NEW WHEEL ASSEMBLY DESIGN ON GRÉGOIRE-BESSON SP/SPL PLOUGH RANGE

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25 November 2008
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NEW WHEEL ASSEMBLY DESIGN
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BREE 495 – DESIGN 3

25 NOVEMBER 2008

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New wheel assembly design
on Grégoire-Besson SP/SPL plough range

1. Abstract

Following the requests of many farmers using Grégoire-Besson SP/SPL Series ploughs, a new undercarriage concept was developed to meet the needs of more precise depth control, improved field and road stability, and increased rollover torque reserve. Extensive analyses were conducted to assure optimal design life, maximum versatility and an attractive design cost.

The design was developed for compatibility with Grégoire-Besson SP/SPL ploughs; however the same wheel assembly concept can easily be applied to competitive products. The only modifications required are at the connection interface between the wheel assembly and the main plough frame.

The final result of this design project is a virtual prototype complete with all the required design calculations. Due to the limited time available for the completion of this project, it was not possible to build a field scale prototype for testing. However, given the interest of an industrial partner, the design will be very easily implemented and the manufacturing of the prototype should not take more than a few days for an experienced shop worker.

Keywords:
- Tillage equipment
- Conventional tillage
- Plough
- Wheel assembly
- Agricultural mechanics

2. Acknowledgements

This design would not have been possible without the support and help of many people from the Department of Bioresource Engineering at McGill University; more specifically, I would like to acknowledge Dr. G.S.V. Raghavan for his guidance and support throughout my project; and I would also like to thank Dr. E. McKyes for his technical help and professional advice.

I would also like to thank the people at Grégoire-Besson Canada, in St-Hyacinthe, Quebec; specifically I would like to show my appreciation for Simon Bourque, Yvon Grégoire and Alexandre Giard, who spent many hours answering all my questions.

Finally, I would like to thank Mr. Gerard Steenbeek and Mr. Warren Schneckenburger for their input and their opinion on the final design. The suggestions of these two farmers, who are typical target users, were essential to ensure the completeness of the design.
3. Introduction

Agriculture has experienced tremendous changes during the second part of the 20\textsuperscript{th} century. Numerous technological and scientific developments during the post-war period have been widely adopted by farmers in developed countries to boost farm productivity and move agriculture from a subsistence activity to a globalized industry. Chemicals use has been one of the major changes, with a widespread utilization of pesticides to control crop pests and commercial fertilizers to balance crop nutrient inputs. Methods of doing field operations have changed, too. In North America, the last two decades saw the gradual replacement of conventional mouldboard ploughing with new conservation techniques such as no-tillage and minimum-tillage using chisel ploughs and discs. Europe is also experiencing a change of tillage practices, albeit at a slower pace than in North America. Many agronomists and machinery specialists have seen in this trend the potential extinction of ploughing as a tillage operation.

There are many factors that make ploughing less attractive in comparison to conservation practices: high fuel cost, relatively slow work rate, high tractive force required, etc. Furthermore, ploughing is often blamed as one of the main causes of soil erosion; this results from the minimal amount of crop residue left at the surface. Although these concerns are totally justified, it is important to make some distinctions based on regional differences. Conservation practices are particularly well adapted to climates where large amounts of crop residue do not impede natural soil drying and warming; these dry and warm climates are mostly prevalent in the American Midwest, the Canadian Prairies, Argentina, Brazil, South Africa and Australia. However many other important arable regions such as most of Europe and North Eastern North America experience high rainfalls and have relatively short growing seasons. In these cases, ploughing can be beneficial as part of a diversified tillage system.

The latter regions often receive large amounts of rain during harvest time, resulting in less than adequate soil conditions for harvest operations. Heavy harvest machinery traffic often results in ruts and other soil damages which are best remedied by ploughing. In comparison to other primary tillage operations, ploughing is the method that produces the best results under wet soil conditions. Also, because it buries most crop residue, it leaves soil exposed for faster drying and
warming in the spring. The burial of crop residue is also very important for some specialty crops (peas, beans) which must not be contaminated with soil. The fact that soil is completely moved to a depth of 150-250 mm also makes of the plough an excellent tillage instrument for roots and tuber crops (carrots, potatoes, etc.) which require a loose seedbed with not surface compaction.

In addition to the potential technical benefits of ploughing, the technique may soon make a come back in our fields as a method of reducing our dependency on chemicals, particularly pesticides. The European Parliament voted a new directive in 2007 calling for 50% reduction of active pesticide ingredients used in agriculture by 2017 (European Parliament, 2007). Ploughing is a method of choice to control crop pests and weeds. In particular, it is a very important and effective tool to control slug populations which are extremely damaging to wheat and oilseed rape crops in Western Europe.

Ploughing still has its place as a major tillage practice. However, to gain the most from the technique it is important to include it in an integrated sustainable farming system that includes tillage rotation, crop rotation and pesticides rotation.

As farmers have to compete in a global agricultural industry, it is important to have efficient equipments that will help reducing production costs and increase productivity. Following the requests of many plough users, the aim of this design project was directed at the improvement of the concept of wheel assembly currently used by most major plough manufacturers for their 4 to 8 furrows ploughs. The concept was entirely based on figures and values obtained from Grégoire-Besson for the SP/SPL plough range. The design process lead to substantial redesign of the existing concepts, but meets all the requirements set by final users.
4. Rollover plough concept

The concept of rollover plough originates from Europe, where it has supplanted the traditional one-way plough many years ago. There are many advantages to the use of a rollover plough over a conventional plough: it allows for more efficient operation in the field, it helps maintain fields well levelled and it has a better penetration capability. These positive points all contributed to the increasing interest in this type of machine, despite the higher price tag.

The current product offering on the market is quite wide. It can be divided in two main types of ploughs: centre-carriage ploughs (fig. 1) and single wheel ploughs (fig. 2). On the earlier one, the wheel assembly consists of two wheels on a rigid axle, installed between the front and rear parts of the plough; the latter is equipped with a single wheel which can be positioned on the side (fig. 3) or at the back (fig 4).

Figure 1: Centre-carriage plough (G-B, 2007). Figure 2: Single wheel plough (Thirouin, 2008).

Figure 3: Side-mounted single wheel. Figure 4: Rear-mounted single wheel.
To activate the rollover mechanism, two different methods can be used: the most common system consists of a pair of hydraulic rams that work in succession to move the plough; the other system, used on the reference Grégoire-Besson SP/SPL plough range consists of a rack-and-pinion system. Two 150 mm hydraulic rams move a rack and convert this linear movement to a rotational movement using a pinion.

Figure 5: Grégoire-Besson SPLB9 plough during the turnover process (Thirouin, 2008). Inset: Rack-and-pinion turnover mechanism (Steenbeek, 2007).
5. Problem statement

The Grégoire-Besson SP/SPL plough range was originally developed in the early 1980’s; it was designed to handle 4 to 6 furrows and tractors up to 135 kW (180 hp) (Thirouin, 2002). However, needs have changed over time and the chassis was modified to accommodate additional furrows, new rock protection systems and various options such as coulters, trashboards, etc.

The SP/SPL plough range is suffering from many problems related to the positioning of the transport wheel on the frame. Of these problems, the main one is the lack of torque to complete the rollover process; the position of the wheel and axis of rotation is too far from the centre of gravity of the plough. All the optional equipments that are added to the frame also contribute to increasing the rollover torque. Also, damp conditions can emphasize the problem as soil sticks to the mouldboards and crop residue accumulate around the shanks (Thirouin, 2000).

Another impact of the positioning of the wheel is the behaviour of the plough in wet and hard soils. Many Canadian customers report that in very difficult conditions the tractor has very little control over the plough (Thirouin, 2000; Grégoire-Besson, 2004). This is mostly due to the positioning of the line of traction; a possible solution may be to put the wheel closer to the centre of gravity.

Many users mentioned that one of the weakest points in the machine design is that the Z-frame can interfere with the wheel during the turnover process if the rear of the plough is not raised first (“Vive la difference”, 2008; Thirouin, 2002). However, raising the backend to leave clearance for the Z-frame makes the plough unstable during turnover and increases the stress on the tractor linkage (Thirouin, 2002). Poor soil conditions on the headlands, such as ruts and holes, can also cause instability.
A design constraint also causes issues to customers who want to have a larger wheel to improve manoeuvrability: the current positioning of the wheel in the Z-frame limits tire diameter to less than 1100 mm. Many users would like to use 24R20.5 flotation tires (\(\Phi\) 1260 mm); this requires a repositioning of the wheel.

Finally, another issue with the current design is that with its current positioning, the rear wheel acts as a pivot point on the plough frame. When the tractor hitch is raised due to uneven terrain or when exiting a furrow, the plough pivots about the wheel and the bodies behind the wheel go deeper. In very wet soil conditions as encountered in Eastern North America, having a single on-land wheel is problematic as it sinks in mud, thus increasing the working depth at the back. Both of these conditions lead to an uneven work depth and quality. A current user (Mr. Gerard Steenbeek) and a potential buyer (Mr. Warren Schneckenburger) of SP series ploughs gave me their opinion on the current product and suggested some improvements. Mr. Schneckenburger (personal communication, January 2008) suggests that the plough should have two wheels at the back: one in the furrow and one on land. This would allow a better depth control in wet soil conditions as the furrow wheel will be on hard soil.
6. Test results

In order to assess the magnitude of the rollover mechanism problems, tests were conducted at the Grégoire-Besson Canada facility in St-Hyacinthe, Quebec. A summary of the test report follows.

<table>
<thead>
<tr>
<th>Date of test: 31 July 2008</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Machine details:</strong></td>
</tr>
<tr>
<td>- Grégoire-Besson SPWP9 6-170-114</td>
</tr>
<tr>
<td>- Serial number: 716750</td>
</tr>
<tr>
<td>- Configuration: 1 + 4 + 1 furrows Variwidth</td>
</tr>
<tr>
<td>- Leaf springs rock protection system</td>
</tr>
<tr>
<td>- TA H6 mouldboards</td>
</tr>
<tr>
<td>- CCM-98 coulters Ø 560 mm (only 5 installed, missing the rear one)</td>
</tr>
<tr>
<td>- Adjustable trashboards</td>
</tr>
<tr>
<td>- Hydraulic weight transfer kit</td>
</tr>
<tr>
<td>- Estimated weight of plough (moving section only): 32880 N</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th><strong>Observations:</strong></th>
</tr>
</thead>
<tbody>
<tr>
<td>- At the end of the previous day, after installation of the trashboards and coulters, it is impossible to get the plough moving.</td>
</tr>
<tr>
<td>- On 31 July, a manometer is installed on the hydraulic line that controls rollover cylinder. Pressure climbs to 20 MPa (2900 psi), but plough will not move (Variwidth set at narrowest position [343 mm] and front furrow adjustment completely closed).</td>
</tr>
<tr>
<td>- A second test is performed with different adjustments (maximum opening of the front furrow adjustment and Variwidth set at 343 mm): the plough rolls over with no difficulty.</td>
</tr>
<tr>
<td>- Since playing with front furrow adjustment is not an acceptable solution, pressure limiters are removed and 3/8” hoses are replaced with 1/2” hoses. Roll over is now possible even when set at the widest work settings, but pressure still climbs high at 17.25 MPa (2500 psi).</td>
</tr>
</tbody>
</table>

From the results of this experiment, it was clear that the capacity of the rollover system was stretched to its limits. For a six furrow plough in the specified configuration, the following results apply:

- At a pressure of 17.5 MPa, with 1/2” hydraulic hoses and pressure limiters removed, the maximum distance between the axis of rotation and the centre of gravity is 1310 mm.
- With the pressure limiters on and 3/8” hydraulic hoses, the restrictions within the hydraulic system are excessive and the rollover mechanism cannot supply the necessary torque to start moving the plough.
7. Design objectives, specifications and requirements

The selected solution to solve the problems stated in the previous section is to develop a new wheel assembly concept that comprises two wheels, one of which operates in the furrow and the other one operating on land.

In order to design the new wheel assembly appropriately, an extensive list of design specifications and requirements was established. These specifications are the basis for all the subsequent design work and decisions. They are based on future use, technical capabilities and manufacturing processes.

1. **Design life: no less than $10^5$ work cycles (rollover and lift/lower)**
   Design life must be approximately equivalent to 500 hours of work per year (1500 ha/year) for a period of 10 years. This is a rather extreme scenario, but it takes into account the heavy use made of this type of machine in large agricultural operations in Russia and the Community of Independent States (CIS), which are two target markets for the revised SP range.

2. **Modularity of design**
   Wheel assembly design must allow for easy upgrading of components or expansion of plough (adding one or two furrows for example). To achieve this, all connections with the existing plough frame must be bolted.

3. **Ease of maintenance**
   The number of maintenance points must be kept to a minimum or, alternatively, the maintenance interval must be extended (i.e. seasonal or weekly greasing vs. daily or every 10 hours).

4. **Maximize components commonality with other models or product ranges**
   This allows reducing production cost, simplifying manufacturing operations and improving access to spare parts for customers.

5. **Hydraulic components adaptability**
   It must be possible to replace easily hydraulic components to fit different markets (i.e. Europe vs. North America).
6. **Possibility of fitting high floatation tires**
   Design must allow fitting tires up to Ø1260 mm (24R20.5) with sufficient clearance.

7. **Sufficient rollover capacity**
   The capacity of the rollover mechanism must offer sufficient torque reserve for a fully equipped 8 furrows plough.

8. **Sufficient ground clearance and working depth**
   Ground clearance must be at least 300 mm and maximum work depth must be at least 250 mm with 24R20.5 tires.

9. **On land wheel must have a minimal footprint**
   European markets require ploughs that can work very close to ditches and hedges; therefore the on land wheel must work as close as possible to the last furrow.
8. Patents review

The literature review for this design consisted mostly of an overview of industrial patents, as well as a review of the designs currently used in the industry.

There are numerous patents that apply to wheel assembly systems on ploughs. All of them are registered in Europe as all the major plough manufacturers are based on the Old Continent. Among the most relevant patents for this design project, there is one which was filed by Charrues Bonnel (France) in 1990 (Bonnel, 1990). The patent was issued for an innovative rollover plough using two rear wheels on a rigid rear axle. Another interesting feature of the wheel assembly of this plough is that both wheels are equipped with an independent direction system which allows for tight turns on headlands (fig. 6). The concept used on the Bonnel “Quart-de-tour” plough is very similar to that of a centre carriage plough with the rear section removed. However this specific model of plough was not a success due to many reasons: for example, it was impossible for farmers to work close to ditches due to the wide stance of the rear axle. Furthermore it was impossible to adapt a variable working width system on the chassis.

Figure 6: Bonnel “Quart-de-tour” plough (GTP, 2008).
Another very similar concept was the one developed by Michel Bugnot in 1992: again a rigid rear axle was to be installed at the back or a rollover plough, but this time the main goal was to have a chassis where to install tillage and seeding implement to create a one pass tillage-seeding train (Bugnot, 1992).

Another interesting concept comes from Kverneland AS. The PW/RW plough range is a centre carriage plough that can be used in 3 configurations: centre carriage configuration, mounted rear plough or semi-mounted plough. The latter configuration is the most interesting for this design; it is a semi-mounted plough with two wheels mounted on a rigid axle at the back of the plough. A traction bar acts as the primary link between the headstock and the rear axle. One of the major advantages of this design is that it permits the use of a Variwidth system, which is often seen as a necessity by farmers. However, the major drawback of this concept is that once again the width of the wheel stance is too large to work close to hedges and ditches (approx. 1800 mm).

Figure 7: Kverneland PW/RW plough, in semi-mounted configuration (Kverneland, 2006).
Figure 8: Kverneland PW/RW plough at work in semi-mounted configuration (Farmphoto, 2008).

Figure 9: View of the wide stance of the rear axle, PW/RW plough (Farmphoto, 2008).
9. Design method

Of all the possible design strategies, the one that was adopted to design the wheel assembly was bottom-up design. It seemed to be the most appropriate method for achieving the design goals in a relatively short period of time. Bottom-up design is a very CAD centric approach, so working on Pro/Engineer was a major time component of the complete design process.

Once the initial ideas were laid down on paper (sketches, photos, etc.), the design process started by designing the basic system parts (beams, connections, etc.) with estimative dimensions. Once all the main components were designed, the more detailed components (cylinder brackets, pins, etc.) were designed using the dimensions of the base parts. Components were then put together in sub-assemblies and then in assemblies. The sub-assembly and assembly levels required more work as all the components had to be accounted for so that there would be no interference would occur. Finally once a clearly defined geometry was generated, the final step was to perform the various design calculations to size appropriately the various parts.

Although bottom-up design can be a very efficient method of designing a machine or machine components, it also has some drawbacks. One of the major one encountered during the realization of this design project was that with all the calculations performed after a complete machine was created, if some design parameters were wrong, it was likely that numerous parts of the assembly would have to be completely redesigned. This is exactly what happened at one point during the project.

A better approach that could have improved the workflow would have been to design sub-assemblies, on which the various engineering calculations would be performed. By having intermediate calculation steps within the design process, modifications would only affect a limited number of components rather than the whole machine
10. Design approach

The optimal design solution for the wheel assembly consists of independently controlled wheels installed on a cross beam. The new design does not make use of the current SP/SPL frame (fig. 10); the beefier SPS series frame is used. Using this new frame allows to eliminate the need for a traction bar that goes to the wheels: it is connected behind the third furrow (fig. 11).

Figure 10: Grégoire-Besson SPP9 plough viewed from above (Thirouin, 2008).

Figure 11: Grégoire-Besson SPSP9 plough.
To meet the need for a modular chassis, the method of connection between the plough frame and the wheel assembly will be a bolted joint. The SPS plough frame consists of a 180 mm x 180 mm and a 150 mm x 150 mm tubes welded together (fig. 13). For the new connection, a 20 mm thick flange is welded on the tube assembly; the flange has 16 holes (Ø 20 mm). This part is a standard part already used elsewhere on the frame, and consequently it does not require redesign or modifications. This flange plate mates with another flange which is welded to a 200 mm x 300 mm tube. In the current design, the tube is cut at an angle of 19.33º (equivalent to 400 mm per furrow and 1140 mm front to rear clearance between shanks); it is possible to install pivot pin at this location the case of a Variwidth plough; although this alternative was not investigated due to time limitations. Following the angled tube, another 200 mm x 300 mm tube is bolted using the same flange system. This straight tube has for main goal to provide the necessary clearance for the last shank when it hits an obstacle. Finally at the end of the beam assembly is a 32 mm thick circular plate welded onto a flange plate. This plate is where the rollover shaft connects to the plough chassis.
Figure 13: Pro/E model of the new connection interface.

Figure 14: Close up of the rollover shaft connection.
One the major design issues pertained to how the main chassis would rotate around the centre axis of the new wheel assembly. Initially, the possibility of a planetary system was investigated. The main chassis would be connected to the sun gear, and would remain centered during the rollover process thanks to the planet gears running on the ring gear. However one of the major disadvantages of choosing an epicyclic gear train is that it cannot deal with axial loading. This loading, which is estimated at approximately 5000 N, would have to be transferred from the wheel assembly to the main chassis without going trough the gear system. Cost is another sensitive issue, as the design and manufacturing of planetary system is quite expensive due to tight tolerances. The risks of failure are also increased due to the complexity of epicyclic gear trains. After evaluation various other possibilities, it became clear that the ideal system would consist of a flanged shaft constrained into a housing (fig. 15).

Figure 15: Cutaway section of the rollover mechanism.
The necessary clearance between the shaft flange and the housing were kept to a minimum to reduce shaft movement and uneven wear. A clearance of 0.5 mm is applied all around the flange. This is sufficient to allow appropriate lubrication with regular lithium grease (agricultural grade). Four lubrication points are installed: two on the front plates and two on the rear plate. An o-ring (5 mm diameter) is used in a machined slot on the front plates; the main reason for its use is to seal lubricant inside the housing and keep dust and debris out. The seal also allows some pressure buildup inside the housing and creates mixed-film lubrication that helps reduce wear (Juvinall & Marshek, 2006). The friction coefficient between the machine shaft and the housing is approximately 0.08.

Figure 16: Complete Pro/Engineer model of the wheel assembly.

The wheel assembly itself consists of a 150 mm square tube (HSS). This beam is clamped rigidly to the rollover mechanism using bolts and holding plates. Hydraulic cylinder brackets and wheel arm brackets are welded to this main beam. Initially it was planned to use a curved wheel arm as the sole component of the lifting system. The hydraulic cylinders would push the arm down and raise the machine. However an estimative analysis using Pro/Mechanica finite element analysis
package showed that the complex geometry of the arm would result in excessive deflection in the wheel arm (more than 140 mm at the end of the wheel arm). Furthermore, more detailed calculations proved that the geometry required excessive hydraulic force to lift the plough. In consequence the complete lifting system was redesigned, taking into account the new geometry limitations.

Figure 17: Original curved wheel arm deflection analysis.

Figure 18: Original wheel assembly with curved wheel arms and vertical lift cylinders.
Extensive geometry analysis was required to develop a replacement concept that met all the design criteria and technical constraints. The updated design consists of two individual 100 mm square solid bars which pivot around two pins each (see fig. 20). To make the most efficient use of this geometry, it was decided to install angled cylinder brackets on the solid bars. With a distance of 525 mm between the top cylinder pin and rotation axis, the force of the hydraulic cylinder is maximized thanks to the large moment arm created. However, due to dimensional constrains of the hydraulic cylinder, it is not possible to have the cylinder in a horizontal position when maximum force is required (when the machine has to be lifted from the lowest position). This means that only a certain percentage of the total linear force is available for lifting; maximizing geometry allowed improving this percentage to approximately 87%.

At the end of the beam is the wheel arm itself. It is made of 32 mm thick plate and is welded around the end of the solid bar. The new wheel arm design is straight rather than curved; as a result the arm can handle larger loads before bending. Only the end part of the arm is slightly curved (R 300 mm). The length of the spindle, which is welded to the arm, can therefore be reduced significantly and bending is less likely to occur.

In order to reduce maintenance and to maximize design life, all pins are greasable. Locking plates and M8 x 16 bolts are used to prevent pin rotation (fig. 19).

![Figure 19: Upper cylinder pin details.](image-url)
Figure 20: Wheel assembly chassis components.
Due to the limited experience of the design team in terms of hydraulic system design, it was decided that hydraulic system design would be limited to component selection and sizing (hydraulic cylinders and accumulator system). The primary functions of the hydraulic system installed on the wheel assembly are to raise and lower the plough. In addition, a hydraulic accumulator is installed to provide a dampening effect by limiting the pressure peaks and valleys caused, for example, by hitting obstacles or falling in a hole, respectively.

Another purpose of the hydraulic system is to help control depth. Since wheels are alternating their position (on-land/in furrow), some kind of phasing device must be used so that the wheel with the lowest setting (the one in the furrow) is the one toward which the plough is rolled over. Therefore, in addition to electronic control valves to adjust work depth, it is necessary to phase the lifting cylinders with the rollover cylinders on the headstock. Ultimately the goal is to be able to program all these variables (work depth, dampening effect, cylinder phasing, etc.) so that they can work with ISO11783-compatible virtual terminals (ISOBUS) now available in most modern tractors.
11. Materials and manufacturing

For the purpose of manufacturing and purchasing simplicity, all the structural components (beams, bars, steel plate, etc.) are made of AISI 1040 steel (table 1). This steel is very commonly used in manufacturing industries and is readily available in a wide variety of sizes and finishes through a number of mills. It is very difficult to obtain exact prices due to the very unstable steel market, but as a general rule, AISI 1040 sells for approximately $2.50 per kg (Novatek, 2008).

All the pins used on the machine are standardized to reduce manufacturing cost and improve manufacturing efficiency. Rather than machining all pins to specific diameters, a standard diameter of 30 mm, which meets all the design requirements, is used. The material used for pins is 30CND8 steel (table 1), which comes from France. It can be purchased in 6 metres long bars, which are then cut to the appropriate length, chamfered and drilled. The cost of this special alloy is approximately $4.50 per kg (Dudas, 2008).

Finally, the rollover shaft has to be manufactured out of high strength steel in order to resist the substantial bending stresses of the frame. The selected specification is AISI 4340 (table 1). Due to the high level of accuracy required for this shaft (very tight tolerances), it will have to be machined at a specialized machine shop that can also produce commercial quality polished surfaces. Although there is no price available anywhere for this alloy, its price can estimated to approximately twice the price of AISI 1040 steel, or $5.00 per kg.

<table>
<thead>
<tr>
<th>Material properties</th>
<th>AISI 1040</th>
<th>30CDN8</th>
<th>AISI 4340</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density (kg/m$^3$)</td>
<td>7845</td>
<td>7850</td>
<td>7850</td>
</tr>
<tr>
<td>Ultimate strength (MPa)</td>
<td>518.8</td>
<td>1350</td>
<td>1279</td>
</tr>
<tr>
<td>Yield strength (MPa)</td>
<td>353.4</td>
<td>1050</td>
<td>861.8</td>
</tr>
<tr>
<td>Brinell hardness (H_B)</td>
<td>149</td>
<td>320</td>
<td>363</td>
</tr>
<tr>
<td>Modulus of elasticity (GPa)</td>
<td>210</td>
<td>210</td>
<td>210</td>
</tr>
</tbody>
</table>

Table 1: Mechanical properties of the main steels used (Juvinall & Marshek, 2006; Böhler, 2007)
12. Benefits of the new design

There are numerous benefits to the new wheel assembly design. The first and most important advantage comes from the higher rollover torque reserve (please refer to Appendix XX for details). This 40% torque reserve will boost headland operations productivity, reducing rollover cycle duration significantly. This will be further emphasised by the improved stability of the plough during the rollover process, thanks to two wide support points at the back rather than a narrow one. The time savings will also come from the elimination of the need to stop every once in a while to remove the debris and soil that accumulate on the shanks and mouldboards.

Another advantage of using the new wheel assembly system will be the improved work quality. This is the result of a better depth control achieved by the rear wheel in the furrow. Since the furrow bottom is harder than the soil surface, any loss of support of the on-land wheel will immediately be compensated by the better supported in-furrow wheel.

By choosing to replace the current SP series frame by a SPS series frame, it was possible to greatly simplify the plough. It has less moving components, fewer fasteners, less maintenance points, all because of the absence of a full length traction bar connecting the headstock to the rear wheel. In total, the new wheel assembly has 12 greasing points (4 on the rollover mechanism, 6 greasable pins and 2 greasing points on hubs). This is only one more maintenance point than the original single wheel design (Grégoire-Besson, 2007), even if the new design has more parts.
13. Cost analysis

It is necessary to conduct a financial analysis of the cost of the design process itself to make sure that this cost meets the company’s financial objectives in research and development. An accurate analysis of this cost is necessary as this investment in R&D will be reflected on the final cost of the product.

The data used to calculate the various hourly rates are based on industry standards for medium engineering firms in Canada. However it should be noted that these references are for consulting firms doing work for external customers. I assumed that they are equivalent the rates used in manufacturing firms. Rates and fees are given in Table 2. Rates for engineers and technicians include administrative fees (secretaries, accounting, legal fees, office supplies, etc.).

<table>
<thead>
<tr>
<th>Item</th>
<th>Hourly rate (CAS)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Engineer</td>
<td>75.00</td>
</tr>
<tr>
<td>CAD technician</td>
<td>32.00</td>
</tr>
<tr>
<td>Computer modelling (Pro/E)</td>
<td>10.00</td>
</tr>
<tr>
<td>Computer modelling (FEA)</td>
<td>10.00</td>
</tr>
<tr>
<td>Travelling</td>
<td>0.50 per kilometre</td>
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<tr>
<td>External consulting</td>
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</tr>
</tbody>
</table>

Table 2: Rates and fees for design.
<table>
<thead>
<tr>
<th>Task</th>
<th>Item</th>
<th>Time (h)</th>
<th>Cost (CA$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Preliminary design process</td>
<td>Engineer</td>
<td>25</td>
<td>1875.00</td>
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<tr>
<td>Analysis of current design</td>
<td>Engineer</td>
<td>5</td>
<td>375.00</td>
</tr>
<tr>
<td>Evaluation of forces on wheel assembly</td>
<td>Engineer</td>
<td>30</td>
<td>2250.00</td>
</tr>
<tr>
<td>Modelling of turnover forces based on plough configuration</td>
<td>Engineer</td>
<td>2</td>
<td>150.00</td>
</tr>
<tr>
<td>Qualitative analysis of SP series plough and competitive designs</td>
<td>Engineer</td>
<td>10</td>
<td>750.00</td>
</tr>
<tr>
<td>Stress analysis in structural components</td>
<td>Engineer</td>
<td>20</td>
<td>1500.00</td>
</tr>
<tr>
<td>Finite elements analysis of stresses (estimate if performed)</td>
<td>Engineer, FEA software</td>
<td>10</td>
<td>750.00, 100.00</td>
</tr>
<tr>
<td>Material selection</td>
<td>Engineer</td>
<td>5</td>
<td>375.00</td>
</tr>
<tr>
<td>Hydraulic system design</td>
<td>Engineer</td>
<td>10</td>
<td>750.00</td>
</tr>
<tr>
<td>Hydraulic components design</td>
<td>Engineer</td>
<td>3</td>
<td>225.00</td>
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<td>Mechanical components design and selection</td>
<td>Engineer</td>
<td>15</td>
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<td>Qualitative analysis of design options</td>
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<td>Final design report development</td>
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<td>External consulting</td>
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<td>Travelling fees</td>
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<td><strong>Total (CA$)</strong></td>
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<td></td>
<td><strong>22625.00</strong></td>
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Table 3: Design process budget details.

It is quite easy to find the estimated cost per unit sold. Assuming that 25 units are sold every year, that the design is used during 5 years before being replaced by a newer design (amortization period of 5 years), and estimating the interest rate at 6%, the total research and development cost per unit sold is approximately CA$215.00. However, prototype building, design of machining and manufacturing processes, jig design and manufacturing are all additional costs not included in the R&D cost analysis presented here. These expenses are likely to represent a significant sum of money.
The other important financial aspect is the cost of manufacturing the wheel assembly. This cost includes the cost of fasteners, steel, steel processing (plasma or laser cutting), welding, hydraulic components, hubs, rims and tires, in addition to work time. The total cost of manufacturing is extremely difficult to calculate for many reasons. The first one is the extreme volatility of steel prices; due to this suppliers are generally very reluctant to give information about their prices. Even more difficult is that the steel price will also vary widely depending on the quantities purchased. The third factor complicating the analysis of manufacturing cost is steel processing: suppliers need to have complete part plans to be able to estimate the cost of laser or plasma cutting. This is a very tedious job that no supplier accepted to conduct. Since the cost of processing often exceeds the cost of steel itself, it becomes practically impossible to determine the final price of a part. For this reason, there is no detailed analysis of the cost of manufacturing, but rather only an estimate based on components weight.

Overall it is estimated that the main chassis steel components (beams, bars, steel plates, pins), including cutting and other processing, will cost approximately $7500.00. An additional $2000.00 must be calculated for all the hydraulic parts (cylinders, accumulator, valves, hoses, etc.). Tires, rims and hubs account for $1200.00, $600.00 and $1000.00, respectively. Miscellaneous account for approximately $1500.00 (paint, welding, etc.). Finally the estimated manufacturing time (welding, set up of jigs, painting, etc.) is 30 hours at a cost of $40.00 per hour, for a total of $1200.00. The final estimated production cost is $15000.00.
14. Conclusion and recommendations

Because of rigorous design practices and engineering calculations, the design presented in this project meets all the requirements and specifications demanded by users and manufacturers. The new wheel assembly provides an innovative alternative to farmers who need the field capacity of a centre-carriage plough, but the simplicity and lower cost of a single wheel plough. In addition, numerous design features can make this concept plough attractive to potentially interested manufacturers. It is a modular concept that allows easy adaptation to a variety of different plough frames; a manufacturer could easily decide to use the same frame (with minor modifications) on centre-carriage ploughs as well as on ploughs equipped to reduce manufacturing costs.

The design can be used in its current form on fixed working width ploughs. However, future developments of this range of plough will include Variwidth machines. Achieving this will require minor modifications to the frame to install pivot and control arms. This feature will make the new plough range more flexible: it will be possible to make adjustments on the go to adapt to soil conditions and crop residue cover. Another important development that could stem out of this new concept is a larger range, 8 to 10 furrows, with hydraulic flex-frame in the middle of the plough beam. This would likely require some modifications to the wheel assembly to make it strong enough to handle the heavier loads. By offering dual rear wheel ploughs of this size, a manufacturer could tap into the growing markets of Eastern Europe, CIS and Russia. The flex-frame would be very advantageous to maintain consistent depth control in all relief conditions.
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Michelin. (2008). *XS – the soft soil all position radial for special service such as emergency response vehicles.*


Appendix 1: Design Calculations

A1 – Calculations: rolling resistance

Using the following approximation, it is possible to estimate the magnitude of the rolling resistance force (McKyes, 1985):

\[ R = N \times \left( \frac{z + \delta}{D} \right) \]

where:
- \( R \) = rolling resistance force (N)
- \( z \) = sinkage depth (m)
- \( \delta \) = tire deflection (m)
- \( D \) = unloaded diameter (m)
- \( N \) = tire load (N)

Sinkage depth varies depending on soil conditions and on wheel positioning (on land vs. in furrow). In consequence, the following assumptions are made:

- \( z_{\text{max}} = 100 \text{mm} \) in loose soil or muddy conditions
- \( z_{\text{min}} = 10 \text{mm} \) on hard furrow bottom

Tire deflection can be approximated as:

\[ \delta = \frac{D}{2} - r_L \]

where:
- \( \delta \) = tire deflection (m)
- \( D \) = unloaded diameter (m)
- \( r_L \) = loaded radius (m)

The largest allowable tire size for the wheel assembly is selected for rolling resistance calculations. The properties of Michelin XS 24R20.5 tires are as follow (Michelin, 2008):

- \( D = 1.374 \text{ m} \)
- \( r_L = 0.620 \text{ m} \)
\[ \delta = \frac{1.374m}{2} - 0.620m = 0.067m \]

There are two options of weight distribution:

- In normal working conditions, tire loading will be equal on both sides;
- Occasionally there is the possibility that the load will be supported by only one tire (for example when going over a bump); it is necessary to take this scenario into account and calculated the impact on individual wheel arms.

Due to the lack of information concerning the SPS frame properties, in particular weight distribution, we have to assume a total plough weight of 40000 N which is supported equally by the tractor linkage and the wheel assembly. Due to the lack of precise information about the plough properties and to varying soil conditions, a safety factor of 2 is used in the calculations; this takes the machine weight to 80000 N.

- 40000 N supported by the wheel assembly;

\[ R_{\text{min}} = 40000N \times \left( \frac{0.01m + 0.067m}{1.374m} \right) = 2240N \]

\[ R_{\text{max}} = 40000N \times \left( \frac{0.1m + 0.067m}{1.374m} \right) = 4860N \]

For the purposes of this design, a total horizontal draught force of 5 KN will be considered on the wheel assembly.
A2 – Calculations: wheel arm assembly forces

Figure 23: Pro/E representation of hydraulic lifting system.

Figure 24: Wheel arm assembly.

Figure 25: Dimensions and forces on wheel arm assembly.

\[ R_w = 40000 \text{ N} \]
\[ R_R = 5000 \text{ N} \]
\[ R_{PIVOT} = \text{unknown} \]
\[ F_{HYDR} = \text{unknown} \]
Finding $F_{HYDR}$ using moment around A:

$$\sum M_A = (F_{HYDR} \cos \theta \times D_2) + (R_w \times D_1) = 0$$

$$= (4000 N \times 897.5 mm) + (F_{HYDR} \cos(29.38^\circ) \times 525 mm)$$

$$F_{HYDR} = 78474 N$$

Finding $R_{PIVOT}$ using sum of forces:

$$\sum F_x = R_w + F_{HYDR} \cos \theta + R_{PIVOT-x} = 0$$

$$= 5000 N + 78474 N \cos(29.38^\circ) + R_{PIVOT-x} = 0$$

$$R_{PIVOT-x} = -73381 N$$

$$\sum F_y = R_w + F_{HYDR} \sin \theta + R_{PIVOT-y} = 0$$

$$= 40000 N + 78474 N \sin(29.38^\circ) + R_{PIVOT-y} = 0$$

$$R_{PIVOT-y} = -78500 N$$

$$R_{PIVOT} = 107457 N \ @ \ 133.07^\circ \text{ CW from horizontal}$$

Figure 26: Lateral view of hydraulic lifting system geometry.
A3 – Calculations: pin safety factors

Assumptions

- All pins are loaded in double shear.
- $S_{sy} = 0.58S_y$ (Juvinall & Marshek, 2006)
- $S_y = 1050$ MPa for 30CND8 steel
- For two shear planes, limit load $F = 2 \cdot S_y \cdot A$

$$S_{sy} = 0.58 \times 1050 \text{MPa} = 609 \text{MPa}$$

$$F = 2 \cdot 609 \text{MPa} \times \left( \frac{\pi \times (30 \text{mm})^2}{4} \right) = 860.95 \text{kN}$$

Safety factors are as follow:

- Lower wheel arm assembly pin: $SF = \frac{860.95 \text{kN}}{107.46 \text{kN}} = 8.01$
- Lower cylinder pin: $SF = \frac{860.95 \text{kN}}{78.47 \text{kN}} = 10.97$
- Upper cylinder pin: $SF = \frac{860.95 \text{kN}}{78.47 \text{kN}} = 10.97$

All the safety factors obtained satisfy very well the requirements for this type of components ($SF = 6$).
A4 – Calculations: rollover shaft sizing

The rollover shaft is the most critical part in the design of the wheel assembly. The shaft acts as the connection between the wheel assembly and the rotating frame. Furthermore, it is subjected to the wheel assembly draught forces as well as very important bending moments. The shaft consists of a 250 mm diameter flange at the frame end and a 300 mm plate at the wheel end. The latter part of the shaft is necessary to hold the shaft into the rollover housing (see cutaway section in figure 27).

Figure 27: Cutaway view of rollover mechanism.
Point A is the critical point in this shaft. Since the complete geometry of the plough is not clearly defined, a conservative approximate analysis shows that the bending moment at point A is equal to 34.125 kN (using a reasonable machine load of 50000 N).

Assumptions:

- shaft diameter = 150 mm
- use AISI 4340 steel ($S_y = 861.8$ MPa)

Using stress concentration factors (Juvinall & Marshek, 2006), it is possible to calculate the maximum stress resulting from bending moment:

\[
\frac{D}{d} = \frac{200\text{mm}}{150\text{mm}} = 1.33 \quad \frac{r}{d} = \frac{3.75\text{mm}}{150\text{mm}} = 0.025
\]

\[
K_{t, \text{bending}} = 2.4 \quad K_{t, \text{axial}} = 2.6 \quad K_{t, \text{torsion}} = 1.95
\]
\[
\sigma_{\text{nom}} = \frac{Mc}{I} = \frac{32M}{\pi d^3} = \frac{32 \times 34125 \text{Nm}}{\pi (0.15 \text{m})^3} = 102.99 \text{MPa}
\]

\[
\sigma_{\text{max}} = \sigma_{\text{nom}} \times K_{t-h\text{-bending}} = 102.99 \text{MPa} \times 2.4 = 247.18 \text{MPa}
\]

Torsional stress is resulting from the friction of the end plate with the housing.

Assumptions:

- well lubricated surfaces
- friction coefficient \( f = 0.08 \) (Juvinall & Marshek, 2006)
- axial force \( W = 5000 \text{ N} = R_R \)
- Head average radius \( r_h = 0.15 \text{ m} \)

Turning torque \( T \) is calculated with:

\[
T = W f \times r_h = 5000 \text{N} \times 0.08 \times 0.15m = 60 \text{Nm}
\]

Without going any further in this analysis, it is clear that the stress resulting from the rollover mechanism torque is negligible compared to the bending stresses.

Axial loading is the last concern for stresses analysis. Again assuming a 5000 N axial force, axial loading stresses are determined by:

\[
\sigma_{\text{max}} = \frac{W}{A} \times K_{t-\text{axial}} = \frac{5000N}{\pi \times (150 \text{ mm})^2} \times 1.95 = 0.55 \text{MPa}
\]

The axial loading stresses are negligible compared to the bending stresses.
Analysis must also include fatigue strength; this is a primary criterion in the design of the machine.

Assumptions:

- Design life of 6000 hours of work or roughly $10^5$ rollover cycles.
- AISI 4340 $S_U = 1279$ MPa (Juvinall & Marshek, 2006)
- Endurance limit $S_n' = 700$ MPa (Nascimento et al., 2001)
- Surface factor $C_S = 0.9$ for commercially polished, machined material
- Gradient factor $C_G = 0.90$
- Load factor $C_L = 1.0$
- Reliability factor $C_R = 0.814$ for 99% reliability (more than sufficient considering that all these values apply for $10^6$ cycles; no data available for $10^5$ cycles)

$$S_n = S_n' C_L C_G C_S C_R = 700MPa * 1 * 0.90 * 0.9 * 0.814 = 461.51MPa$$

$$\sigma_{af} = S_n \left(1 - \frac{\sigma_m}{S_U}\right) = 461.51MPa \times \left(1 - \frac{\sigma_{af}}{1279MPa}\right) = 339.15MPa$$

$$SF = \frac{339.15MPa}{247.18MPa} = 1.37$$

Although the safety factor is lower than the desired 1.5, we must consider that for this analysis the different factors are based on a design life ten times greater than the targeted design life. Therefore any positive safety factor can be considered appropriate. However it must be noted that for such a critical component, it would be essential to conduct laboratory testing with a shaft of the appropriate geometry and material to make sure that it meets the design criteria.
A5 – Calculations: rollover shaft bolts

Assumptions:
- 8 holes flange
- 4 bolts in tension and 4 bolts in compression
- Assume all 4 bolts carry the total tension
- Safety factor = 5

Select SAE grade 12.9 bolts:
- \( S_p = 970 \text{ MPa} \)
- \( S_y = 1100 \text{ MPa} \)
- \( S_u = 1200 \text{ MPa} \)

Moment at bolted joint: 34.125 kN

Tension at bottom of bolted joint: \( F = \frac{34.125kN}{200mm} = 170.625kN \)

\[ F_{clamping} = \frac{170.625kN}{4} = 42.656kN \]

\[ F_{req.} = 42.656kN \times SF = 213.4kN \]

\[ A_t = \frac{F_{req.}}{S_p} = \frac{213.4kN}{970MPa} = 220mm^2 \]

\( \rightarrow \) from Juvinall & Marshek, 2006, M20 bolt has area = 245 mm²

Effective safety factor:

\[ SF = \frac{245mm^2 \times 970MPa}{42.656kN} = 5.57 \]
Tightening torque is calculated as:

\[ T_1 = W \left( \frac{d_m}{2} \right) \left( \frac{f \pi d_m + L \cos \alpha_n}{\pi d_m \cos \alpha_n - fL} \right) \]

\[ d_m = \frac{d + d_r}{2} = \frac{20\text{mm} + 16.9\text{mm}}{2} = 18.45\text{mm} \]

\[ d_n = 30\text{mm} \quad L = 2.5\text{mm} \]

\[ T_1 = 213400 N \left( \frac{18.45\text{mm}}{2} \right) \left( \frac{0.14 \pi \times 18.45\text{mm} + 3\text{mm} \times \cos 30^\circ}{\pi \times 18.45\text{mm} \times \cos 30^\circ - 0.14 \times 3\text{mm}} \right) = 423.57 \text{Nm} \]

The shaft bolts must be tightened at 423.6 Nm.
A6 – Calculations: hydraulic cylinder sizing

Assumptions:

- Maximum tractor hydraulic pressure $P_{\text{max}} = 2900 \text{ psi} = 20 \frac{N}{\text{mm}^2}$
- Use a safety factor of 1.5 (use of a nitrogen bladder-type damper reduces impact loading)

The maximum theoretical extension force is calculated with

$$F_{\text{HYDR}} = \pi \frac{D_b^2}{4} \frac{P}{SF}$$

where:

- $F_{\text{HYDR}} =$ maximum hydraulic force (extension) (N)
- $D_b =$ bore diameter (mm)
- $P =$ operating pressure (MPa)
- $SF =$ safety factor

$$F_{\text{HYDR}} = \frac{\pi \cdot D_b^2 \cdot 20 \text{MPa}}{4 \cdot 1.5}$$

$D_b = 86.56 \text{mm}$

Closest standard bore: 90 mm.

<table>
<thead>
<tr>
<th>Bore</th>
<th>90 mm</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rod diameter</td>
<td>75 mm</td>
</tr>
<tr>
<td>Stroke</td>
<td>320 mm</td>
</tr>
<tr>
<td>Length (pin centre to pin centre, retracted)</td>
<td>412 mm</td>
</tr>
<tr>
<td>Length (pin centre to pin centre, extended)</td>
<td>732 mm</td>
</tr>
<tr>
<td>Maximum force</td>
<td>127.2 kN</td>
</tr>
<tr>
<td>Safety factor (force) for intended use</td>
<td>1.62</td>
</tr>
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</table>

Table 4: Hydraulic cylinder specifications.
A rod buckling analysis must be performed to ensure that the diameter is sufficient to prevent buckling. Using Euler strut theory (Burton & Bitner, 2008):

\[ K = \frac{\pi^2 EJ}{L_e^2} \]

where:

- \( K \) = buckling load (N)
- \( E \) = Modulus of elasticity (kg/cm\(^2\)) (2.1\( \times \)10\(^6\) kg/cm\(^2\) for steel)
- \( J \) = second moment of inertia for rod (cm\(^4\))
- \( L_e \) = equivalent free length (cm)

For a cylinder and a rod both connected on pivots, \( L_e = 0.5L \) (Burton & Bitner, 2008).

\[
K = \frac{\pi^2 \times 2.1 \times 10^6 \frac{kg}{cm^2} \times \frac{\pi \times (7.5 cm)^4}{64}}{\left(32 cm/2\right)^2} = 12574 kN
\]

From this analysis it is clear that buckling is not an issue with the selected hydraulic cylinder size selected.

![Figure 29: 90 mm bore, 75 mm rod, 320 mm stroke hydraulic cylinder. Inset: hydraulic cylinder in extension.](image)
Work parameters – Michelin XS 24R20.5 tires and 1800 mm shanks

<table>
<thead>
<tr>
<th>Specification</th>
<th>Value</th>
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<td>Maximum work depth</td>
<td>280 mm</td>
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<tr>
<td>Ground clearance</td>
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</tr>
<tr>
<td>Total lifting range</td>
<td>750 mm</td>
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Table 5: Key design specifications for lifting system.

Figure 30: Wheel assembly in lowered (left) and lifted (right) positions.
A7 – Calculations: Hydraulic accumulator sizing

In order to cushion the ride during transport and to absorb part of the shocks when travelling on rough terrain, the wheel assembly’s hydraulic system is fitted with a hydraulic accumulator. A bladder-type nitrogen accumulator was selected due to its numerous advantages (Burton & Bitner, 2008):

- Efficient with large volumes of oil;
- Positive sealing between gas and oil to prevent contamination;
- Quick pressure response, which is ideal for use as a dampener.

Assuming the process to be adiabatic, the following applies:

$$V_1 = \frac{V_x \left( \frac{P_3}{P_1} \right)^{\frac{V_x}{V_1}}}{1 - \left( \frac{P_3}{P_2} \right)^{\frac{V_x}{V_1}}}$$

where:

- $V_1 =$ accumulator capacity (cm$^3$)
- $V_x =$ volume of oil discharged from accumulator (cm$^3$)
- $P_1 =$ gas precharge pressure (Pa)
- $P_2 =$ maximum system pressure (Pa)
- $V_2 =$ compressed volume of gas at maximum pressure (cm$^3$)
- $P_3 =$ minimum system pressure at which additional fluid is needed (Pa)
- $V_3 =$ Expanded volume of gas at minimum system pressure (cm$^3$)

Figure 31: Sizing parameters for hydraulic accumulators (Burton & Bitner, 2008).
Assumptions

- Accumulator must supply enough fluid for the hydraulic cylinders to travel 15 mm: approximately 200 cm$^3$.
- Maximum operating pressure is 20 MPa (2900 psi).
- Minimum operating pressure is 13.8 MPa (2000 psi).
- Precharged with nitrogen at 12.5 MPa (1800 psi).
- $n = 1.4$ for an adiabatic process.

\[
V_i = \frac{200cm^3 \times \left( \frac{13.8MPa}{12.5MPa} \right)^{1.4}}{1 - \left( \frac{13.8MPa}{20MPa} \right)^{1.4}} = 922cm^3
\]

The ideal size is one litre or approximately one quart, which is a standard size in the industry. For this purpose, model A1QT3100 from Accumulator Inc. (Accumulators inc., 2008) was selected.

Figure 32: Nitrogen bladder accumulator model A1QT3100.
A8 – Calculations: Hub selection

The hub was selected on the hypothesis of the entire load being applied to one wheel only, with a safety factor of at least 1.25. For a load of 40000 N, we get a desired hub capacity of at least 50000 N (5100 kg or 11200 lbs). Looking at hub suppliers’ catalogues, the selected hub model is a CTD H1010-1 with a rated capacity of 53509 N. The effective safety factor is 1.34. Standard with this model of hub is a 76 mm diameter spindle (S1010) (Canadian Tools and Dies, 2008).

CTD H1010-1 hub specifications

<table>
<thead>
<tr>
<th>Specification</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rated capacity</td>
<td>53509 N (12000 lbs)</td>
</tr>
<tr>
<td>Bolt circle diameter</td>
<td>285.75 mm</td>
</tr>
<tr>
<td>Pilot diameter</td>
<td>220.73 mm</td>
</tr>
<tr>
<td>Studs</td>
<td>10</td>
</tr>
<tr>
<td>Stud size</td>
<td>M18 x 95</td>
</tr>
<tr>
<td>Spindle model</td>
<td>S1010</td>
</tr>
<tr>
<td>Spindle diameter</td>
<td>76 mm</td>
</tr>
</tbody>
</table>

Table 6: Selected hub specifications (Canadian Tools and Dies, 2008).

Figure 33: CTD H1010-1 hub.
A9 – Calculations: Tire and rim selection

Tire and rim selection were based on design specifications which stated that the maximum recommended tire size should be 24R20.5. Michelin XS series tires are used as the design reference.

Figure 34: Michelin XS 24R20.5 tire mounted on 450 mm x 610 mm rim.

Michelin XS 24R20.5 tire specifications

<table>
<thead>
<tr>
<th>Specification</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Loaded radius</td>
<td>620 mm</td>
</tr>
<tr>
<td>Unloaded diameter</td>
<td>1364 mm</td>
</tr>
<tr>
<td>Overall width</td>
<td>602 mm</td>
</tr>
<tr>
<td>Maximum tire load and pressure setting</td>
<td>7100 kg @ 5.9 bar</td>
</tr>
<tr>
<td>Maximum rim width</td>
<td>610 mm</td>
</tr>
</tbody>
</table>

Table 7: Tire specifications (Michelin, 2008)
Appendix 2: Rollover torque reserve

First we must find the torque generated by the rollover mechanism.

Assumptions:

- Maximum hydraulic pressure $P = 18$ MPa (2600 psi)
- Rollover ram diameter $D = 150$ mm
- Pinion pitch diameter $D_{pitch} = 280$ mm

\[
F = P \times A
\]
\[
F = 18 \frac{N}{mm^2} \times \left( \frac{\pi \times (150mm)^2}{4} \right) = 318.09kN
\]
\[
T_{rollover} = F \times R_{pitch}
\]
\[
T_{rollover} = 318.09kN \times 140mm = 44532Nm
\]

Using the original configuration with a weight of 40542 N
\[
T_{required} = 40542N \times 1127mm = 45691Nm
\]

The required torque to rotate a 7 furrow SPW plough with complete equipment (coulters, trashboards and TA H8 mouldboards exceeds the available torque at the headstock by 2.5%.

Using the same plough configuration, but with the new wheel assembly:

Assumptions:

- Assuming plough weight to be equal to SPW model = 40542 N
- Centre of gravity can conservatively be located at 660 mm from the axis of rotation

\[
T_{new} = 40542N \times 660mm = 26750Nm
\]

The required torque with the new assembly is 40% below the available torque at the headstock.

\[
W_{max} = \frac{44532Nm}{660mm} = 67000N
\]

The maximum theoretical weight for the rollover system to work has increased to 67000N when the centre of gravity is at 660 mm from the axis of rotation.
Figure 35: Rollover torque reserves for various configurations of plough.
Appendix 3: Bill of materials

Total:

- 144 different parts
- 625 components

BOM report: NEW_VERSION_X-BEAM

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<th>Name</th>
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<td>Sub-Assembly</td>
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Summary of parts for assembly X-BEAM_EXT_CYL:

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1  Part  ROLLOVER_SHAFT
2  Part  X-BEAM_ENDS
8  Part  M20X45_BOLT
4  Part  X-BEAM_CYL_BRACKETS
4  Part  X_BEAM_CYL_BRACKET_NO_HOLE
1  Part  AXLE_BEAM_SQ100X250
2  Part  AXLE_BEAM_PIN_PLATE.NEW
1  Part  AXLE_ARM_REVISED
1  Part  CURVE_CYL_PLATE_NO.HOLE
1  Part  CURVE_CYL.PLATE_HOLE
2  Part  AXLE_BEAM_END
4  Part  AXLE_ARM_PIN
8  Part  AXLE_LOCK_PLATE
8  Part  AXLE_LOCK_PIN
8  Part  M8X16_BOLT
2  Part  X_BEAM.NEW_CYL_BRACK.H
2  Part  X_BEAM.NEW_CYL_BRACK.NH
4  Part  WASHER.175
4  Part  WASHER.225
4  Part  HC_MOUNT_PIN.185MM
1  Part  AXLE_BEAM_SQ100X250_MIR
2  Part  AXLE_BEAM_PIN.PLATE.NEW_MIR
1  Part  AXLE_ARM_REVISED_MIR
1  Part  CURVE_CYL.PLATE_NO.HOLE_MIR
1  Part  CURVE_CYL.PLATE_HOLE_MIR
2  Part  AXLE_BEAM.END_MIR
2  Part  HUB.WHEEL
2  Part  AXLE_SHAFT.76MM
8  Part  M5X20_HUBCAP
2  Part  RIM.24R205
2  Part  24R205.TIRE
20  Part  M18.WHEEL.STUD
20  Part  M18.STUD.NUT
1  Part  N_ACCUMULATOR.TOP
1  Part  NITROGEN_ACCUMULATOR.BOTTOM
6  Part  M10X16_BOLT
10  Part  M10.NUT
1  Part  BRACKET1
2  Part  M10X14_BOLT
1  Part  BRACKET2
2  Part  M10X35_BOLT
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1  Part  HSS_SQ180-16_MAIN_FRAME
1  Part  HSS.150_MAIN
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1  Part  ANGLED_BEAM
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**Table 8: Bill of materials**