that it only uses two moving parts (pump and motor).

Appendix E: Letters of Permission

Valmar Airflo Inc.

Hi Dallas,

A reservoir with filter would probably cost us around $200.

Each hydraulic line would cost approximately $30 with fittings.

Each hydraulic motor would cost $200.

You also have permission to use photos from our website for your project.

Thanks,

Dennis

----- Original Message -----

From: Dallas Gade

To: Dennis Rice

Sent: Monday, November 14, 2011 9:29 PM

Subject: Costs

Good Evening,

We are busy working on the final designs and report. We were hoping to get some costs for a final cost analysis on the alternate designs. We need:

-Cost of the reservoir

-Cost of the hydraulic lines

-Cost of the hydraulic pump

We also need permission to use pictures from you and off the Valmar website for the final report. The other report weren't official so they did not require permission, but the final reports will be available in the engineering library so we must ask for permission to use them.

Thanks,

Dallas

Engineering Toolbox

Gmail - Permission to use Figure Rob McDougall <robmcdougall86@gmail.com>

Permission to use Figure

1 message

Rob McDougall <robmcdougall86@gmail.com> Tue, Nov 29, 2011 at 11:00 AM

To: editor.engineeringtoolbox@gmail.com, robmcdougall86@gmail.com

Hi there,

I am a Mechanical Engineering student at the University of Manitoba in Winnipeg, Canada. I am involved with writing a technical report involving a hydraulic drive system for an agricultural application. In presenting some of the hydraulic theory/background, we are discussing the invalidity of assuming that fluids are incompressible, and are hoping to include a figure from EngineeringToolbox.com depicting the change of water density with temperature. I'm referring to this figure - http://docs.engineeringtoolbox.com/documents/309/water-density-temperature-pressure_2.png from this page - http://www.engineeringtoolbox.com/fluid-density-temperature-pressure-d_309.html. Basically, I am wondering if you will grant us the permission to use the figure in question. This report is purely for educational purposes, but will be made public following submission, thus the requirement of your permission. Image credit will be given immediately following the image. All I would require is an email from somebody in a permission of authority documenting the granting of permission.

Thanks in advance for your consideration.

- Rob McDougall

Parker

Hi there,

I am a student at the University of Manitoba in Winnipeg, Canada. I am involved with writing a technical report involving a hydraulic drive system for an agricultural application. In presenting some of the hydraulic theory/background, we are hoping to include an actual performance curve (or curves) of typical gerotor pumps. Parker has made such curves readily available online, however we require permission from you (Parker) to use any proprietary figures due to copyright regulations.

Basically, I am wondering if you will grant us the permission to use some images of performance curves for the TB series LSHT gerotor pumps as found in the pump manuals available online. This report is purely for educational purposes, and Parker's figures are requested as Parker is well known to be a reputable manufacturer in the field of hydraulics. All image credit will be given immediately following the image. All I would require is an email from somebody in a permission of authority at Parker documenting the granting of permission.

Thanks in advance for your consideration.
- Rob McDougall

Elliott Manufacturing

Pat L

10 nov.

à Jeff

Jeff,

Thanks for the insight,

Yes, we use Solidworks here also. What do we need to do for the non-disclosure agreement?

I should mention that our final project report will be made public, i.e. it will be available for viewing in the engineering library here at the U of M. It will also be posted electronically and will be accessible for free to a worldwide audience from the University of Manitoba's digital repository called MSpace located at http://mspace.lib.umanitoba.ca/index.jspand from Library and Archives Canada's Theses Portal located at http://www.collectionscanada.gc.ca/thesescanada/index-e.html.

If this is too much infringement on your proprietary design, we were wondering if we could still use pictures from Elliot Manufacturing's website and the FlexSeeder PDF you handed me in our report. We also have a copyright permission form that we could pass on to you.

Regards,

Patrick

2011/11/9 Knight, Jeff <Jeff.Knight@elliottmfg.com