STUDIES TOWARDS THE DESIGN AND PROTOTYPING OF A RECONFIGURABLE, MULTIPURPOSE UTILITY VEHICLE WITH MODULAR TRANSMISSION

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STUDIES TOWARDS THE DESIGN AND PROTOTYPING OF A RECONFIGURABLE, MULTIPURPOSE UTILITY VEHICLE WITH MODULAR TRANSMISSION

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MECHANICAL ENGINEERING

ABSTRACT

Systematic researches have been performed in the past on the design of modular, reconfigurable machine tools, robots and air and spacecraft. Very few reports are available however on the design of reconfigurable wheeled vehicles that use modular transmissions. This thesis attempts to fill this gap by applying modern Design Theory and Methodology and CAD/CAE tools to the creation of novel, reconfigurable utility vehicles, equipped with modular transmissions. A first embodiment of this concept is a telescopic chassis-frame vehicle that can operate both as a front- and rear-engine tractor or ATV, with or without cargo bed. It consists of the engine-transmission unit of an existing walking tractor that engages with a second gear transmission that provides the transport speeds. In this arrangement, the basic walking tractor will be simpler, thus more affordable to purchase. It will also be safer to operate because the fast transport speeds are available only in the four-wheel configuration that is achieved in conjunction with the second module. A detailed CAD model was generated using Pro\Engineer Wildfire, while engineering analyses were performed using Pro\Mechanica and
Pro\Mechanism. In addition, MATLAB and Excel spreadsheets were used to perform stability and traction calculations and analyses. A digital manikin was employed throughout the design process for ergonomic placement of the vehicle’s controls, and for mass properties determination.

Keywords: wheeled vehicle, gear transmission, reconfigurable machine, Computer Aided Design.
DEDICATION

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LIST OF ABBREVIATIONS

2WD = two wheel drive
3D = three dimensional
4WD = four wheel drive
CVT = continuous variable transmission
FBD = Free-body-diagram
ha = hectare
HP = horse power
kg = kilogram
km/h = kilometer per hour
kw = kilowatt
m = meter
N = Newton
PTO = power take off
ROPS = Rollover Protection Structure
rpm = rotations per minute
CHAPTER 1

INTRODUCTION

The subject of this work is on the design of reconfigurable utility vehicles with modular transmissions. Today, most of the garden machines are reconfigurable in the sense that the same engine-gear box unit can be used in conjunction with various implements and perform tasks like: mowing, mulching, tilling, shredding etc. There is also the possibility to convert a garden machine into a four-wheel utility vehicle. Garden machines that can be converted into four-wheel utility vehicles are equipped with more powerful engine (up to 20 HP) and their transmission is provided with supplementary, faster gears required for transport.

There are a number of disadvantages of the known vehicles as follows: Firstly, they have limited hill climbing and off-road transportation capabilities because of their front wheel drive (it is known that a rear wheel drive and all-wheel drive vehicles are better suited for off-road conditions).

Secondly, their gears cannot be changed in motion because their transmissions are not equipped with synchronizers. The driver needs to stop the vehicle, switch to other gear and then restart.

Lastly, the more powerful walk-behind tractors are heavy and therefore difficult to handle. The increased weight is mainly due to the engine and the complex gear transmission.
Research Goal

The purpose of this research is to design a reconfigurable, modular gear transmission for a utility vehicle of the garden tractor type that can be used both as a front wheel and rear wheel drive vehicle in an economic, reliable and secure way.

Furthermore, this research provides potential solutions to the increased demand for affordable agricultural machineries for smallholder agriculture in the Developing World.

It could also be a solution for the mechanization of local, small organic farms, which have seen an increase in number in recent years.
CHAPTER 2

AN OVERVIEW OF THE DESIGN OF GARDEN MACHINES

Two-wheel tractor, walk-behind tractor or power tiller is the common term used to define a self-propelled machine used in agricultural activities, that is powered by a gasoline or diesel engine of between 4 and 20 HP, and are equipped with a gear transmission of between 1 and 8 forward gears, and between 1 and 4 reverse gears.

The first rotary tiller was invented in 1910 by Dr. Konrad Meyenburg of Basel, Switzerland, when he applied for a patent called “Machine for Mechanical Tillage”. He received the Swiss patent no. 1,018,843 on February 27, 1912. In the same year, Arthur Clifford from Gilgandra, New South Wales, Australia, found that soil could be mechanically tilled without packing, as it was the case with traditional plowing. His tiller device used a steam engine as a power source (Rotary Tiller).

Today there are more than 60 companies that produce almost 400 different models of two-wheel tractors and many more implements. The key concept in designing a walk-behind tractor is to have a main module which consists of: engine, gear transmission and handlebars; and a number of attachments used to perform the required work. Some of the attachments can be simple, for instance: a plow or a blade, others can incorporate moving parts, like gears or chains and sprockets as it is the case of a rotary mowers, snow blowers, rear-tine tillers, sickle mowers etc. But, no matter how simple or complex the implement attachment is, they should have at least one interface that facili-
tates the connection to the main module. Interfaces are grouped in two main categories: those which transmit the torque from the main gear box of the machine via power take off (PTO), and those which do not. Usually gear transmissions are designed with one or two PTOs in front or to the side of the machine. Figure 1 shows the two-wheel drive tractor Agria 2500, and a number of accessories that can be attached to the main unit.

Figure 1: Two-wheel drive tractor model Agria 2500 together with some of its implements
Walk-behind tractors are powered by gasoline or diesel engines. Gasoline engines have power between 4 HP, Kawasaki FE120 D and 20 HP, Briggs & Stratton V-Twin 20 HP. The power of diesel engines is spread on an interval starting with 5 HP, Lombardini 15 LD 225 and ending with 18 HP, TR18R-PM. Figure 2 presents the overall allotment of the two types of engines. The graph is the result of an extensive research over the gardening machinery manufactures. As it can be seen the distribution between gasoline and diesel engines are almost half and half, with slightly more gasoline engines being used.

![Diagram of engine market distribution](image)

*Figure 2: Gasoline and diesel engines market distribution*

As can be seen, tractors are usually equipped with engines with power between 6.1 and 10 HP. Certainly the choice for the engine power comes out after a rigorous engineering research; the main purpose of the figure above is only a statistical result.
The power flow from the engine is transmitted to the wheels through a transmission box. Inside the transmission box there is a friction clutch mechanism that is used during gear shift and for small stops of the machine. Some on the more advanced models are equipped with differentials systems and wheel brakes. Differential systems provide improved maneuverability and ease handling. In conjunction with the wheel brakes, they allow tight turns to be made, by blocking the wheel to the inside of the turn.

Figure 4: Types of drive systems and their occurrence.
The power from the engine is transferred to the transmission box using one belt, chain or gears drives. Figure 4 shows the distribution of each of these types among 289 different walking-tractor models. As can be seen, geared system are more common than the other two.

An example of belt drive system is the model TC14 produced by Agritech-Yanmar, in Brazil (Figure 5). This two-wheel drive tractor has following configurations: low speeds forward: 1.27 km/h; 2.07 km/h; 2.62 km/h; high speeds forward: 4.25 km/h; 9.02 km/h; 14.66 km/h and reverse speeds are: 0.83 km/h; 1.7 km/h and 5.83 km/h.

The walking-behind tractors are equipped with two types of transmissions: hydrostatic and geared. The hydrostatic transmissions equip some European manufacturers: Rapid (Swiss), Agria 2500 Hydro (Germany), Aebi CC56 and Aebi CC66 (Swiss). These transmissions can vary the speed continuously between 0 and 9 km/h in both directions.

Majority of walk-behind tractors employ gear transmissions with a minimum number of speeds of 2 (1 forward and 1 backward) and a maximum of 8 (6 forward and 2 reverse). The method of design a transmission with a required number of gears is called stepping. The gearbox must provide ratios between engine speed and wheel speed allowing the two-wheel tractor to move under difficult conditions, reach the required maximum speed and operate in the fuel-efficient ranges of the engine performance map.(Lechner & Naunheimer, 1999). Figure 6 is the result of a research over 337 types
of gear boxes and it shows the distribution of different types of gear boxes in function of number of speeds they provided (forward + reverse).

![Distribution of gear boxes](image)

Figure 6: Distribution of gear boxes

Gear boxes of type: 1+1 and 2+1 provide speeds which are suitable for walking activities such as mowing or tilling. Their speed range is: 0.28 – 3.07 km/h for the first speed, and 0.92 – 4.8 km/h for the second speed. They have the simplest design with only one PTO and they equip the cheapest two-wheel tractors. Some examples are: BCS 710 (1+1 configuration) and Honda FR750 (2+1 configuration).

When the gear boxes are of type 3+1 or 4+1, the last gear is used for transportation and it can provide a speed range between 6.8 and 15 km/h. Some of the models a outfitted with blocking differential. An example is model Agrimac 7708 RV with a 4+1
type gear box with following characteristics: forward speeds: 1.2 km/h, 2.3 km/h, 5 km/h, 15 km/h and reverse 1.8 km/h.

Geared transmissions of type 2+2, 3+3 or 4+4 are designed in such way that forward and reverse ratios have the same values. This characteristic is achieved through a bevel gear inverter mechanism. Also, these transmissions provide two PTO. An example of power tiller from this category is Volpino DF produce by Alpine Tractors (UK).

Transmissions of type 3+2, 4+2, 5+2 or 6+2 are complex. The gear ratios are different for forward and reverse direction. The configuration 6+2 offers the best coverage of the power curve. These transmissions have two PTO which offer the possibility of transportation in both directions. Most of them are equipped with a locking differential.

Transmissions of type 6+3 are actually transmissions of type 3+3 outfitted with belt drive system, which composed of two cogs on each shaft. In function of their diameter one can set low or high speed configuration.

![Bar chart showing frequency of the number of forward speeds.](image)

Figure 7: Frequency of the number of forward speeds
Figure 7 shows the frequency of maximum forward gear over a collection of 337 geared transmissions. It can be seen that the transmissions with 4 forward gears are encountered most frequently.

The same type of statistical analysis demonstrates that transmissions with 2 reverse speeds are most frequently encountered (see Figure 8).

Figure 8: Frequency of the number of reverse speeds

![Bar chart showing frequency of reverse speeds](chart1.png)

Figure 9: Stepping methods and their distribution

Figure 9 shows the result of a research over 54 models with at least 4 forward gears, out of the 341 types of transmissions, from the point of view of gear stepping em-
ployed. There are three types of stepping methods: *arithmetic*, *progressive* and *geometric*. As can be seen the most commonly used is the progressive method.

All tractors are outfitted with one constant rpm PTO, while some models have a second PTO the rotation of which is synchronous with the wheels. Figure 10 depicts the PTO rpm distribution of 118 walk-behind tractors for which this type of data was available. It shows that the most probable speed range is between 901 and 1000 rpm.

Figure 10: Rpm range distribution of the constant rpm PTO

Figure 11 depicts the maximum speed distribution. The graph was generated based of maximum speed reached in the highest gear. For this distribution there were investigated 81 tractors. The highest speed is 21.4 km/h and it is reached by Valpada 3000 in the forth speed.
Figure 11: Maximum speed in the highest gear

No. of tractors

Max. speed [km/h]

Transmission Type

Engine hp

<table>
<thead>
<tr>
<th>Max. speed [km/h]</th>
<th>No. of tractors</th>
</tr>
</thead>
<tbody>
<tr>
<td>max. 5</td>
<td>26</td>
</tr>
<tr>
<td>5.1 - 10</td>
<td>9</td>
</tr>
<tr>
<td>10.1 - 15</td>
<td>34</td>
</tr>
<tr>
<td>15.1 - 22</td>
<td>12</td>
</tr>
</tbody>
</table>

Total of 81 tractors

No. of tractors

<table>
<thead>
<tr>
<th>Engine hp</th>
<th>Transmission Type</th>
</tr>
</thead>
<tbody>
<tr>
<td>4</td>
<td>2+1</td>
</tr>
<tr>
<td>5.5</td>
<td>3+1</td>
</tr>
<tr>
<td>6.5</td>
<td>4+1</td>
</tr>
<tr>
<td>7</td>
<td></td>
</tr>
<tr>
<td>8</td>
<td></td>
</tr>
<tr>
<td>9</td>
<td></td>
</tr>
<tr>
<td>10</td>
<td></td>
</tr>
<tr>
<td>11</td>
<td></td>
</tr>
<tr>
<td>12.5</td>
<td></td>
</tr>
<tr>
<td>13.5</td>
<td></td>
</tr>
</tbody>
</table>

Figure 12: Number of combinations between the engine HP and transmission type
The figure above shows the correlation between the number of gears and the engine power. According to Figure 12-a, two-wheel tractors with 2+1 and 3+1 gears are equipped with engines ranging between 4 and 6.5 HP and 5.5 – 9 HP respectively, with the exception of Aebi AM9 (9.5 HP engine) and AM41 (14 HP engine). Engines associated with transmission with 4+1 gears usually are in the range of 7 - 11 HP. Figure 12-b shows the distribution of tractors with transmissions that have the same number of gears for forward and reverse; most of them are outfitted with engines with power of 3.8 – 9 HP. Exceptions are: for 2+2 transmission is Agria 2500 hydro (13 HP engine), for 3+3 transmission are BCS 852 and Alpine Castro DF (13 HP engine). Interesting is also that walk-behind tractors with 5.5 HP engines and 1+1 transmission are encountered in 13 different models. Figure 12-c shows that most of these tractors have 4+2 gears and 12 HP engines. Also, tractors with 4+3 gears are furnished with a 10 HP engines. Two wheel-tractors with transmission with 6+2 are outfitted with engines ranging between 5 and 18 HP.

![Bar chart showing weight distribution of 269 tractors](chart.png)

Figure 13: Weight variation over an assortment of 269 tractors.

Figure 13 shows the weight distribution of a number of tractors. As it is illustrated, the majority of have weight of maximum 200 kg.
CHAPTER 3

PATENT, PRODUCT AND LITERATURE REVIEW

Since the inception of the two-wheel tractors until today, there were attempts to improve its design, and numerous research paper and particularly patent descriptions are available. Their main objectives were to improve efficiency and reliability but also to adapt their design in such way that with the same walk-behind tractor one can perform as multiple tasks.

There are hundreds of patents related to two-wheel tractors, each of them proposing improvements for some components, or for the whole systems. Relevant to the purpose of this thesis research are the patents which deal with reconfigurable or modular garden tractors and with gear transmissions.

Walk-Behind Tractors Patents

Figure 14 shows the assembly of a convertible garden machine (patent US4237983) capable of operating as a tiller or cultivator. The machine includes a frame 10 which acts as a support for a gasoline engine 11, power transmission 12, chain case 13 and handle 14 that can be revolved from front to rear side. Also, frame 10 accommodates a drive axle 16 which supports wheel 17 and 18. Axle 20 is linked to chain case 13. On axle 20 there are a number of tines 22. Hood 25 is fastened to frame 10, and serves to support safety flap 27. A depth cut adjustment level 30 is connected to arm 31.
Attached to mounting members 14 is a U-shaped handle 34 fastened with a screw 35. The handle can be adjusted using: an extension 36, solid line position 37 and a clamp 39. Position dotted 38 shows the configuration in cultivator mode.

Figure 14: View of a tiller and cultivator assembly (Allen, 1980)

The power from engine 11 is transferred to the output shaft 42 via transmission 12 and output pulley 50 (Figure 15). The rotation of pulley 50 is transmitted to driven pulley 52, mounted on shaft 54, using the belt 51. Driven gear 56 is secured on shaft 54. Wheel 18 has a ring gear 60 around its inner rim and is in permanent engagement with wheel drive pinion 62 fixed on wheel input shaft 64. Driven gear 66 is mounted on shaft 64 and in continuous meshing with drive gear 56. Also on shaft 64 there is mounted
sprocket 70 which cooperates with chain 72 to create a driving connection to sprocket 74.
Sprocket 74 is mounted on shaft 20.

Figure 16 shows the power transmission assembly, which includes power take-off assemblies 100 and 102 between the input shaft 42 and output drive 108. Each assembly 100 and 102 has a clutch 104, 106 which is selectively operable independently of other.

Figure 15: Schematic view of the power drive train (Allen, 1980)
In tilling configuration pinion 90 receives power from the engine 11, the two bevel gears 110 and 112 which are supported by bushing 114 and 116 are driven in opposite directions. The clutch control level may be manually rotated using the wire control 120, connected to operating level 122 which is mounted on handle 34, to impose a desired counterclockwise drive to wheel 17, 18 and tines 22.

![Cross-section of the transmission](image)

Figure 16: Cross-section though the transmission (Allen, 1980)

In cultivator mode, the handle is set in position 38 the wire control 120 is retracted using operating level 122. Wheels 17, 18 and tines 22 have a clockwise rotation. The transmission is built in such way that it does not allow bevel gears 110 and 112 to engage simultaneously to the output shaft 108.
Figure 17 shows another design of a tiller (Enters & Bacon, 1981). The garden machine includes: a frame supported by a pair of wheels 14, an engine and a guiding handle 24. The frame also supports a plurality of tilling tines 16 fixed on driven shaft 18. The operator is protected from tines using a housing 20. The engine is drivingly connected to the tiller tines 16 on the tiller shaft 18 through a chain drive 79 to a drive pulley 80. The engine has a horizontal crankshaft 82 and a horizontal camshaft 84 and one end of both of them extends outwardly for supporting pulleys. As shown in figure 10 drive pulley 80 is driven by a belt 86 which in turn driven by a pulley 88 supported by crankshaft 82. A tensioning pulley 90 controls the engagement of drive pulley 80 with belt 86.

Figure 17: View of the rear tiller components (Enters & Bacon, 1981)
The drive wheels are mounted on an axle 100 and are driven by a drive chain 102. The chain 102 is in turn driven by a traction drive pulley 104 driven by the engine through a belt 106 (Figure 18 a). The engine crankshaft 82 also accommodates a reverse driving pulley 108 spaced inwardly from pulley 88 (Figure 18 b). The belt 106 is reeved over the wheel driving pulley 104 and is supported to selectively contact the forward and reverse driving pulley 108 and 110. It should be noted that crankshaft 82 rotates at a speed twice that of camshaft 84, therefore the two-wheel drive tractor is driven in reverse at twice the speed as it is driven in forward.

Figure 18: Forward and reverse configuration (Enters & Bacon, 1981)

The selective and alternative engagement of the pulleys 108 or 110 is provided by a tensioning pulley 116, supported by the bracket 118.

Another design of a two-wheel tractor is depicted in the patent US4519459 (Figure 19). The engine drives the traction wheels through a transmission and it also, drives the shaft with tines through another transmission. The second transmission allows the shaft with tines to be rotated in the same or opposite direction of the wheels. It has also a neutral position.
Figure 19: Power drive rear tiller with reversing gear transmission for tines (Reaume, 1985)

Figure 19 shows the general assembly of the tiller, which includes: traction wheels 11, tines 12 and an engine 13 with a drive shaft 20 which is the input for the transmission 21. Traction wheel transmission 23 is connected to the transmission 21 through a belt transmission 22. The tine transmission 24 is connected to the transmission 21 through shaft 28. A conventional differential 25 translates axial rotation of shaft 26, from the tine transmission, to the tined shaft 27. Except the tine transmission assembly 24 all other mechanisms may be considered conventional for this type of garden machine.

The output from the engine shaft 20 and the output from the transmission 21 for tines 12 enter in transmission 24 via shaft 28. Figure 20 shows cross sections through the tine transmission. A gear 29 is slidable on the shaft 28. A shift spool 30 is integral with gear 29 and is part of a mechanism by which the gear 29 is positioned. The gear 31 is not shiftable and it is provided with three pins 32 fixed therein with corresponding holes 33.
from gear 29. When gear 31 is connected with gear 29, the output shaft 26 which is dri-
vingly connected within the gear extension 34 will be rotated with gear 31.

Figure 20: Cross section trough tine transmission (Reaume, 1985)

For the reverse rotation (dotted position) the rpm from gear 29 is transferred to a
gear cluster comprised of gears 35r (right) and 35l (left) mounted on an axle 36. Mounted
under the gear 35r is a gear idler 37 fixed on an axle 38 mounted within bracket 39. The
gear idler 37 meshes with gear 35r and with the output gear 31. Gear 35r does not mesh
with gear 31.

Kesting proposed another design of a garden machine (Figure 21).
The main improvement of its design consists in transmission 50 which protrudes through frame P99. A reversing clutch shifter 58 projects a power input shaft 56 to a larger diameter sprocket P138 meshing with a roller chain 140. A smaller diameter sprocket receives chain P140 and concentrically supports sprocket P142 on a shaft not shown. Another roller chain P144 meshes with sprocket P142 and a smaller sprocket P146 mounted on a shaft which also supports stepped pulley P148. A V-belt P150 joins pulley P148 with a smaller pulley P152 through a clutch pulley P154 on an engine not shown. Figure 21 shows also the handle P156 with controls P158, P160 and P162.
Figure 22: Main transmission (Kesting, 1996)

Figure 22 shows the main transmission of the garden machine. Inlet shaft 56 is attached to an external spline 72 through a dampening spring 70 and lock with a snap ring 73. Shaft 56 also accommodates a jawed sleeve 74 which supports a right jawed sprocket 68 and press fit a collar 60. Sleeve 74 is attached to transmission female half housing through a bushing assembly consists in parts 62, 64 and 66. The other end of shaft 74 is supported by the housing 52 through shaft 76 and locked with snap ring 77 and bushing assembly of parts 82 and 84. Shaft 74 is permanently engaged with jawed sprocket 78.

Housing 52 also supports outlet shaft 102 through a bushing assembly of parts 86, 88 and 90. Another bushing assembly of parts 94, 98 and 100 accommodates shaft 102 in housing 54. On shaft 102 there is pressed fit the sprocket 92. Shaft 104 is supported in housing 52 and 54 by bushing assemblies of part 126, 128, 130, 114, 116 and 118. Counter
rotating spur gear 122 is pressed fit on shaft 104. In the same manner sprocket 124 and spur gear 120 are fixed on outlet shaft 106. Gear 120 meshes with a counter rotating spur gear 122. Two bushing assemblies of parts 108, 110, 112, 132, 134 and 136 accommodate shaft 106 in housing 52 and 54. A roller chain 80 meshes with sprockets 68, 78, 92 and 124.

The power from the engine goes in transmission 50 through pulley P152 after is reduced 10 times. Inlet shaft 56 takes the rpm from pulley P152 and drives sleeve 74 through spring 70 and spline 72. Sleeve 74 turns freely between sprocket 78 and 68 in the neutral position of shifter assembly 58. When sleeve 74 engages sprocket 78, chain 80 drives sprockets 92 and 124 in the same direction and sprocket 68 in reverse. When sleeve 74 and sprockets 68 are engaged, chain 80 drives sprockets 124, 78, and 92 in the opposite direction. In summary shafts 102 and 106 always rotate in the same direction and opposite to the direction of shaft 104.

Figure 23 shows the transmission 49, a second version of the transmission 50.

Figure 23: Second version of the transmission (Kesting, 1996)
For this configuration chain 80 is separated, reversed and reassembled to form a new chain 81 and sprocket 124 is transferred from shaft 106 to shaft 104. Consequently, shaft 104 rotates in opposite direction to shaft 102. Therefore, gear 122 meshes 120 and shaft 106 spins in the same direction as shaft 102.

Figure 24: Third embodiment of the transmission (Kesting, 1996)

Figure 24 shows the transmission 51, third configuration of the transmission 50. In this case gears 120 and 122 are eliminated and pair of sprockets 123 and 121 is added. An elongated chain 79 meshes sprocket 124 and drives it counterclockwise. Then it rotates sprocket 123 cw, sprocket 121 ccw, sprocket 78 cw, sprocket 92 ccw and sprocket 68 ccw. Therefore, all three configurations of the transmission provide identical rotary outputs.

Figure 25 shows another design for a convertible two-wheel tractor which operates in at least three modes, SRT (“standard rotating tine”), CRT (“counter rotating tine”) and front tine tiller.
Figure 25: Convertible garden tiller (Roberts & Altamirano, 1998)

Figure 25 shows the tiller in SRT mode. The garden machine is includes: an engine 46, a chassis 44 supported by a pair of wheels 36 disposed on a axle 37, pulley housing 72, a drive transmission 45, handles 32, tines 38, a hood over the tines 50 and a tine transmission 40. The engine is positioned over the wheels to add traction. A belt is used to transfer the rpm from the engine to the wheel transmission 45.

Figure 26 depicts the wheel gear box. The transmission is of type worm gear. The gear change mechanism is composed of a transmission level 148 which is disposed on shaft 149 and a wheel engagement level 166 which engages an abutment 168. This abutment is received by a circular channel 170 disposed on the disk clutch 172. The clutch rotates and is slid able on axle 37. Clutch 172 has a number of engagement pins 174
which engage with holes 176 disposed on drive gear 178 as a response to the axial movement of the clutch disk. Gear 178 rotates independently from axle 37 and is permanently driven by the engine.

Figure 26: Traction transmission (Roberts & Altamirano, 1998)

Figure 27 shows the tine-shaft transmission, which is composed of: a power take-off shaft 40 linked to the transmission 46, a worm gear 80 supported by the case 66 and the transmission output shaft 47. The worm gear engages worm wheel 82 drive tine shaft 48 which rotates tines 34.
Sugimoto proposed in his patent no. 6886646 a self-propelled tiller, crawler-type carrier, which has a hydraulic CVT as a power source (Figure 28).

Figure 28: Self-propelled tiller with hydraulic CVT (Sugimoto & Sueshige, 2005)
The carrier includes: a pair of crawler belts 12, a load carrying platform 13 mounted on the frame 11, an engine 14, a hydraulic CVT 15, transmission mechanism 16, stand 17, and operating handles 18. Each crawler belt includes a drive wheel 21, a driven wheel 22 and a crawler belt 23 running between them. The crawler unit includes a number of rotary wheels, also.

Figure 29: Schematic power train of the carrier (Sugimoto & Sueshige, 2005)
Figure 29 shows the power train of the self-propelled operating machine. The power from engine is transmitted through the hydraulic CVT 15 to the transmission mechanism 16 and then to the axle 38. The power train is composed of: engine 14; hydraulic CVT 15; with the input shaft 15a and out shaft 14a; a drive spur gear 31; left and right driven spur gears 32; left and right transmission shafts 34; left and right spur gears 38 connected to the transmission shafts 34; a driven bevel gear 62 meshing with drive bevel gear 61 and a power take-off shaft 63. The drive bevel gear 61 and spur gear 31 are mounted on the output shaft 15b of the CVT. Gears 32 mesh with 31 and are connected with shaft 34 via left and right clutches 33. Shafts 34 and 38 are connected via a worm mechanism 35. At the end of shafts 38 there are mounted the drive wheels 21.

The hydraulic CVT 15 is a forward/reverse switchable mechanism which allows transmissions shifts with respect to the power of the engine 14. The behavior of the of the carrier is managed with the help of a control level 41, a transmission control arm 43 and a wire cable 43.

Each clutch 33 is a aw clutch with a plurality of clutch claws 51 located at the lower and of driven gears 31, slotting in a plurality of clutch claws 52 on the upper portion of the transmission shaft 34. A spring 53 is provided to ease the clutch claws connection. The engagement or disengagement of the clutch is controlled by the operating arms 56 which operate via wire cables 55, thereby lifting the driven spur gear 32.

A power take-off mechanism 60, composed of: drive bevel gear 61, driven bevel gear 62, and power take-off shaft 63 is integrated into transmission mechanism 16. A belt 67 running between shaft 63 and driven pulley 66 mounted to a first external load 65, use to take power from the engine to drive the external load 65 as desired.
The engine 14 has also a power take-off pulley 71 mounted on the output shaft 14a. A belt 74 running between the pulley 71 and a second external load 72 allows taking the power from the engine outside to drive the second external load 72 as desired.

Transmission Patents

Figure 30: Transaxle transmission with planetary differential (Rundle, Garden Tractor Transaxle with Planetary Differential Input, 1986)

Figure 30 depicted a top input transaxle transmission which incorporates a planetary differential design for small tractors. The transmission provides three forward and one reverse normal speed drive and two or three forward creep speeds for extreme reduction ratio.
The transmission is composed of: lower case 60, upper lid (not shown), main shaft 1 and axle shafts 9 and 10. Power from the engine is clutched and introduced via an input shaft (not shown) protruding vertically. The input shaft ends with a bevel gear 62 shown dashed and revolved 90° from true location. The bevel gear 62 meshed with another bevel gear 63 which is keyed and fixed to main shaft 1. Bevel gear 63 is common part with reverse drive sprocket 64 that drives reverse driven sprocket 65, rotatable on shaft 2 with a chain (not shown). Pinion 72 keyed and fixed on shaft 1, meshes with gear 73 rotatable integral with low pinion 71 on countershaft 2. Low pinion 71 drives low gear 70 rotatable on main shaft 1. Cluster 98 is engages by a shift fork and is rotatable and free to slide axially on main shaft 1. When is moved to the right, pinion 66 engages with internal aperture gear 89 of the gear 63 and it become locked on shaft 1. When is moved to the left pins 87 engage holes 88 in low gear 70 to be driven.

In first and third gear, pinion 66 drives first and third gear 67 rotatable on shaft 2. In second and fifth gear pinion 68 drives gear 69 rotatable on shaft 2. Axle shaft 9 passes coaxially within hollow shaft 2. Shaft 2 is supported by center wall 61 and case 60. Gear 33 is located at the end of countershaft 2. Internal ring gear 35 is secured in the case with lug 36 fitted in the aperture 54 to avoid rotation. Differential carrier 42 is fixed by three pins 12. Rotatable mounted on pins 12 and meshing with sun gear 33 and internal ring 35 are planet gears 34. Pin 13 is fixed on carrier 42 and it supports gears 39 which mesh with gears 40 mounted on the shafts 9 and 10. Axle shaft 9 and 10 are rotatable in the case and are ended with the tractor wheels.

Shift collars 74 and 75 are keyed to countershaft 2 and moved axially when they are engaged by fork shifters. When 74 is moved to the left dog 91 engages with dogs 90
of cluster 99, to obtain fourth gear. When 74 is moved to the right, pins 92 engages pins 93 of the gear 69 establishing second or fifth gear, depending of the range selection.

When collar 75 is moved to the right pins 96 connect with pins 97 of the reverse sprocket 65, obtaining the reverse speed. When collar 75 is moved to the left pins 95 slot in holes 94 of the gear 67 establishing first or third speed depending of the range of selection.

Shift forks and linkage are controlled by the movement of two gear shift levels “in line” and H-pattern. The overall speed range provided by this transmission is between 0.5 and 5 mph without redundancy. Reverse and forward speed are ready available by using only one level.

Figure 31: Five forward speeds plus reverse transmission for a garden tractor (Rundle, Transmission and PTO System for Tractors and Utility Cycles, 1987)
Figure 31 illustrates a longitudinal section through one of the gear transmissions proposed by Rundle in his patent no. 4658662. By that time 1987, there were a number of disadvantages of the transmission tractor design, some of them are: lack of safety in operation, gear changes requires the movement of two gearshift levers and the range of transmission will reduce the PTO rpm.

The transmission is mainly composed of: a clutch 55P, constant mesh gear transmission 55S and a differential 55D. Primary input shaft 40 is supported by housing 55P and is direct connected to the engine at 41. Pinion 43 is common with shaft 40 and drives spur gear 44, rotatable on shaft 10. Bevel section 68 of the gear 44 together with cone output 45 keyed to main shaft 10, and Bellville spring form the clutch friction mechanism.

A transmission output shaft 11 is supported by housing 55S. The rpm is transferred from shaft 10 to shaft 11 through a number of gears. Reverse drive sprocket 12 is locked on shaft 01 and drives sprocket 13 rotatable on shaft 11 with a chain. Third pinion 14 is fastened to shaft 10 and meshed with gear 15. Gear 15 is rotatable in countershaft cluster 6 with transferred pinion 16 on countershaft 11. Pinion 16 meshes with gear 17 rotatable in main shaft cluster with first pinion 18 on main shaft 10. Pinion 18 meshes with gear 19 rotatable on shaft 11. Second and fifth pinion 20 rolls on shaft 10 and meshes with second and fifth gear 10, locked on countershaft 11. Pinion 20 and gear 21 are slidable axially. Forth pinion 22 keyed to shaft 10 meshes with gear 23 which spins on shaft 11. Collar 24 slides on shaft 11 when is engaged by shift fork (not shown). There are 3 forks, interconnected by a single control level.
When collar 24 is moved to the left, then the dog-clutch 25 engages sprocket 13 resulting the reverse speed. When is moved to the right the dog-clutch 26 engages third gear 15 to set the third speed.

When pinion 20 is shifted to the right, the dog-clutch 30 engages pinion 22 to set up the fifth gear. When in moved to the left, dog-clutch 29 connects pinion 18 to create second speed. When gear 21 is moved to the left, dog-clutch 27 connects with gear 19 to establish first gear. When is moved to the right, dog-clutch 28 joined gear 23 and set up forth gear.

A single level controls all six speeds of the transmission. The six speeds are: first 0.4 mph, second 1.0 mph, third 2.5 mph, fourth 3.9 mph, fifth 6.3 mph and reverse 2.5 mph.

Figure 32: Three forward speeds plus reverse transmission for a garden tractor (Rundle, Transmission and PTO System for Tractors and Utility Cycles, 1987)
Figure 32 shows a simplified version of previous transmission. It delivers only three speeds forward and one reverse. As main structure is similar with previous one, that is: a clutch 55P, a speed transmission 55S and a differential 55D. Also the transmission provides two PTO of 540 rpm and 2000 rpm and a cone clutch to engage the PTO speed.

40A is the input shaft which is supported by housing 55P and receives power from the engine at 41. PTO shaft 71 passes through the hollow main shaft 10A. Pinion 43 is common part with shaft 40A and meshes with gear 44A which spins on shaft 71. Gear 44A has a bevel surface which serves as a clutch friction surface. Clutch output 60A is locked on shaft 10A and pushed to the right by spring 46. The shift fork will engage on channel 61.

A countershaft 11A is rotatable on housing 55S, also. The pinion 14 is keyed on shaft 10A and meshes with gear 16 which goes around in cluster 6 with transfer pinion 16 on countershaft 11A. Pinion 16 meshes with gear 17 which revolves on shaft 10A. Pinion 20 which pins on sleeve 91 on shaft 10A meshes with gear 21 keyed to shaft 11A. Both 20 and 21 are slidable axially.

A reverse gear 76 spins on a lay shaft (not shown). Pinion 22A made part with shaft 10A meshes with gear 76 which meshes with reverse gear 23A which goes around shaft 11A.

When pinion 20 moves to the right, the dog-clutch 30 connects with reverse pinion 22A and delivers the third speed drive. When is moved to the left, the dog-clutch 29 engages the transfer pinion 17 and set up the first drive speed.
When gear 21 is slide to the left, the dog-clutch 27 connects with pinion 16 and produces the second drive speed. When is moved to the right, the dog-clutch 30 slots in reverse gear 23A to establish the reverse speed.

A fast pinion 69 is keyed on shaft 40A meshes with gear 70 which spins on shaft 71. A clutch 75 is locked on gear 70 and has a beveled face 66. A clutch output 63 is keyed and slidable axially on shaft 71. A clutch fork engages the wheel 63 in groove 62.

When clutch output 63 is forced to the right the fast PTO is established. When is forced to the left the slow PTO drive is set up.

Figure 33: Modular transmission, 12 forward and 4 reverse speeds (Shirley, Huckler, & Eckhardt, 1993)
Figure 33 shows a modular transmission with 12 forward and 4 reverse speeds. It is mainly composed of: a speed gear box 10, a gear range module 12, a differential 14 and a front wheel drive gearbox 16. The whole assembly is built in such way that is a particular configuration is not desired, it allows some modules to be removed.

S1 is supported by the speed gear box housing and in line with shaft S3 which belongs to range gearbox module 12. Right end of shaft S1 is rotatably mounted inside the left end of shaft S3. Shaft S1, also, accommodates a number of gears and clutches.

The speed gear box housing holds shaft S2, too. Shaft S2 accommodates a number of gears. The idler gear GIR is fixed on shaft SI and meshes continuously with gears G2R and G1R.

In addition to shaft S3, the differential case supports shaft S4. Gears G4D, G4C, G4B and G4A are rotatably mounted on shaft S4.

Module 16 is composed of shafts SF1 and SF2, a number of gears and a friction clutch.

The input is at the left of shaft S1, the operator can select one of the clutches C1, C2, C3 or C4 to engage in reverse, first, second or third speed, the S3 will be the output shaft. The operator can also select one of the clutches CD, CC, CB or CA, so that the output will be the differential. If the front wheel module is mounted the output can be gear G40. If clutch CF is engaged power is delivered to the front wheel.

In the above figure is depicted a complex version of previous transmission. An intermediate gear box module has been added in between speed gearbox module 10 and range gearbox module 12. A supplemental module 13 was also added. In this configuration the new transmission can deliver 18 forward and 6 reverse speeds.
Figure 34: Modular transmission, 18 forward and 6 reverse speeds (Shirley, Huckler, & Eckhardt, 1993)

Figure 35: Modular transmission (Bouche, 2001)
Figure 35 shows another design for a modular gear transmission is depicted. The transmission is composed of: a worm-gear module, which can be directly, connected to the engine and one or more spur gears modules, located in between the worm-gear module and the engine.

The worm-gear module comprises: a case 5, which supports an input shaft 6 which meshes with a worm wheel 7, keyed on output shaft 8, bearings 10 and shaft seal 11. At the right end, shaft 6 has external teeth 14 to accommodate the clutch 16 through internal teeth 18, when the engine is connected to the worm-gear module.

When module 22 may be insert in between engine and worm-gear module. It contain following items: a case 23, flanges 24 and 25, an input shaft 27, a double row bearing 26 and an externally serrated clutch 28. Pinion 30 is made with shaft 27 and is meshes with spur gear 31. The spur gear 31 spins around journal 32, which is part of case 23.

A plurality of modules 22 may be inserting to obtain the desire ratio between the output rpm of the engine and the final output of the worm-gear module 4.

Glockler proposes a different design for a modular transmission in his application for patent (Figure 36). The functionality of the transmission is: clutch 21 is closed in the first operating range, consequently the output shaft E is bond to the ring 16. The power from the engine is transferred via ring 16 to the transmission stage 20 and CVT 5. The rpm of output shaft A is in function of shaft 18 and gear 9. In this arrangement the stress on traction mechanism box 6 is reduced

In the second configuration, the coupling 21 is deactivated and the second coupling 22 is closed. In this case the power from the engine travels via gear 9, transmission 19 and CVT 5. In this mode the load on CVT is reduced.
Product Review

First model to analyze is Agria 240 (Figure 37). It is equipped with an engine model 4-T Hirth of 6.5 HP and a 3+3 geared transmission.
Figure 38 presents schematically the 3+3 transmission. It is composed of: a number of 6 shafts, 13 gears and a worm-gear mechanism connected to the wheels. One of the features of this transmission is the placement of the reverse gear after the clutch, in this way the tractor has the same number of speeds for forward and reverse.

Figure 38: The transmission of the Agria 2400 walk-behind tractor

When the operator shifts in first gear, the power from the engine passes through the clutch, gear 1 meshes with gear 2, which engages gear 5 (section A-A). The block of gears 6-7-8 can slide on splines along shaft IV. When in the first speed the gear block is moved to the left such that gear 6 meshes with gear 9. The rpm passes through the worm-gear mechanism to the wheels.
When the second gear is employed, the configuration of gears is the same till the power reaches shaft IV. Then, the sliding gear block moves to the right, gear 8 meshes with gear 11 keyed on shaft V.

To engage the third speed, the operator slides the sliding gear block to the middle position, gear 7 engages with gear 10.

For the reverse configuration, the torque from the engine passes through gear 1 to gear 2. Sliding gear 2 is displaced until it engages with gear 3, which is solidly fixed with gear 4. Gear 4 meshes with gear 5. Then the transmission can be set up in one of the 3 speeds available.

The PTO can also be turned on or off. If the operator wants to use the PTO, gear 12 slides to the right and meshes with gear 13, keyed on shaft VI.

![MAB Talpina tractor](image)

Figure 39: MAB Talpina tractor

Figure 39 shows another walk-behind tractor which is outfitted with a transmission that delivers three speeds forward and three speeds reverse. MAB Talpina model are equipped with gasoline and diesel engines of 10 to 12 HP.
Figure 40 describes schematically the transmission of this garden machinery.

When it is set in first gear, main clutch is closed, shaft S1 rotates with the same rpm as the engine output shaft, gear 1 meshes with gear 5, clutch C1 (which is keyed to shaft S2) is engaged. Gear 6, which is fixed on shaft S2, meshes with gear 9, rotatable on shaft S3. Clutch C3 is engaged, worm-gear mechanism is activated and torque is sent to the wheels.

Second speed configuration follows the same path, but gear 7 (located on shaft S2) meshes with gear 10. Clutch C3 is disengaged.
In the third gear configuration, gear 8 which is fixed on shaft S2 meshes with gear 11, which can freely spins on shaft S3. Clutch C4 is activated and power is transmitted to the wheels through worm-gear mechanism.

For reverse configurations an inverter is activated. Shaft S1 takes the torque from the engine; gear 3 engages gear 2 which meshes with gear 4, then the power path is similar with forward speeds, clutch C1 is deactivated. MAB Talpina models are outfitted with two PTO. First PTO is activated when clutch C2 is closed. The rpm of shaft S2 is the output of this PTO. Second PTO is synchronized with shaft S3. In function of the speed settings it can have variable rpm.

![Two wheel tractor model Barbieri Ghepard](image)

Figure 41: Two wheel tractor model Barbieri Ghepard

Figure 41 shows the model Ghepard of the Italian tractor company Barbieri. This model is equipped with gasoline or diesel engines of 5 – 7 kw and the transmission has 4 speeds forward and 3 speeds in reverse.
Particular to this 4+3 transmission is the different ratios between forward and reverse speeds (Figure 42). The transmission includes: a clutch, two shafts and two hollow shafts, 10 gears some of them outfitted with dog-clutches, and a worm-gear mechanism to transfer power to the wheels.

First speed configuration is: torque from the engine is clutched and transferred to the shaft S1, gear 7 fixed on shaft S1 meshes with gear 8 which is keyed and slidable on shaft S3, gear 2 meshes with gear 1, clutch C1 is activated and the torque is delivered to the worm-gear mechanism and then to the wheels.

When the second speed is desired gear 4 located on the shaft S3 meshes with gear 3 keyed on hollow shaft S2 and then the power is transmitted to the wheels through worm-gear mechanism.
In third speed configuration, gear 6 meshes with gear 5, which is fastened on the same shaft S2 as gear 3. Power flow follows the same path like in previous setting.

When fourth gear is setup, clutch C2 is activated, and the torque is directly transferred to the worm-gear mechanism through shaft S2.

First reverse power flow path: clutch C2 is deactivated, gear 9 fixed to shaft S1 meshes with gear 10. Gear 8 slides on hollow shaft S3 and engages with gear 10. Gear S2 meshes with gear 1, clutch C1 is closed. The rpm is delivered to the wheels through worm-gear mechanism.

Second reverse gear follows almost the same path, the difference consist of the meshing between gear 4 and 3. Also clutch C1 is deactivated.

Third reverse gear is similar to the previous configuration, but is this set up gear 6 meshes with gear 5.

If the clutch C3 is closed then the PTO is activated and has the same rpm for all forward or reverse speed configurations.

Figure 43: Two wheel tractor BCS 853
Figure 43 shows one of the walk-behind tractor products of the BCS Company. This model is delivered with diesel engines (Lombardini LGA 340 or Yanmar L70AE-DE) and is equipped with transmissions of type 4+3.

Figure 44 depicts an exploded view of the transmission. The transmission is mainly composed of: two shafts, two hollow shafts, seven gears and three sliding mechanism used to change the speeds or to engage the PTO.

First speed configuration is as follows: shaft 18, which is the input shaft of the transmission, meshes with gear 36, which is locked to gear 23. Gear 23 is rotatable fixed on sliding gear block 37. The small gear from the sliding gear block 37 engages gear 20, which is rotatable on shaft 33. Gear 20 clutches with sliding gear block 21 which is fastened to shaft 33. The torque is transferred to the worm-gear mechanism.

The power for the second speed has the following path inside the transmission: shaft 18 meshes with gear 36, which is attached to sliding gear block shaft 37 through gear 36. The largest gear from sliding gear block 37 engages with smaller gear of sliding gear block 21. Sliding gear block 21 is connected to shaft 33. Worm-gear mechanism transmits the torque to the tractor axle wheel.

Third speed follows almost the same path. The difference is that the middle gear of shaft 37 now meshes with the largest gear of sliding gear block 21.

For the forth speed, the input shaft 18 is directly connected to the output shaft 33 of the transmission through sliding gear block 21. Worm-gear mechanism receives the engine’s rpm.

First reverse speed has following configuration: input shaft 18 meshes with reverse gear 31 which engages gear 23. Gear 23 is connected to sliding gear block 37. The
smallest gear of sliding gear block 37 engages gear 20, which clutches with sliding gear block 21. Sliding gear block 21 is tied to output shaft 33.

![Diagram of BCS 853 transmission](image)

**Figure 44**: An exploded view of BCS 853 transmission

Second reverse speed path is: shaft 18 meshes with gear 31, which engages gear 23, locked on sliding gear block 37. The largest gear of hub 37 meshes with the smallest gear of hub 21. Hub 21 is connected to output shaft 33.

Third reverse speed is similar to the second the only difference is that the middle gear of hollow shaft 37 meshes with the largest gear of sliding gear block 21.

The PTO can be engaged by connecting the slider 22 to the driven gear 36.
Last tractor presented in this review is Reform 746 (Figure 45). It has a gasoline engine, ACME ACT340 of 11 HP, and a transmission with four speeds forward and three in reverse.

![Walk-behind tractor Reform 746](image)

Figure 45: Walk-behind tractor Reform 746

Figure 46 presents schematically the transmission of this model, which is 4+3 type with only one PTO.

When is engaged in first forward speed, main clutch is closed, shaft S1 rotates with the same rpm as the output engine shaft. Gear 4 is fixed on shaft S1 and meshes with gear 10, slidable on shaft S3. Clutch C3 is closed. Gear 5 engages gear 1, rotatable on hollow shaft S2. When the clutch C1 is closed the torque is transferred to the wheels through the worm-gear mechanism.

When second forward speed is desired, gear 6, which is fastened on hollow shaft S4, meshes with gear 2 is fixed on shaft S2, clutch C1 is open. The rpm is transmitted to the worm-gear mechanism.
In third forward speed configuration, gear 7 meshes with gear 3, which is located on the same sliding gear block as gear 2 and fixed to hollow shaft S2.

Figure 46: Reform 746 transmission

Fourth forward speed is the direct drive speed. Clutch C2 is closed, as the result the rpm from the shaft 1 is directly transferred to shaft S2 and from here to the worm-gear mechanism and then to the wheels.

First reverse speed set up is as follows: gear 4 meshes with gear 9 rotatable on shaft S4, gear 9 also engages gear 8, clutch C3 is opened, gear 5 connects with gear 1, clutch C1 is closed. The torque is transmitted two the wheels.

For the second reverse speed, gear 6 engages gear 2, clutch C1 is open.

In the third reverse speed, gear 7 meshes with gear 3, clutch C1 is also open.
To switch on the PTO, gear 4 engages gear 10 and clutch C4 is closed.

**Literature Review**

Two wheel tractors are mostly used in small farms or in residence for gardening activities. Today manufactures provide walk-behind tractors with a large range of engine power and outfit them with a variety of implements. These configurations allow customers to perform almost all the necessary tasks for a small farm or landscaping.

In the last decades the structure of land farms has changed mostly in Europe and United States, small farms increase their agricultural land and became large or very large farms. Consequently, they need to use more powerful and complex machineries. Many small tractors companies went out of business or changed production as the result of this transformation.

Walk-behind tractors see increased use in the developing countries from Africa, Eastern Europe and Asia. International organizations, such as Food and Agriculture Organization (FAO) and United Nation Industrial Development Organization (UNIDO) support this trend with funds, conferences, and report publication.

There is a correlation between the available power per hectare and crop yield, which is about 0.4 kw to an output of 2.5 ton/ha (Giles, G. W., 1975). It is known that a healthy, well fed man can provide 0.07 - 0.1kw power work. The difference up to 0.4 kw is provided by draft animals like horses, oxen, donkeys or dromedaries. The downsize of using animal power is that these need to be feed; it was estimated that between 1/3 and 1/2 of the harvested crops is needed to feed the work animals (Crossley, 1983).
Figure 47 shows a multipurpose tractor was developed by G. Schillaci (Schillaci, G., 1990). The tractor has a main module, and by attaching other assemblies, the tractor can be transformed: a) in a low level tiller, that work under-canopies for olives or citrus trees; b) small tractor, where the rear wheels are synchronized with front wheels, 4WD and c) a transporter. To convert the tractor configuration in tiller, one needs to change front wheels with tines and rear wheels with special wheels.

The utility vehicle is equipped with a diesel engine Ruggerini type MD151 of 16HP and has following characteristics:

Table 1: Multipurpose under-canopy cultivator characteristics

<table>
<thead>
<tr>
<th></th>
<th>Tiller</th>
<th>Tractor</th>
<th>Transporter</th>
</tr>
</thead>
<tbody>
<tr>
<td>Total length [m]</td>
<td>2.55</td>
<td>2.00</td>
<td>3.80</td>
</tr>
<tr>
<td>Gear box (forward + reverse)</td>
<td>3+1</td>
<td>3+1</td>
<td>3+1</td>
</tr>
<tr>
<td>Total weight [kg]</td>
<td>472</td>
<td>410</td>
<td>710</td>
</tr>
</tbody>
</table>
CHAPTER 4

ENGINEERING DESIGN APPROACH

Objective: Design a reconfigurable, modular utility vehicle

Product Review: Identified the characteristics of present designs

Literature Review: studied some related patents and papers

Design Approach: created a new design of a reconfigurable modular utility vehicle

Optimum configuration: find the optimum configuration of the utility vehicle

Modular Gear Box: determine the characteristics of the transmissions used for reconfigurable utility vehicle

Drive Train: design the drive train of the reconfigurable utility vehicle

Ergonomics: design the controls and driver position

Research Design

Figure 48: Flowchart of the Design Approach
The main objective of engineering design is to apply the scientific and engineering knowledge to find solutions of technical problems, and then to optimize the solution within the constraints of manufacturing process.

Figure 48 shows the line diagram of the complete approach followed in this thesis. The design process is divided in three main stages: understand the problem and review products, patents and articles related to small garden machineries; apply product development strategies to find the best concept; design the best solution to the problem.

This thesis proposes a design for a modular gear transmission that transform a simple walk-behind tractor in a reconfigurable utility vehicle. All other systems, such as: drive train, driver ergonomics, chassis etc, were designed to easy allow the configuration of the entire tractor.

The digital model of the utility vehicle was designed using Pro\Engineer Wildfire. Some of the parts were verified using finite element analysis software ANSYS™. Fasteners and all other accessories were imported from vendors’ digital libraries and incorporated into assembly design.
CHAPTER 5

DESIGN METHOD

Axiomatic Design method was chosen to develop the design of the new reconfigurable, modular gear transmission and to transform a two-wheel tractor in an utility vehicle. By using this method a designer can understand how the designs for constituent systems are interrelated. Additionally, changes in design and their influence under other systems can be easily revealed.

Axiomatic Design Method

Axiomatic Design is a method used to find one or more good solutions to a design problem. The method was developed by Nam P. Suh at MIT. His aim was “to find a set of fundamentals laws or principles for engineering design and use them as a basis for a rigorous theory of design” (Suh, 2001).

Axiomatic design process consists of a number of steps represented by state spaces that are used to generate design models. State spaces encompasses: consumer attributes (CAs) – variables that characterize the design in the consumer domain; functional requirements (FRs) – variables that characterize the design in the functional space; design parameters (DPs) – variables that describe the design in the physical solution space; process variables (PVs) – variables that describe the design in the process (manufacturing) domain.
Axiomatic design method comprises two axioms: 1) independence axiom – “an optimal design always maintains the independence of the functional requirements of the design”; 2) information axiom – “the best design is a functionally uncoupled design that has the minimum information content” (Suh, 2001).

The mechanization of agriculture is a necessity to increase the productivity. The two-wheel drive tractor need to perform a number of tasks and to be suitable for a variety of conditions.

Consumer domain consists in all the customer requirements from a product. Most of the time, these requirements are vague. Below are described some of the important characteristics a agricultural machinery should have.

First of all, tractor should perform the cultivation using different plough types, such as: by moldboard; disc; chisel or tine. It should be design to accommodate all this types of plough. Some of them require a simple fastened connection to the main unit, for instance the moldboard plough. Others such are tines require a PTO connection, increasing the complexity of mechanical transmission.

Other operations that need to be achieved with a tractor are planting; weeding, spraying and harvesting.

Planting is usually hand-based operation and depends of what type of seed is planted. Small seeds can be broadcast on a tilled surface, and covered over with a no-till system attached to the tractor (Okigbo, 1981). Large seeds require holes that can be dig with a special implement connected to the tractor. Often a mulching operation is performed at the end.
Weeding is a great challenge for small farms due to rapid growth not only of plants crop but also of weeds, which chock the former unless is controlled. This operation is suitable for mechanization.

Spraying of insecticides, fungicides, molluscicides etc., can be easily performed with a knapsack-sprayer type mounted on the tractor. There are two types of spraying: conventional, which requires a large quantity of water, and spinning discs, which used a smaller amount of water than former method.

Harvesting often involves the separation of food portion from vegetation portion. This activity should be executed either in the field or elsewhere. The vegetation part of the plant can also be prepared for other proposes such as: bedding, building, fuel etc.

Conditions in which the walk-behind tractor should perform are of three types: physical, geographical or technical and social.

Physical conditions encountered by agricultural machinery are: soil properties; trash; rocks, roots and stumps; dust; heat and altitude; field size and access; slopes; ridges and terraces.

Geographical conditions are related to the region and infrastructure development of the country where the tractor will be used. Also, weather type is a factor that needs to be taken in account in tractor design.

Possible technical and social conditions encountered by a two-wheel drive tractor are: operator misuse; servicing facilities and spares availability and social acceptability. Experience with small-scale machineries in Africa has shown that in difficult operating conditions, simple and robust tractors are most successfully (Crossley, 1983).
After reviewing the tasks a tractor needs to do, it can be concluded that the minimum requirements for a modular transmission box, in each configuration are:

1) tiller: 1 forward speeds and 1 reverse speed with the possibility of inversion;
2) front wheel drive tractor: 3 forward speeds plus one reverse;
3) rear wheel drive tractor: 3 forward speeds plus one reverse.

Axiomatic Design method was chosen to develop the design of the new reconfigurable, modular gear transmission and to transform a two-wheel tractor in an utility vehicle.

Figure 49 shows the functional requirements (FRs) for the modular transmission. The most general functional description appears in the top and is labeled “Reconfigurable tractor”. At the next lower levels the functions are broken up by employing the strategy of functional decomposition.

Figure 50 shows the hierarchy of the design parameters (DPs) related to the design requirements from previous figure. Both diagrams are built in the same time level by level. Functional requirements cannot be decomposed until one decides the corresponding design parameter. To keep design simple is important to identify the most important functional requirements at each level of the tree; otherwise the design process becomes too complex to manage. Also there can be multiple design solutions that satisfy the functional requirements.

Next step is to use matrix algebra to develop a reliable design. The solution can be found by generating the design matrix from below equation, (Suh, 2001):

\[ \{FR\} = [A] \cdot \{DP\} \]  

(1)
Figure 49: Hierarchical display of functional requirements the modular utility vehicle
Figure 50: Hierarchical display of design parameters for modular utility vehicle
Inside matrix A, “each element \( A_{ij} \) represents the change in the \( i^{th} \) functional requirements due to the value of the \( j^{th} \) design parameters”, (Dieter & Schmidt, 2009). This is a theoretical formulation and there is no expectation that a specific value exists for any \( A_{ij} \) term. This formulation is powerful because it brings insights to the design problem, even when it is analyzed using symbols and not values.

Figure 51 shows the Axiomatic Design equation with functional requirements and design parameters integrated. Design matrix A was also filled in correspondence with axiomatic design rules. As can be seen there are more than one design parameter necessarily to accomplish a specified functional requirement, such a design matrix is called coupled.

The strength of the Axiomatic Design consists of: its mathematical model with axioms, theories and corollaries; method to relate functional requirements to design parameters; decomposing the design process and base comparing alternative designs.

The weaknesses of the Axiomatic design process are: the acceptance of the axioms, that are true, especially the independence axiom and the procedure needs to be repeated for each new design.
$$\begin{align*}
\{ FR_1 : & \text{Forward – reverse switcher} \} = \begin{bmatrix}
1 & 1 & 1 & 1 & 1 & 0 & 0 & 1 & 1
\end{bmatrix}, \\
FR_2 : & \text{Forward speed I} \\
FR_3 : & \text{Forward speed II} \\
FR_4 : & \text{Forward speed III} \\
FR_5 : & \text{Forward speed IV} \\
FR_6 : & \text{Forward speed V} \\
FR_7 : & \text{Reverse speed I} \\
FR_8 : & \text{Reverse speed II} \\
FR_9 : & \text{Reverse speed III} \\
FR_{10} : & \text{PTO rear} \\
FR_{11} : & \text{PTO front}
\end{align*}$$

\begin{align*}
\{ DP_1 : & \text{Inverter} \\
DP_2 : & \text{Gears} \\
DP_3 : & \text{Input shaft} \\
DP_4 : & \text{Output shaft} \\
DP_5 : & \text{Locking device} \\
DP_6 : & \text{Clutch} \\
DP_7 : & \text{Reverse gear} \\
DP_8 : & \text{Liaison shaft} \\
DP_9 : & \text{Engager}
\end{align*}

Figure 51: Design matrix of the modular utility vehicle
CHAPTER 6

OPTIMUM DESIGN SELECTION

This chapter explains a methodology to find an optimum solution from all potential designs. There will be analyzed all possible systems which can be integrate in the new modular gear transmission.

Drive Systems

Walk-behind tractor manufactures have adopted three types of drive systems to transmit power from engine to transmission, using either friction belt or several belts running in parallel, chains and gears.

There are two types of friction belt system drives: flat and V shape. The efficiency of a flat belt drive is around 98%, which is closed to the gear drive. For a V-belt drive the efficiency is in between 70% and 96%, (Shigley, Mischke, & Budynas, 2004).

There are a number of advantages of using belt-drive systems. First of all, by using multiple pulleys and the same belt, the total number of speeds can be doubled or tripled. Many two-wheel tractors employing this arrangement to achieve low and high speed ranges, like the DT10DE model of the DaeDong Company (see table 2).

<table>
<thead>
<tr>
<th></th>
<th>Low speeds [km/h]</th>
<th>High speeds [km/h]</th>
<th>Reverse [km/h]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Speed 1</td>
<td>1.3</td>
<td>Speed 4</td>
<td>Speed 1</td>
</tr>
<tr>
<td>Speed 2</td>
<td>1.9</td>
<td>Speed 5</td>
<td>Speed 2</td>
</tr>
<tr>
<td>Speed 3</td>
<td>2.9</td>
<td>Speed 6</td>
<td></td>
</tr>
<tr>
<td>Speed 5</td>
<td>4.5</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Speed 6</td>
<td>6.6</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Speed 7</td>
<td>10.4</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Speed 8</td>
<td>1.1</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Speed 9</td>
<td>4</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
The second advantage of using a belt system is that the engine can be transversal mounted, parallel with the wheel axle. This is a simplifying feature because the transmission consists of only spur gears and occasionally roller-chain and sprocket pairs, which are less expensive.

In addition, in some models, to further reduce cost, decoupling the engine from transmission (for short stops and during gear shifts) is done inexpensively by controlling the tension in the belt.

Another advantage of transversely mounted engines and a belt drive system is that relative larger misalignments between the engine crankshaft and the input shaft of the transmission can be compensated by the belt’s elasticity.

Yet another advantage is the possibility taking power directly from the engine output shaft, using a second belt drive system to drive front-mounted implements (see Figure 52).

Figure 52: Example of implement which takes power directly from the engine, walk-behind tractor model Huiyou KGJ
The major disadvantage of the belt-drive system is slip between the pulleys and the belt, and the creep of belt. In addition, because of the transverse axis transmission, power cannot be delivered easily to a second pair of wheels of an articulated trailer. As it was mentioned previously, front-wheel drive vehicles exhibit less off-road and hill-climbing capabilities, than rear-wheel or all wheel drive vehicles.

Chain drive system eliminates the slippage and creep, because of its constant ratio. Furthermore it has a longer life and ability to drive multiple shafts from a single source of power as it was described in Chapter 3.

In practice, the use of chains to drive both the front and rear transmissions is difficult to attain. It adds additional systems and constraints, such as: grease system, chain protection and precise alignment between engine output shaft and rear transmission input shaft over a long length and in between assembled parts.

Most of European and American manufactures of walk-behind tractors equipped their products with gear drive systems. These systems are more expensive then belt and chain systems, but they offer higher power transmission efficiency, are more reliable and they have a longer service life. All gear drive systems have splash lubrication, and their efficiency is 92 – 97 %, (Lechner & Naunheimer, 1999).

**Clutch Systems**

Tractors outfitted with belt drive system use a operator controlled tension mechanism as clutch. For chain and gear drive systems a friction clutch is used. The main advantage of this type of clutch consists in its increased durability.
Inside the transmission box, gears are engaged using a dog-clutch system, or using sliding gears.

**Power Multiplication Systems**

To the author’s knowledge, all walking tractors with transport capabilities use gear transmissions that ensure both the slow and fast speeds i.e. between one and three slow speeds for walk-behind operations, and one or two additional fast speeds for transports. The combination of both slow and fast gears in the same transmission can be uneconomical, because the owner will have to pay for the components that ensure the transport speeds, even if (s)he has no interest in such activities. Moreover, the accidental engaging of a fast gear when in a walk-behind configuration can be dangerous, even if an emergency engine stop is in use.

The mechanical transmission box usually incorporates a number of spur gear pairs and one bevel or worm-gear pair as the final ratio. Spur gear efficiency is in between 92% and 97% when are lubricated by splashing. Petry-Johnson et al show that the most important parameter which influences the efficiency of a parallel-axis gear pair is the module and less surface finishing or oil selection, (Petry-Johnson, Kahraman, Anderson, & Chase, 2008). Other parameters, such as speed or shaft misalignment slightly influence the mechanical efficiency, while manufacturing errors, lead crown, face width and face contact ratio have no influence, (Xu, Hai; 2005).

Bevel gears have lower efficiency than spur gear but they can transmit power between crossing shafts. They provide 94% and 97% efficiency. Their main disadvantage is the cost, which is higher than worm-gear reducers, especially for reduction ratios smaller than 10:1.
The main advantages of using worm-gear consist in their higher ratio and the ability to transfer power in one direction only. The efficiency of a worm-gear reducer depends mostly to its speed-reduction ratio. To avoid a decrease in efficiency units are designed with smaller gear-tooth lead angle which causes more surface contact between them (Stoeber & Schumacher, 1995). The efficiency of a worm-gear reducer is between 79% for a 300:1 ratio and 90% for a 5:1 ratio.

Bearings efficiency is direct proportional with the oil type and is affected mostly by contamination particles, such as: dust, burrs and metal fillings.

**Systems Preference**

Table 3: Summation of the main parts of the transmission box for several walk-behind tractors

<table>
<thead>
<tr>
<th>Product</th>
<th>Transmission type</th>
<th>P T O</th>
<th>Gears</th>
<th>Clutch</th>
<th>Bearings</th>
<th>Shafts</th>
<th>Shifting</th>
<th>Total</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td>Spur</td>
<td>Worm-gear</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Agria 2400</td>
<td>3+3</td>
<td>1</td>
<td>14</td>
<td>1</td>
<td>0</td>
<td>7</td>
<td>6</td>
<td>3</td>
</tr>
<tr>
<td>MAB Talpina</td>
<td>3+3</td>
<td>2</td>
<td>12</td>
<td>1</td>
<td>4</td>
<td>7</td>
<td>4</td>
<td>3</td>
</tr>
<tr>
<td>Goldoni Joly</td>
<td>3+3</td>
<td>1</td>
<td>14</td>
<td>1</td>
<td>3</td>
<td>4</td>
<td>7</td>
<td>4</td>
</tr>
<tr>
<td>Barbieri Ghepard</td>
<td>4+3</td>
<td>1</td>
<td>11</td>
<td>1</td>
<td>3</td>
<td>6</td>
<td>5</td>
<td>3</td>
</tr>
<tr>
<td>BCS 853</td>
<td>4+3</td>
<td>1</td>
<td>11</td>
<td>1</td>
<td>3</td>
<td>4</td>
<td>5</td>
<td>3</td>
</tr>
<tr>
<td>Reform 746</td>
<td>4+3</td>
<td>1</td>
<td>11</td>
<td>1</td>
<td>4</td>
<td>5</td>
<td>5</td>
<td>3</td>
</tr>
<tr>
<td>Grilo G107</td>
<td>4+3</td>
<td>1</td>
<td>11</td>
<td>1</td>
<td>3</td>
<td>5</td>
<td>4</td>
<td>4</td>
</tr>
</tbody>
</table>

Relative cost per component: spur-gear 0.15; worm-gear 0.2; clutch 0.1; bearings 0.15; shafts 0.3; shifting mechanism 0.1

From the point of view of costs and reliability, a transmission is better if it consists in as fewer systems as possible for the same number of forward and reverse speeds.
Table 3 presents an analysis of seven transmission box of 3+3 and 4+3 types. Main elements of the transmission were numbered and multiply by a coefficient representing the relative cost of the respective component. As can be seen from the transmission of type 3+3, the lowest number of integrated parts and systems has MAB Talpina. Moreover this model offers two PTOs, one constant speed and one synchronous with the wheel speed. This transmission provides about the same features as those of the Agria 2400 and Goldoni Joly walking tractors, but with fewer spur gears and shafts.

Also table 3 shows that each of the transmissions of type 4+3 has fewer elements than transmissions of type 3+3. This feature is obtained by using the same spur gear for two speed power flow configurations.

From all transmissions of type 4+3 the best results is obtained Grilo G107. This score is obtained by using a more compact design.

Table 4 presents a rough computation of efficiency for seven walk behind tractors in each speed configuration. Due to the design of the transmission boxes some of the efficiencies are equal. Model MAB Talpina is the only one that has different efficiencies for each speed. An additional efficiency calculation was made when the PTO is engaged, when the configuration with the lowest efficiency was setup. The efficiency in this configuration will be slightly lower due to additional elements.

The efficiencies were selected as following: 0.95 for spur gear, 0.94 for the worm-gear, 0.99 for the ball bearing and 0.85 for simple bearing. The values were set the same for all models.
The best efficiency is achieved by Agria 2400 in first speed, because the power flow is transmitted using fewer gears and all the shafts are supported by ball bearings. According to Table 3, this model has the most number of elements.

Goldoni Jolly appears to have the worst efficiency 0.784 which is the same for all speeds. The major reason why this transmission has the lowest efficiency from all other models is because it uses the greatest number of simple bearings, either if it uses a small number of gears to transmit power.

Table 4: Computation of the transmission efficiency for a number of two wheel tractors

<table>
<thead>
<tr>
<th>Product</th>
<th>Speeds</th>
<th>Gear pairs</th>
<th>Ball bearings</th>
<th>Simple bearing</th>
<th>Overall Efficiency</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Spur</td>
<td>Worm-gear</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Agria 2400</td>
<td>All forward</td>
<td>4</td>
<td>1</td>
<td>5</td>
<td>1</td>
</tr>
<tr>
<td></td>
<td>All reverse</td>
<td>5</td>
<td>1</td>
<td>5</td>
<td>2</td>
</tr>
<tr>
<td></td>
<td>Forward + PTO</td>
<td>5</td>
<td>1</td>
<td>7</td>
<td>1</td>
</tr>
<tr>
<td></td>
<td>Reverse + PTO</td>
<td>6</td>
<td>1</td>
<td>7</td>
<td>2</td>
</tr>
<tr>
<td>Barbieri Ghepard</td>
<td>Forward 1</td>
<td>3</td>
<td>1</td>
<td>4</td>
<td>4</td>
</tr>
<tr>
<td></td>
<td>Forward 2+3</td>
<td>4</td>
<td>1</td>
<td>4</td>
<td>5</td>
</tr>
<tr>
<td></td>
<td>Forward 4</td>
<td>2</td>
<td>1</td>
<td>4</td>
<td>4</td>
</tr>
<tr>
<td></td>
<td>Forward + PTO</td>
<td>4</td>
<td>1</td>
<td>4</td>
<td>4</td>
</tr>
<tr>
<td></td>
<td>Reverse 1</td>
<td>3</td>
<td>1</td>
<td>4</td>
<td>3</td>
</tr>
<tr>
<td></td>
<td>Reverse 2+3</td>
<td>4</td>
<td>1</td>
<td>4</td>
<td>4</td>
</tr>
<tr>
<td></td>
<td>Reverse + PTO</td>
<td>4</td>
<td>1</td>
<td>4</td>
<td>4</td>
</tr>
<tr>
<td>MAB Talpina</td>
<td>Forward 1</td>
<td>3</td>
<td>1</td>
<td>6</td>
<td>1</td>
</tr>
<tr>
<td></td>
<td>Forward 2</td>
<td>3</td>
<td>1</td>
<td>6</td>
<td>3</td>
</tr>
<tr>
<td></td>
<td>Forward 3</td>
<td>3</td>
<td>1</td>
<td>6</td>
<td>2</td>
</tr>
<tr>
<td></td>
<td>Forward + PTO</td>
<td>3</td>
<td>1</td>
<td>7</td>
<td>3</td>
</tr>
<tr>
<td></td>
<td>Reverse 1</td>
<td>3</td>
<td>1</td>
<td>6</td>
<td>2</td>
</tr>
<tr>
<td></td>
<td>Reverse 2</td>
<td>3</td>
<td>1</td>
<td>6</td>
<td>4</td>
</tr>
<tr>
<td></td>
<td>Reverse 3</td>
<td>3</td>
<td>1</td>
<td>6</td>
<td>3</td>
</tr>
<tr>
<td></td>
<td>Reverse + PTO</td>
<td>3</td>
<td>1</td>
<td>7</td>
<td>4</td>
</tr>
<tr>
<td>BCS 853</td>
<td>Forward 1</td>
<td>3</td>
<td>1</td>
<td>4</td>
<td>3</td>
</tr>
<tr>
<td></td>
<td>Forward 2+3</td>
<td>3</td>
<td>1</td>
<td>4</td>
<td>4</td>
</tr>
<tr>
<td></td>
<td>Forward 4</td>
<td>1</td>
<td>1</td>
<td>4</td>
<td>2</td>
</tr>
<tr>
<td></td>
<td>Forward + PTO</td>
<td>3</td>
<td>1</td>
<td>5</td>
<td>4</td>
</tr>
<tr>
<td></td>
<td>Reverse 1</td>
<td>3</td>
<td>1</td>
<td>4</td>
<td>3</td>
</tr>
</tbody>
</table>
In conclusion, it can be stated that for a high efficiency value the new transmission must transmit the power using a low number of gears and all the shafts should be ball bearing supported. Using many ball bearings will increase the cost of the tractor.

The new tractor is designed for developing countries, the transmission, as well tractor design should be simple and easily to maintain and change parts.

The design of the new tractor should balance between cost, complexity, maintenance and weight. It should also have an appealing design and integrate ergonomics. Additionally, the design of the new tractor should show reliable and powerful machinery.
CHAPTER 7

MODULAR UTILITY VEHICLE POWER TRAIN DESIGN

The aim of this chapter is to determine the minimum characteristics of reconfigurable utility vehicle design. Minimum characteristics are defined as the simplest systems that need to be integrated in the design.

Figure 3 shows, that the minimum power requirement for an engine is at least 6 HP. Also, the highest numbers of 6HP engines have a transmission of type 1+1 (Figure 12). Transmissions of type one speed forward one reverse are the simplest design from all other transmission tailored to the design goal.

Figure 53 shows Carpatina mower a product available Tubal S.A. in Bucharest, which incorporates both type of engine and transmission with the described characteristics. The mower consists in following systems: gasoline engine, a transmission (type one speed forward, one reverse and a neutral position), blades and handles with controls.

Figure 53: Carpatina mower
Engine

Carpatina is equipped with a four-stroke gasoline engine with counterclockwise rotation and splash lubrication. It has a centrifugal governor with balls and magneto with contacts ignition. Start is manual using a rope. Other characteristics are presented in the below table.

Table 5: AL-75 B engine characteristics [from owner’s manual]

<table>
<thead>
<tr>
<th>Characteristics</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Displacement</td>
<td>287 cm³</td>
</tr>
<tr>
<td>Power</td>
<td>4.1 kW (6 HP) at 3200 rpm</td>
</tr>
<tr>
<td>Fuel consumption</td>
<td>1.9 l/h</td>
</tr>
<tr>
<td>Fuel tank capacity</td>
<td>4 l</td>
</tr>
<tr>
<td>Weight</td>
<td>23 kg</td>
</tr>
</tbody>
</table>

MATLAB software was used to digitize the torque, power and specific fuel consumption curves of the AL-75 B engine. Appendix 1 presents the code to determine the torque, power and efficiency curves by fitting a second degree polynomial function.

Figure 54: Engine AL-75 B torque curve
Figure 54 shows the engine torque curve. Data points were extracted from the owner's manual and used to determine the regression parabola of equation (2). This equation was extensively used in the traction analysis of the proposed vehicle (Figure 61).

\[ T(x) = 0.0133123 + 0.0011406 \cdot x - 0.0000002 \cdot x^2 \] (2)

Front Transmission

Figure 55: An inside and outside views of the transmission of Carpatina mower
Figure 55 shows an inside and outside view of the transmission together with the engine. As can be seen the power flow from the engine is clutched using a friction clutch, then it is transferred to gears G1, which is part of the shaft S1.

Gear G1 meshes with gear G2, which is fastened keyed on shaft S2. Shaft S2, as well as other shafts of the transmission, is supported by the transmission case using ball bearings. On the shaft S2 there are located bevel gears G3 and G4. The two bevel gears mesh continuously with bevel gear G5. To change in forward or reverse speed the gear change handle is employed (small image). In function of the chuck 1 position, the torque is transmitted to one of the small bevel gears, the other one spinning freely on shaft S2.

To transfer the power to the PTO, chuck 2 needs to be engaged. Chuck 2 is locked on shaft S4, which at the other end has a triangular flange for easy connection with accessories of rear module.

Figure 56 shows a 120° revolved image of the Carpatina transmission. As it can be seen, the bevel gear is mounted on shaft S3 which is supported by lid 1 using ball bearings. Shaft S3 has a geared sector G6 which meshes with gear G7, locked on wheel shaft.

The final transmission ratio is 0.016 and the overall efficiency is 0.847 (calculated by the same method as transmissions from chapter 6).
Rear Transmission

The second transmission used for the modular utility vehicle design is from Opel. It is a three speed forward, one reverse manual transmission.

Figure 57 shows an isometric view of rear module transmission. The input shaft is S5, which is supported by transmission case using a ball bearing. The output shaft is S9.
Figure 57: Opel manual transmission used as the second transmission of the modular utility vehicle

Figure 58 presents a front view of the same transmission. In the first gear configuration, geared sector G13 of shaft S5 meshes with gear G8. Gears G8, G9, G10 and G11 are common parts with hallow shaft S6. Gear G9 engages gear G14 rotatable on shaft S9. Slider 1 moves to the right and engages sync ring 1 and (when rpm are equal) ring 1, which is locked on shaft S9.
Second speed setup is as following: gear G13 meshes with gear G8. Gear G10 engages gear G15. The slider geared G16 slides to the left and it connects with sync ring 2. Ring 2 is locked on shaft S9.

In the third gear, slider 1 moves to the left and connects input shaft S5 directly to the output shaft S9 through grooved ring 1. Hollow shaft S6 spins freely on shaft S7.

In reverse arrangement, gear G11 of shaft S6 meshes with gear G12, rotatable on shaft S8. Sync ring 2 moves to the right and engages with gear G12.
To maintain speed configurations during service, the gear change mechanism uses indexing bars with multiple positions.

Final transmission ratios are: 0.309 for the first speed, 0.595 for the second speed and 1:1 for the direct drive configuration. The efficiency for all speeds is approximately 0.874.

Rear Axle and Differential

![Figure 59: 3D model of Ford Mustang rear differential](image)

Figure 59 shows the Ford Mustang differential used to transmit power to the rear wheels. It is outfitted with friction plates breaks and it has a power ratio of 1:4.

Ergonomics

Figure 60 presents the manikin used in the design process for power train characteristics. It was modeled with the respect of physiological dimensions between the principal joints of the human body at a height of 1.80 m.
The weight of the digital model was imposed to be 95 kg. Also the center of gravity was set accordingly with human body.

Additional, the manikin was used for ergonomics. Steering wheel, controls were positioned in accordance with the driver location for ease access. All positions of the handle were designed in such way that their placement requires minimum effort from the driver. The location of chair is adjustable and it keeps the same distance from the steering wheel in all tractor configurations. ROPS frame was dimensioned to provide safe to the driver if the tractor rolls to one side.

All other systems were designed to allow an easy access and exploitation.

Power Train Characteristics

Using the mechanical characteristics of engine, Carpatina transmission, Opel transmission and Ford Mustang differential, the power train properties can be calculated.

Figure 61 shows the traction diagram for the three forward and reverse speeds of the utility vehicle. For a real traction computation the efficiency in each gear was set as following: 0.81 for speed 1, 0.84 for speed 2 and 0.86 for speed 3.
Figure 61: Traction diagram of the new utility vehicle
CHAPTER 8

RECONFIGURABLE, MODULAR UTILITY VEHICLE

This chapter will provide a design example for reconfigurable, modular garden machinery. The design was made using following software: Pro\Engineer Wildfire 4.0, Pro\Mechanica and Pro\Mechanism. The tractor will be presented in all configurations as a front and rear 2WD tractor.

Walk-Behind Tractor - Tiller

Figure 62: Two-wheel tractor – tiller configuration
Figure 62 shows the first arrangement (tiller) of the reconfigurable garden machine. The torque from the engine is transmitted to the gear box through a gear drive system and a friction clutch. The final ratio of the transmission is 0.016. The wheel axle is incorporated into the transmission.

The support structure is made out of steel type 1040. The support is welded structure and it is fastened directly on the transmission box using M10 bolts with corresponding nuts and split locking washers, (Figure 62 a). Engine support (yellow) is attached to the transmission structure and locked using bolts. Engine is fastened to the engine support and transmission using M10 bolts.

The tiller is guided using a handle which supports all the controls. Figure 62 b) shows the system used to lock the handle to the transmission structure. The handle position can be adjusted by matching the teeth of complementary parts (Figure 62, b and c). Figure 62 d shows the controls mounting on the handle. The control levers were designed in such way that they can be actuated using either hand or foot.

Figure 62 e shows other two welded structures mounted on the transmission box and support. They were be use to add more strength to whole assembly, especially when it will be use as a utility vehicle.

The tiller is outfitted with agricultural tires with following size: 8×16×4.

**Walk-Behind Tractor – Mower**

Figure 63 depicts the second configuration of the garden machine. In this setup the walk-behind tractor can be used as a mower or sweeper.
For this arrangement the gear change handle of the transmission was rotated 180°, as well as the handle. Additionally the handle was locked in the upper section of the transmission support. Controls maintain their position relative to the human body.

Figure 63: Two-wheel tractor rear configuration

Rear Drive Utility Vehicle, Long Configuration

Figure 64 shows the reconfigurable utility vehicle outfitted with a cargo space. For this setup the front wheels were disassembled. All other parts were not changed.

To convert the walk-behind tractor from previous configurations following systems were added: front wheel mechanisms, steering wheel assembly, the Opel transmission, a ROPS frame, a rear train support and cargo.
Figure 64: Rear drive utility vehicle with cargo - long configuration

Figure 65: Steering mechanism and front axle arrangement of the vehicle
Figure 65 shows the front systems of the utility vehicle. As can be seen the assembly consists of: wheels which are supported by a sliding gear block and a welded structure (Figure 65 a); a rectangular frame which is connected to engine support via a hallow shaft (Figure 65 b) and a directional trapeze linked to the steering wheel mechanism via a bent rod.

The elements of the directional trapeze which counts for turning the wheels are linked together through spherical joints.

![Figure 66: Free-body-diagram (FBD) for rear wheel drive with cargo configuration](image)

Figure 66 depicts the FBD of first configuration use to determine the maximum climbing angle $\alpha$. The weight of the tractor together and driver is 4900 N (500 kg). Cargo weight was set 3000 N (200 kg). The tangential drive wheel force $F_2$ delivered by the engine is equal with 1293 N.
\[ \sum F_x = 0: F_{f_2} - W \cdot \sin(\alpha) - W_c \cdot \sin(\alpha) - R = 0 \] (3)

The maximum climbing angle is calculated in function of rolling resistance force \( R \) by using equation (6):

\[
R = W \cdot \cos(\alpha) \cdot f_R
\]

\[
F_{f_2} - W \cdot \sin(\alpha) - W_c \cdot \sin(\alpha) - W \cdot \cos(\alpha) \cdot f_R = 0
\]

\[
\sin(\alpha) \cdot (W + W_c) + W \cdot \cos(\alpha) \cdot f_R = F_2
\]

\[
\alpha = \arctan2((W + W_c), W \cdot f_R) \pm \arctan2\left(\sqrt{(f_R \cdot W)^2 + (W + W_c)^2} - F_2^2, F_2^2\right) = 6.8^\circ
\]

where: \( f_R \) is equal 0.045 and it represents the rolling resistance coefficient for bad earth tracks, (Mistchke, 1982).

The reaction force of the ground on drive wheel is calculated taking in consideration the maximum climbing angle, \( \alpha \) and using equation (6).

\[
\sum M_B = 0: N_2 \cdot 2.07 - W \cdot \cos(\alpha) \cdot 1.05 - W \cdot \sin(\alpha) \cdot 0.53 - W_c \cdot \cos(\alpha) \cdot 2.06 - W_c \cdot \sin(\alpha) \cdot 0.55 = 0
\] (4)

\[
\Rightarrow N_2 = 4971 N
\]

The value of angle is also correlated with the industry standard for weight distribution 75% on drive axle and 25% on driven axle. The reaction forces at the front and rear sides of the vehicle are: \( N_1 = 2874 N \) (36%) and \( N_2 = 4971 N \) (64%).

In order to verify is the utility vehicle can move forward, the reaction force \( N_2 \) must verify the equation (7).

\[
R + \sin(\alpha) \cdot (W + W_c) < \min(\mu \cdot N_2; F_{f_2})
\] (5)

\[ 1288 \text{ N} < \min (1293 \text{ N}, 2983 \text{ N}) \]
where: \( \mu \) is equal with 0.6, and represents the adherence coefficient bad, worn road surface (Bosch, 2007).

Figure 67: Rear drive train of the utility vehicle

Figure 67 depicts the rear drive system; the manikin was eliminated for a better visualization. It consists of a frame which supports: the Opel transmission, steering wheel mechanism, ROPS frame, chair, a connect shaft and differentia system with rear wheels.
Frame consists in two sliding structures, which are mainly made out of C channels and steel plate welded together. The sliding structures are locked together through four M16 bolts. Cylinders were welded to the structure to provide feet support.

A welded structure is fastened to the frame using M12 bolts, (Figure 67 a). This structure connects the first transmission support to the rear frame. The center of mass for the entire utility vehicle is positioned close to this structure. For this reason it was designed to undergo all forces and momentums tractor can experiment.

The steering wheel mechanism is secure to the previous welded structure. It has an output shaft which can be connected at both ends to the directional trapeze. For some vehicle configurations the steering wheel system needs to be rotated 180° to maintain the natural driving movement. The entire steering wheel mechanism was taken for a car (Dacia 1300).

Figure 67 a, b and c shows the round rubber plates used to connect the transmissions, intermediate shaft and differential. Output and input shafts are connected to the rubber plates using bolts, washers and nuts.

Figure 67 d shows the ROPS frame, which supports the chair. The chair has similar lock mechanism as the handle. Additionally, to allow changes between driver positions the tubes that sustain the chair and handle has the same angle with a horizontal plane. The ROPS frame is used to avoid the rolling of the utility vehicle in case of an accident. ROPS frame is secured to C channels and Opel transmission through bolts.

Figure 67 e depicts a special feature of the rear frame. It allows the differential assembly to be revolved 180°. This is necessary when utility vehicle is in front wheel drive configuration; so that the tractor will keep the three forward and one reverse speeds.
Cargo space was designed to offer as much storage space as possible. It is supported by rear frame through rotatable assembly and locked to the ROPS frame.

Figure 67, also shows the utility vehicle in forth configuration. In this setup the cargo device is eliminated. The empty space can be used to accommodate other devices such as a sprayer tank.

Rear Drive Utility Vehicle, Short Configuration

Figure 68: Rear driven utility vehicle – short arrangement

Figure 68 shows another configuration of the utility vehicle. The total length of the vehicle was reduced by 350 mm. This setup was possible by changing the intermediate shaft between the transmissions with a shorter one (Figure 67 a).
Figure 69 depicts the FBD of the utility vehicle when in front drive and in short configuration. Cargo weight was eliminated. The location of the center of mass is at 0.48 m from the ground.

By using the same set of equation (without $W_C$) and the same values for rolling and friction coefficients, angle becomes $12.7^\circ$.

The reaction forces are: $N_1 = 2256 \text{ N (47\%)}$ and $N_2 = 2524 \text{ N (53\%)}$.

To check if the utility vehicle can climb, equation (7) is called.

$$1292 \text{ N} < \min(1293 \text{ N}, 1514 \text{ N})$$
Figure 70: Front driven utility vehicle – long configuration

Figure 70 shows another setup of the utility vehicle. In this arrangement the driver chair is mounted on the Carpatina transmission support and the handle is fixed on the ROPS frame. Their position is adjusted to allow an ergonomic driving.

The differential assembly was rotated 180° from horizontal axes.
The steering wheel system was rotated 180° from the vertical. The vehicle directional mechanism is linked to the other end of the steering wheel assembly.

Figure 71: FBD of the front wheel drive utility vehicle in long configuration and with cargo

Figure 71 Presents the FBD of the front wheel drive utility vehicle when is setup in long configuration and with cargo. The values of weights remain the same for vehicle and cargo, but the location of the center of masses is changed.

Using the same equations as in previous setups the maximum climbing angle is 6.8°. The reactions forces are: The reaction forces are: \( N_1 = 3329 \, N \) (42%) and \( N_2 = 4516 \, N \) (58%).

To check if the utility vehicle can climb, equation (7) is employed.

\[ 1288 \, N < \min (1293 \, N, 2709 \, N) \]

If the cargo is taken off, the utility vehicle is transformed its seventh arrangement.
Front Drive Utility Vehicle, Short Configuration

Figure 72 presents the utility vehicle as a short front driven vehicle. The vehicle was shortened, by using the same small shaft as in rear drive short configuration.

All other systems remain unchanged.

Figure 72: Rear-engine tractor – short wheelbase arrangement

Figure 73 depict the FBD of this configuration. The maximum climbing angle is 12.8°. The reaction forces on the two axles are: \( N_1 = 2450 \text{ N (51%)} \) and \( N_2 = 2330 \text{ N (49%)} \).

Verify the climbing angle: \( 1292 \text{ N} < \min (1293 \text{ N}, 1470 \text{ N}) \)

95
Figure 73: FBD of the front drive utility vehicle in short wheelbase setup
CHAPTER 9

CONCLUSIONS

The market study performed part of this research shows that there is a wide variety of walk-behind tractors that are produced by numerous independent manufactures, and have very little standardization. The broad variety of walking tractors is due to the diverse combinations of engines (either diesel or gasoline, longitudinal or transverse), drive systems and transmissions (with belts, gears and roller-chains). It was also noticed that there is no standardization of the PTO rpm output, and of the attachment system dimensions and design. Implements too are not standardized, and therefore no interchangeability of implements can be expected, which reduces competition and ultimately affects the consumer.

Transmissions that ensure both walking and transport speeds tend to be complicated, and also unsafe to operate, because one of the fast speeds can be engaged accidentally while in the walk-behind mode. It was also found that for the same number of forward and reverse speeds, some transmissions appear to be simpler than others, and also have better efficiencies. To overcome some of these problems, a modularization of the transmission was proposed part of this work.

Reconfigurability can ensure an increased versatility, in that the tractor can be used either a garden machine, or as a rear- or front-engine four-wheel utility vehicle. The design of such a reconfigurable machine was developed part of this research, and is proposed in this thesis. It uses an existing 1+1 transmission of a 6 HP mower, which is
augmented with a telescopic frame, secondary transmission and rear axle with brakes. Steering is ensured through hand wheel, steering box, drag-link mechanism and Ackerman linkage of the small, steerable wheels of the vehicle.

It is acknowledged that the maximum speed of 35 km/h of the proposed design is beyond the maximum documented speed of the existing hand tractor – articulated trailer arrangements available today. It is therefore advisable to use a rear axle that has a reduction ratio greater than 4:1. This way the maximum speed of the vehicle will be reduced to safer limits, and maximum slope that it can climb will increase.

The present research can be extended and modular transmissions can be designed new, optimized for increased mechanical efficiency and reduced manufacturing and maintenance costs.


Giles, G. W., (1975), The reorientation of agricultural mechanization for the developing countries, Rome, FAO/OECD report of expert panel.


APPENDIX

AL75 –B torque curve fitted using polynomial regression code

% x = rpm
% T = torque
x = [1842 1900 2000 2100 2200 2300 2400 2500 2600 2700 2800 2900 3000
     3100 3200 3300 3386];
T = [1.37 1.382 1.4 1.42 1.44 1.457 1.469 1.477 1.48 1.48 1.471 1.455
     1.435 1.409 1.383 1.355 1.329];
n = length(x);
% m = polynomial degree
m = 2;
% matrix with the coefficient of linear equations
c = zeros(m+1);
b = zeros(m+1,1);
% help matrix - power matrix
a = zeros(m+1);
for i = 1:m+1;
    for j = 1:m+1;
        a(i,j) = i+j-2;
    end
end
t = zeros(1,n);
for i = 1:m+1;
    for j = 1:m+1;
        for w = 1:n;
            t(w) = x(w)^a(i,j);
        end
        c(i,j) = sum(t);
    end
end
c(1) = n;
s = zeros(1,n);
for j = 1:m+1;
    for u = 1:n;
        s(u) = x(u)^a(1,j)*T(u);
    end
    b(j)=sum(s);
end
d = [c,b];
% Gauss elimination
S1 = diag(ones(m+1,1));

for k1 = 2:m+1;
    for ll = 1:k1-1;
        S1(k1,ll) =-d(ll,k1)/d(ll,ll);
    end
    d = S1*d;
    S1 = diag(ones(m+1,1));
end

for k2 = 1:m+1;
\[ S_1(k_2,k_2) = \frac{1}{d(k_2,k_2)}; \]
\[ d = S_1 \times d; \]
\[ S_1 = \text{diag(ones}(m+1,1)); \]
\]
end
\]
format long
for k3 = 1:m+1;
    for l3 = (k3+1):(m+1);
        S1(k3,l3) = -d(k3,l3);
        d = S1 \times d;
        S1 = \text{diag(ones}(m+1,1));
    end
end
T1 = 0.01331233394426 + 0.001140672507418 \times x - 0.000000222278068 \times x.^2;
plot(x,T1,'-r')
title('Torque curve');
xlabel('rpm');
ylabel('kgfm');