Title: Design, Construction and Performance Evaluation of Ground Wheel Driven Wheel Barrow Boom Sprayer

By

Yonas Mulatu

A Thesis Submitted to

The department of Mechanical systems and Vehicle Engineering

School of Mechanical, Chemical and Material Engineering

A Thesis Submitted to program of Mechanical systems and Vehicle Engineering School of Mechanical, Chemical and Material Engineering in Partial Fulfillment of the Requirement for Acquired the Degree of Masters of Science in Agricultural Machinery

Office of Graduate Studies

Adama Science and Technology University

Adama, Ethiopia

March, 2018
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Major Advisor: Dr Simie Tolla (Associate prof.)

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Office of Graduate Studies
Adama Science and Technology University

Adama
March, 2018
Approval of Board of Examiners

We, the undersigned, members of the Board of Examiners of the final open defense by Yonas Mulatu have read and evaluated his/her thesis entitled “Design, Construction and Performance Evaluation of Ground Wheel Driven Wheel Barrow Boom Sprayer” and examined the candidate. This is, therefore, to certify that the thesis has been accepted in partial fulfillment of the requirement of the Degree of Master’s in MSc. Degree in Agricultural Machinery

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DECLARATION
STUDENTS DECLARATION

I hereby declare that the work which is being presented in the thesis entitled “design, construction and performance evaluation of ground wheel driven wheel barrow boom sprayer” for the partial fulfillment of the requirement for the award of MSc. Degree in Agricultural Machinery Engineering is an authentic record of my own work carried out from May 2017 to January 2018 under super vision of Simie Tolla(Dr) and Fekadu Lemessa(Dr) program of Mechanical systems and Vehicles Engineering, Adama Science and Technology University, Ethiopia

The matter embodied in this thesis has not been submitted by me and other for the award of any other degree or diploma. All relevant resources of information used in this thesis have been dully acknowledged.

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ADVISORS DECLARATION

We hereby declare that we have checked this project and in our opinion this project is satisfactory in terms of scope and quality for the award of Msc in Agricultural Machinery Engineering.

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ACKNOWLEDGEMENT

First and foremost, I give glory and honor to the Almighty God, for giving me the endurance and patience in accomplishing this piece of work.

I would like to appreciate my advisors Dr. Ing. Sime Tola and Dr. Ing. Fekadu Lemessa for their priceless and constructive comments on the thesis starting from proposal development to the final thesis preparation. I greatly acknowledge their unmatched assistance and provision of related study ideas, encouragements, insight, guidance and professional support for the improvements of this work in all directions.

I am so grateful to the Ethiopian Institute of Agricultural Research (EIAR) for making it possible for me to study here. Furthermore I would like to thank staff of Agricultural Mechanization Research Directorate especially Mr. Frew Kelemu and Melkassa Agricultural research Center.

I would like to express my gratitude to my sisters Etsegenet Mulatu and Rahel Mulatu, and my brother Yared Mulatu and to all my family members as well as to all my friends.

I would like to thank lecturers and technicians at the School of Mechanical, Chemical and Material Engineering at Adama Science and Technology University for their valuable comments and sharing their time and knowledge.
DEDICATION
I dedicate this thesis to my dearest family and my sisters Etsegenet Mulatu and Rahel Mulatu, and my brother Yared Mulatu
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ACRONYMS

A  Area of piston, cm$^2$
C$_p$  Center to center distance between sprockets in pitches
D  Nozzle discharge rate, l/min
D$_p$  Diameter of piston, cm
F  Load due to power transmission, N
F$_c$  Centrifugal force on the chain, N
F$_f$  Friction force, N
F$_p$  Force on the piston of the pump,
N  Number of nozzles
N$_1$  Number of teeth on smaller sprocket
N$_2$  Number of teeth on larger sprocket
n$_v$  Volumetric efficiency of pump% expressed as decimal fraction
P  Chain pitch
P$_i$  Chain pull (tension side), Kgf
Q$_b$  Boom requirement,
RPM  Revolution per minute of the sprocket
S  Speed, Km/hr
S$_{fl}$  Chain service factor
S$_n$  Nozzle spacing on the boom, cm
T  Input shaft torque
V  Chain speed, m/min
V$_s$  Swept volume of pump cm$^3$
ABSTRACT

Pest, insect and weed problems, in crop production serious both in rain fed and irrigated farms in Ethiopia. Farmers are forced to spray insecticides, pesticides and herbicides frequently using machines which are laborious, time consuming, and dangerous in terms of operators’ safety and health environmental pollution. The objectives of this thesis saw to design develop and evaluate manually operated boom sprayer. In search of immediate solution to the problem a prototype wheel barrow boom sprayer was designed, constructed, and the performance was evaluated both on field on laboratory. The materials used in manufacturing the sprayer were selected based on the design assumptions and calculations. The sprayer had application rate of 281.3 l/ha, effective field capacity of 0.83 ha/hr theoretical field capacity of 1.04 ha/hr. and field efficiency of 82.7% compared to the manually operated of 0.4 ha/day and 56% knapsacks sprayer the prototype sprayer had improved the effective field capacity and field efficiency. Based on the on the performance result the newly developed can cover one hectare of land with about an hour with a better spray uniformity. With all the advantages the sprayer could be further improved in terms of capacity by improving the designs of some of the parts.

Keywords: Boom sprayer, wheel barrow, wheel driven pump, design, performance evolution, effectiveness and efficiency
CHAPTER ONE

1. INTRODUCTION

1.1. Back Ground and Justification

Over 85 percent of Ethiopia’s population; about an estimate of 81 million, live in rural areas and depend on agriculture for food and other basic necessities. The country’s population is estimated to reach 130 million by 2030 (CSA, 2013). This has a serious implication on the sustainability of the natural resource base and the efforts to attain national food security given that nearly half of the current population is classified as undernourished.

Population growth and land degradation contribute most to the increasing risk of food insecurity and famine in Ethiopia. On top of these obvious factors, the average crop loss due to pests was estimated to reach between 30 and 40% annually.

Crop yield loss by major insect pests is dependent on the type of crop pest and the crop variety. According to the estimate by the Holetta Research Station (1986), the average pre-harvest loss for field crops (e.g. Cereals, Pulses and Oil Seeds) ranged between 15 to 40%, for horticultural crops (e.g. Root and Tubers, Vegetables and Fruits) between 13-29%, for coffee between 9 to 48% and for cotton between 21- 60%. The average pre harvest crop losses due to insect pests ranged between 17 and 41%.

Losses to migratory pests can be catastrophic. For example, in 1958 the desert locust caused an estimated loss of 267, 000 tons of grain in Ethiopia. This was estimated to be enough to feed 1 million people for one year (Anon, 1993).

Crop diseases caused by fungi, bacteria, viruses, and plant parasitic nematodes inflict a significant amount of losses on crops. For instance, according to the field study by Holeta Agricultural Research Station (1986), losses on field crops ranged between 32-52%. Similarly the average loss on industrial crops ranged between 22 and 44%, and on horticultural crops ranged between 35 and 62%.

However, pesticide use has enabled farmers to modify production systems and to increase crop productivity without sustaining the higher losses likely to occur from an increased
susceptibility to the damaging effect of pests. The concept of integrated pest/crop management includes a threshold concept for the application of pest control measures and reduction in the amount/frequency of pesticides applied to an economically and ecologically acceptable level. Often minor crop losses are economically acceptable; however, an increase in crop productivity without adequate crop protection is very difficult, because an increase in attainable yields is often associated with an increased vulnerability to damage indicted by pests (Cooke, 1998).

Chemical application has been very successful in pest control but must be handled properly, applied in rationed proportions and spray effectively. Specialized equipment is thus essential. In fact chemical application is the only fully mechanized farming operation. Machines previously developed for chemical application include the knapsack sprayers, the ultra-low volume sprayers and tractor boom sprayers (Liu, 2008).

Although chemical pesticide use in Ethiopia was historically low, recent developments in increased food production and expansion in floriculture industry have resulted in higher consumption of chemical pesticides (Tadesse, et al, 2008).

Recently, Ethiopia has been considered as having the largest accumulations of obsolete pesticides in the whole of Africa. It was estimated that there were 402 stores at 250 sites containing 1,500 tons of obsolete pesticides (MOARD, 2007). This estimate does not include the massive but unquantifiable amounts of pesticides soaked in soils. Nor does it include contaminated building materials, pallets, shipping containers and other miscellaneous items.

The Ethiopian Obsolete Pesticides Disposal Project, a project that mainly aimed at removing obsolete pesticides has been operational in Ethiopia for the last five years. It has been reported (MOARD, 2007) that a significant portion of the obsolete pesticides have been removed since then. However, it should be noted that as the obsolete pesticides are removed, new pesticides are imported and are possibly contributing to further accumulation.

According to the survey conducted by (Tadesse, et al, 2008) in the central rift valley of Ethiopia, 84.4% of the farmers depend only on farming as a sole livelihood, 94.3% of the farmers used pesticides as part of their agriculture input and 28.7 % of the farmers use DDT for Agriculture. The protective equipment utilization in the area was almost none; alongside which 31% of the respondents claimed illness after spraying pesticide and 14.2% indicated the
occurrence with in the family of a health related pesticide incident. The training given to farmers on pesticide issues was also very minimal which lead to low level of awareness. About 50% of the respondents used empty pesticide containers for water/food storage and about 7% of them indicated that they sell empty containers for others to use. About 31% of the respondents stored pesticides anywhere in the house and about 6% of them stored pesticides even in the kitchen.

An effective chemicals use needs a scientific and effective way of handling rationing and proper application method, aiming at eliminating pests, diseases and weeds, and ensuring stable and high yield of crops using appropriate and suitable crop protection machinery and implements is crucial. In a narrow sense, equipment items used for crop protection spray chemicals to protect crops against pests, diseases and weeds ranges from big tractor mounted sprayers to manually operated knapsack sprayers (Matthews, 2008).

1.2. Statement of The Problem

Small and medium farmers mostly use knapsack sprayers to apply pesticides (FAO, 1994). More than 5 million hand operated sprayers are sold annually in the world and most of them are sold in Southeast Asia and Africa (Matthews, 1992). The quality of a number of these sprayers, and their ability to be used to apply pesticides accurately and efficiently is of great concern due to their design and operation. The majority of the sprayers performed poorly, indicating that they are poorly designed with poor materials and mishandled by the farmers, (Mamat and Omar, 1992). It was estimated that about 50-80% of applied pesticides wasted due to poor spray machinery and inappropriate application methods (Khan, et al. 1997).

The knapsack sprayers though successful have their limitations. Apart from the human fatigue which leads to unsteady walking steps, their field capacities are small. They barely cover about 0.4 hectare per hour. Their small swath implies that a sizeable farm would take several days to cover. Moreover, maintaining a constant walking speed and constant distance between nozzle and plant tops ensures uniform distribution of spray material per unit time. Varying the walking speed or distance between the nozzle and plant tops causes uneven distribution of the spray.
The distance between the nozzle and tops of the plants should be maintained at around 30 cm (IRRI, 1988). (Garman and Navasero, 1984) reported that there was a chance of overlap or missed areas during swing of knapsack sprayers’ lance operation and the nozzle height was changed by 10% in each swing of lance. That means it is quit impossible to maintain a constant nozzle height during swing of the lance.

Climate change created an erratic weather and often it is desired to spray a large farm within hours or few days to avoid adverse weather interference. It is also often required that a large farm be covered within a short period to avoid re-emergence of weeds before crop emergence. Deployment of many human powered knapsack operators to large farms has not been successful. Large farm spraying require boom equipment with larger swath. Reduced error in swath overlaps and spraying within the shortest possible time are then assured.

Tractor boom sprayer could be a possible solution but it has become very difficult for farmers to easily engage tractors even for the more laborious jobs of tillage. The cost of tractor hire is very high and beyond the reach of the average farmer. Farmers, who could afford tractors, find it difficult to access attachment boom spraying equipment. And when they possibly do, spare parts, maintenance and calibrations still pose insurmountable problems. It is also uneconomical to deploy a tractor for small farm operations. 50 hectares is the minimum farm size for economic tractor deployment (Takeshima and Salau, 2010). Thus a gap exists between the very small scale farms suited for knapsack and ultra-low volume deployment and the tractor boom spraying suitable for large scale farming. These problems continue unless it is properly addressed. Therefore this project is to bridge this gap and ease spraying at all small and medium levels.

1.3. Objectives

1.3.1. General Objective

➢ To design, manufacture and evaluate ground wheel driven wheel barrow boom sprayer

1.3.2. Specific Objectives

➢ To design ground wheel driven wheel barrow boom sprayer
➢ To manufacture ground wheel driven wheel barrow boom sprayer
➢ To evaluate the performance of ground wheel driven wheel barrow boom sprayer
1.4. **Significance of The Study**

The expected gains from the successful development of this equipment are enormous and justify financial input and research work towards the realization of the project. These gains include:

- **Solution Provision:** Farming in our country is currently on the small and medium scales. Tractor boom sprayers are for large scale farms while human powered knapsack are for small and medium scale farms. Current practice where many human powered knapsack sprayers are deployed is saddled with attendant difficulties. This project is the much needed crop protection solution.

- **Skill, Adaptability and Availability:** The skill requirement is low, making it possible for use by virtually all cadres of farmers. The adaptability is far greater than knapsack sprayers.

- **Capacity:** The capacity weighs favorably to that of the tractor, but far greater swath than those of human powered Knapsack sprayer.

- **Efficiency and Economy of Scale:** The efficiency of boom spraying over that of single nozzle knapsack sprayers is significant.
CHAPTER TWO

2. LITERATURE REVIEW

2.1. Chemical Spraying

Liquid chemicals include fertilizers, herbicides, pesticides, and growth-regulating hormones. These may be water emulsions, solutions, or suspensions of wet table powders. Liquid pesticides may be either contact or systemic type. Contact pesticides kill insects, fungi, etc., by coming in contact. To be effective, full coverage of the target, normally achieved by smaller droplets, is necessary. Systemic pesticides are taken in by the plant and they translocate within the plant. Full coverage of the plant is not required and larger droplets that are less prone to drift are acceptable (Ajit, et al, 2012).

2.1.1. Atomization

The main objective of atomization is to increase the surface area of the liquid by breaking it into many small droplets for effective coverage of plant and soil surfaces. During atomization, energy is imparted to the liquid to break it into small droplets by overcoming surface tension, viscosity, and inertia. (Ajit, et al, 2012)

In pressure atomizers, pressure energy is used to break up a liquid jet. Pressure atomizers (but not rotary atomizers), often referred to as nozzles, produce several different spray patterns (Figure 2.1), described below.

Regular flat-fan nozzles are used for most solid applications of herbicides and for certain pesticides when it is not necessary to penetrate foliage. These nozzles produce a tapered-edge flat-fan spray that requires overlapping of pattern to obtain uniform coverage. The spray angle varies from 65° to 110° with 80° being the most common. Nozzle spacing is generally 50 cm on the boom.

Fig2.1 Tapered-edge flat-fan nozzle
The boom height varies with spray angle and the amount of overlap desired. A minimum of 50% overlap is needed for uniform coverage. The operating pressure is generally 100 to 200 KPa when applying herbicides to produce medium to coarse droplets that are not susceptible to drift. Finer droplets are produced as the pressure is increased. Some herbicides are applied at pressure of 275 to 413 KPa to generate finer droplets for maximum coverage. The LP or “low pressure” flat-fan nozzle develops normal pattern at pressures of 69 to 172 KPa. Operating at lower pressures results in larger drops and less drift (Carroll, et al, 2012).

2.1.2. Droplet Size and Size Distribution

When liquid is atomized, droplets of various sizes are formed. The spray droplets are classified by their diameters, typically measured in microns (µ). The performance and effectiveness of an atomizer depends upon the droplet size and size distribution. The area covered and the volume of liquid in individual droplets is important in achieving effective and efficient application. Smaller droplets of the same volume provide more coverage. For example, one 200 µ droplet when broken into 64 droplets of 50 µ diameter will cover four times more area than the 200 µ droplet. The droplet distribution is also important from the point of view of spray drift (Carroll, et al, 2012).

Droplet size, often expressed as Dv.5 (volume median diameter), is affected by nozzle type, spray angle, flow rate, and operating pressure. Generally, the hollow-cone nozzles produce the finest droplets and flat-spray nozzles the next-finest, while the full-cone nozzles produce the coarsest spray. The droplets become finer as the width of spray increases, due to spreading of the liquid sheet to a greater angle, which produces more fines at the edges. For a given type of nozzle the smallest capacity nozzle produces smaller droplets and vice versa. As the operating pressure increases the droplet size decreases. It is, therefore, important to realize that while increasing the application rate by increasing pressure, the droplet size would decrease and may result in higher drift. Liquid viscosity and density have very little effect on droplet size in the range commonly found in agricultural application. Increasing the surface tension increases the volume median diameter (Roger, et al, 2012).
2.1.3. Uniformity of Coverage

Liquid chemical application Sprayer performance is evaluated by the uniformity of coverage and spray patterns, droplet size and size distribution, and target deposition and drift.

The uniformity of coverage is determined by (a) the type of nozzle, (b) the nozzle spacing, (c) the boom height, and (d) the angle of the spray nozzle. As shown in Figure 1.1, the most uniform coverage is produced with a flat fan nozzle with a wide angle, with the boom height set at the minimum recommended height. Raising or lowering the boom results in over- or under-application. For good uniformity and coverage it is important to have good spray overlap between the adjacent nozzles on the boom (Dennis, et al, 2012).

The overlap is defined as the width covered by two adjacent nozzles divided by the width covered by a single nozzle, expressed in percent. The boom height can be calculated for a given amount of overlap and nozzle spacing. However, manufacturers’ recommended minimum boom height should be used because the actual spray width is somewhat less than the theoretical value as calculated by the spray angle and the boom height. Spray height in the field should be adjusted to overlap approximately 30% of each edge (Ajit, et al, 2012).

\[ \text{Overlap} = \frac{\text{Width covered by two nozzles}}{\text{Width covered by a single nozzle}} \times 100\% \]

**Fig2.2. Adjacent nozzles overlap**

2.1.4. Methods for Application of Liquid Chemicals

Liquid chemical application methods vary depending on whether they are applied pre-planting, during planting, or post-planting. Pre-planting applications generally are fertilizers and herbicides and may include subsurface or surface application. Post-planting chemical applications may include fertilizers and all types of pesticides (Ajit, et al, 2012).
2.1.5. Functional Processes of Applying Liquid Chemicals

A hydraulic sprayer consists of a tank to hold the liquid chemical, an agitation system to keep the chemical well mixed and uniform, a pump to create flow, a pressure regulator valve to control rate of flow, a series of nozzles to atomize the liquid, and miscellaneous components such as boom, shut-off valves, fittings and strainers (Roger, et al, 2012).

The main functional processes are pumping, agitation, and atomization. With positive-displacement pumps, the output is not affected by the output pressure and the flow is created by positively displacing a volume by a mechanical means such as a piston or plunger. In contrast, with a centrifugal pump, flow is created by the action of centrifugal force. The output drops as the output pressure is increased. Positive-displacement pumps found on sprayers include piston (plunger), rotary, and diaphragm types (Dennis, et al, 2012).

Piston (plunger) pumps are a kind of positive-displacement pump that is well suited for high-pressure applications such as high-pressure orchard sprayers and multipurpose sprayers designed for both high- and low-pressure spraying. They are more expensive than other types, occupy more space, and are heavy, but they are durable and can be constructed so they will handle abrasive materials without excessive wear (Dennis, et al, 2012).

The volumetric efficiency of a piston pump in good condition is generally high (90% or more), and the discharge rate is essentially a direct function of crank speed and volumetric displacement. Crank speeds on the smaller piston sprayer pumps [38 L/min and less] are mostly 400 to 600 rev/min. High-pressure piston sprayer pumps [4.1 to 5.5 MPa] are usually operated at 125 to 300 rev/min have capacities of 75 to 225 L/min. Mechanical efficiencies may range from 50% to 90%, depending on the size and condition of the pump (Carroll, et al, 2012).
2.1.5.1. Pump

A piston pump is a positive displacement machine consisting of one or more cylinders, each containing a piston or plunger. The pistons or plungers are driven through slider-crank mechanisms and a crankshaft from an external source. The capacity of a given pump is governed by the rotational speed of the crankshaft (William, et al, 2001).

Unlike a centrifugal pump, a power pump does not develop pressure; it only produces a flow of fluid. The downstream process or piping system produces a resistance to this flow, thereby generating pressure in the piping system and discharge portion of the pump. The flow fluctuates at a rate proportional to the pump speed and number of cylinders. The amplitude of the fluctuations is a function of the number of cylinders (Miller, 1987).

All piston pumps are capable of operating over a wide range of speeds, thereby making it possible to produce a variable capacity when coupled to a variable speed drive. Each pump has maximum suction and discharge pressure limits that, when combined with its maximum speed, determine the pump’s power rating. The pump can be applied to power conditions that are less than its maximum rating but at a slight decrease in mechanical efficiency (William, et al, 2001).

2.1.5.2. Slider-Crank Mechanisms

Many applications require a machine with reciprocating, linear sliding motion of a component. Engines, compressors and piston pump, which require a piston to move through a precise distance, called the stroke, as a crank continuously rotates.

Other applications such as sewing machines and power hacksaws require a similar, linear, reciprocating motion. A form of the slider-crank mechanism is used in virtually all these applications In-Line Slider-Crank Mechanism An in-line slider-crank mechanism has the crank pivot coincident with the axis of the sliding motion of the piston pin. An in-line slider-crank mechanism is illustrated in Figure 2.3. The stroke, $|\Delta R4|_{\text{max}}$, is defined as the linear distance that the sliding link exhibits between the extreme positions. Because the motion of the crank (L2) and connecting arm (L3) is symmetric about the sliding axis, the crank angle required to execute a forward stroke is the same as that for the return stroke. For this reason, the in-line slider-crank mechanism produces balanced motion (David, et al 2012).
Assuming that the crank is driven with a constant velocity source, as an electric motor, the time consumed during a forward stroke is equivalent to the time for the return stroke.

2.1.5.3. Offset Slider-Crank Mechanism

The mechanism illustrated in Fig2.4a is an offset slider-crank mechanism. With an offset slider-crank mechanism, an offset distance is introduced. This offset distance, L1, is the distance between the crank pivot and the sliding axis. With the presence of an offset, the motion of the crank and connecting arm is no longer symmetric about the sliding axis. Therefore, the crank angle required to execute the forward stroke is different from the crank angle required for the return stroke. An offset slider-crank mechanism provides a quick return when a slower working stroke is needed. In Fig2.4a, it should be noted that A, C1, and C2 are not collinear. Thus, the stroke of an offset slider-crank mechanism is always greater than twice the crank length. As the offset distance increases, the stroke also becomes larger. By inspecting Fig2.4a, the feasible range for the offset distance can be written as:

\[ L1 < L3 − L2 \]

Locating the limiting positions of the sliding link is shown in Fig2.4a. The design of a slider-crank mechanism involves determining an appropriate offset distance, L1, and the two links lengths, L2 and L3, to achieve the desired stroke, |ΔR4|_{max}, and imbalance angle, β (David H. et al 2012).
The graphical procedure to synthesize a slider-crank mechanism is as follows (David, et al. 2012):

1. Locate the axis of the pin joint on the sliding link. This joint is labeled as point C in Fig 2.4 of a.
2. Draw the extreme positions of the sliding link, separated by the stroke, $|\Delta R_4|_{\text{max}}$.
3. At one of the extreme positions, construct any line M through the sliding link pin joint, inclined at an angle $\theta_M$. This point is labeled C1 in Fig 2.4 of b.
4. At the other extreme position, draw a line N through the sliding link pin joint, labeled C2 in Fig 2.4 of b, inclined at an angle $\beta$ from line M. Note that

$$\theta_N = \theta_M - \beta$$

5. The intersection of lines M and N defines the pivot point for the crank, point A. The offset distance, $L_1$, can be scaled from the construction Fig 2.4 of b.
6. From the construction of the limiting positions, it is observed that the length between C1 and D is $2L_2$. Note that this arc, C2D, is centered at point A. Because both lines are radii of the same arc, the radius AC2 is equal to the lengths $AC_1 + C_1D$. Rearranging this relationship gives
C1D = AC2 − AC1

Substituting and rearranging, the length of the crank, L2, for this offset slider-crank mechanism can be determined as

\[ L2 = \frac{1}{2}(AC2 − AC1) \]

7. From the construction of the limiting positions, it is also observed that

\[ AC1 = L3 − L2 \]

Rearranging, the length of the coupler, L3, for this offset slider-crank mechanism is

\[ L3 = AC1 + L2 \]

2.2. Some of the Works Done on Agricultural Chemical Sprayers

Khan et al. (1997) found through investigation on spray application and safety measures in Pakistan that poor pump pressure causes bigger droplet formation, which ultimately adds to soil pollution. They also stated that non-uniform pesticide distribution results phytotoxicity (due to over dosing) and resistance (due to under dosing) of pests.

Wang, et al. (1995) carried out a laboratory experiment on spray distribution pattern uniformity for agricultural nozzles and showed that nozzle height had a strong effect on spray distribution uniformity.

Hussain, et al. (1993) conducted an experiment on the field performance of a tractor mounted boom sprayer and studied the effect of swath, nozzle height and nozzle position on spray deposit. They showed that spray deposition decreased with increasing tractor speed as well as wind speed. They also found that spray deposition on plant soil and drift were 60%, 38% and 20%, respectively at speed 2m/s and reported that deposition decreased as the height of nozzle was increased.

Since the early 1970s, a lot of efforts have been done mainly towards tractorization. It is however evident that this has not yielded the expected results for a number of reasons including (a): Lack of skilled operators and maintenance personal (b): Lack of suitable implement and spare parts (c): Farm land fragmentations and (d): Increase in the cost of tractors and implement (Gwani, 1988).
In the early 1960s, an ox-drawn ground wheel-driven piston pump sprayer was developed at the Gatooma Research station in Zimbabwe. Limited numbers were manufactured in Zimbabwe and also in South Africa, by one Henry Plenn of Nogel district. The sprayer however proved cumbersome and unmanageable in wet weather and was discarded. Instead a scotch cart was arranged to carry a human powered knapsack sprayer. Simultaneously and also in Zimbabwe, Taurus spraying systems of Harari developed an Animal – drawn Ground wheel powered boom sprayer called Pedze Nhuma (Fowler, 2000).

Anibude, et al, (2016) developed Animal Drawn Hydraulic Boom Sprayer. It consists of; 100 liters spray tank capacity, mainframe, operator seat, 3Hp petrol engine, piston pump, Boom, ten flat fan nozzles, wheel and axle shaft. The petrol engine was used as the power source for operating the piston pump during spraying and pair of bullocks was used for hauling purpose. Application rate of 260 L/ha was achieved, theoretical field capacity of 1.16 ha/hr, effective field capacity of 1.04 ha/hr and 89.6% field efficiency. Comparing the results with what was obtained using the manually operated knapsacks sprayer represents 62% and 37% increase in effective field capacity and field efficiency respectively.

Shivaraja, et al, (2014) developed wheel and pedal operated sprayer that uses reciprocating pump with an accumulator to provide a continuous flows of liquid to create necessary pressure for the spraying action. This wheel operated pesticide spray equipment consumes less time and avoids the pesticide from coming from front of the nozzles which will in contact of the person who sprays pesticides.
Dileep, et al, (2017) attempted to develop Pedal operated multipurpose bicycle with sprayer and water. The setup was portable as it is mounted on the same bicycle. The pedal operation of this cycle reduces alternate source of energy. At the same time it can be used for spraying the chemicals with the same pedal power. The time of operation depends upon the efficiency of this reciprocating pump used to lift the water. This mechanism was more reliable and useful in the remote village to get the sufficient source of power. The multi ability makes it more special. The pump setup includes a housing in which a foot pedal and drive shaft rotate an eccentric disc rotating with the drive shaft moves a connecting rod which in turn causes push rod to move linearly.

Laukik, et al, (2013) developed agricultural pesticides sprayer and weeder, the equipment operates when pushed forward. While pushing by using handles, front wheel rotates and the gear mounted at the axle of wheel starts to rotate and its rotation then transferred to the pinion through the chain drive. The rotary motion of the pinion is converted into the reciprocating motion by the single slider crank mechanism, due to this arrangement the connecting rod moves upward and downward which then reciprocate the piston of single acting reciprocating
pump mounted at the top of storage tank. During the upward motion of the connecting rod the pesticide is drawn into the pump and during the downward motion of connecting rod the pesticide is forced to the delivery valve, the delivery is connected to the pipe carrying the number of nozzles.

**Fig 2.8 CAD model of agricultural pesticide sprayer and weeder**

Sumit, et al, (2014) developed pedal operated reciprocating pesticide sprayer for agricultural and drainage line use. A reciprocating pump essentially consists of a piston or plunger which moves to and fro in a closed fitting cylinder. A cylinder was connected to suction and deliver pipes, each of which is provided with non-return or one way valve called suction and delivery valve respectively. A function of non-return or one way valve is to admit liquid in one direction only. Thus the suction valve allows the liquid only to enter the cylinder and delivery valve permits only its discharge from the cylinder. The piston or the plunger is connected to a crank by means of a connecting rod. As the crack is rotated at uniform speed by a driving the paddle of trolley the piston or the plunger is connected to a freewheel by means of a connecting rod.

**Fig 2.9 Full View of Tricycle with Pedal Operated Reciprocating Sprayer**
Also Abayineh, et al, (2016) developed an animal drawn operated by diesel engine. The chemical distribution in the field by the sprayer is regulated by nozzle spray discharge rate and walking speed of animal. The chemical sprayer is operated by a single equine animal. And spray tank and diesel engine is mounted on the sheet metal plat form, which is operated by draft power of the animal/s. The spray swath width of 7.5 m length is provided with fourteen numbers of hollow cone nozzles which are adjustable according to row spacing of crop. The wheel /tires sprayer are also adjustable according to row spacing of different crops and the unit is provided with a plastic tank of 150 liters capacity.

*Fig2.10 Developed sprayer being tested on the field and its major parts*
CHAPTER THREE

3. MATERIALS AND METHODS

3.1. Study Site Description

The machine was designed and manufactured in Melkassa Agricultural Research Center. Both the laboratory and field testing were conducted in the research center. Melkassa is found 115 km from Addis Ababa in the Central Valley of Ethiopia. The place is situated at an altitude of 1466 m above sea level and located at a geographical coordinates of 8° 24’ 0” N, 39° 20’ 0” E Latitude and Longitude respectively. It receives 763 mm mean annual rainfall, of which 70% falls during the major cropping season: June to September. The dominant soil type in the area is sandy loam. Because of its agro-climatic condition, most varieties of maize crop grow well in the area.

3.2. Materials

Table 3.1 List of materials used

<table>
<thead>
<tr>
<th>No</th>
<th>Part Name</th>
<th>Dimensions</th>
<th>Material used</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>structural frame</td>
<td>$\frac{3}{4}$in X 164in, 2 piece</td>
<td>Round steel pipe</td>
</tr>
<tr>
<td>2</td>
<td>tank</td>
<td>20 lit, 150 x180 x360</td>
<td>plastic</td>
</tr>
<tr>
<td>3</td>
<td>Piston pump</td>
<td>Ø34mm, stroke length 65mm</td>
<td>Brass, plastic</td>
</tr>
<tr>
<td>4</td>
<td>shaft</td>
<td>Ø24mm, 200mm</td>
<td>Milling steel</td>
</tr>
<tr>
<td>5</td>
<td>wheel</td>
<td>Ø65 cm</td>
<td>Pneumatic spoke wheel</td>
</tr>
<tr>
<td>6</td>
<td>hose</td>
<td>5m</td>
<td>Plastic</td>
</tr>
<tr>
<td>7</td>
<td>boom</td>
<td>2, 1.5m each</td>
<td>Galvanized steel pipe</td>
</tr>
<tr>
<td>8</td>
<td>nozzle</td>
<td>04 size</td>
<td>Tee Jet flat fan</td>
</tr>
<tr>
<td>9</td>
<td>chain</td>
<td>1155.7mm, pitch 12.7</td>
<td>Treated steel</td>
</tr>
<tr>
<td>10</td>
<td>sprocket</td>
<td>Teeth no 42, Ø17.4cm</td>
<td>Treated steel</td>
</tr>
<tr>
<td>11</td>
<td>sprocket</td>
<td>Teeth no 14, Ø5.6cm</td>
<td>Treated steel</td>
</tr>
<tr>
<td>12</td>
<td>dyes</td>
<td>3liters</td>
<td>Blue and black</td>
</tr>
<tr>
<td>13</td>
<td>Measuring tap</td>
<td>30m</td>
<td>Plastic</td>
</tr>
<tr>
<td>14</td>
<td>Graduated cylinder</td>
<td>100ml</td>
<td>Plastic</td>
</tr>
</tbody>
</table>
3.3. Methods

3.3.1. Design of The Sprayer Parts

The machine was designed and constructed at Melkassa Agricultural Research Center workshop. Both the laboratory and field testing was also conducted in the research center using laboratory facility at the center. Part, assembly and manufacturing drawing were produced using CATIA and the prototype was developed according to the design.

3.3.1.1. Operation of The Machine

The sprayer prototype is made basically of the main frame, spray tank, pump/prime mover, traction wheel, boom, nozzles and flexible rubber hose. The main frame is mounted on the axle shaft with single traction wheel and carries the spray tank and pump integration, and a boom assembly of 3m length with six sprayer nozzles. The spray tank is connected to the boom with the aid of distributing flexible rubber hose via the integrated piston pump. The boom frame is bolted at the front end of the main frame. The boom frame is designed in the way that the boom height could be adjusted as per the crop height between 30cm – 120cm above the ground. The chemical in the spray tank is pumped to the flexible hose by the piston pump integrated with the tank. The pump is actuated by an offset slider-crank mechanism, which gets the power from the ground wheel. During the operation the operator simply puts the boom in a horizontal position and pushes the sprayer it into the rows of the crop. While pushing using the handles, the ground wheel rotates transferring power to the attached driving sprocket which in turn drives a smaller sprocket that is attached to a shaft through the chain drive. The rotary motion of the smaller sprocket then converted into the reciprocating motion by the single slider crank mechanism, which actuates the single acting reciprocating piston of pump installed in the tank pumping the chemical to the boom.

Fig3.1. Sprayer prototype
3.3.1.2. Design of The Slider Crank Mechanism

Determination of dimensions in the slider crank mechanism

At the top dead center position the distance between the crank arm center and the top dead center is the sum of the lengths of the crank arm and the link that is A+B and at the bottom dead center the distance between the crank arm center and the bottom dead center is the difference of the lengths of the crank arm and the link that is A- B as shown in the fig3.2 Therefore, taking right angle triangle ABC (David, et.al 2012).

Taking right angle triangle ΔABC2

\[ AC2 = \sqrt{50^2 + 27^2} \]  
\[ Eq(3.1) \]

Taking right angle triangle ΔABC1

\[ AC1 = \sqrt{43.5^2 + 27^2} \]  
\[ Eq(3.2) \]

Solving the two equations

\[ AC1 = 51.19 \]
\[ AC2 = 56.82 \]
\[ L2 = \frac{1}{2}(AC2 - AC1) \]
\[ = 2.8 \text{ cm} \]
\[ L3 = AC1 + L2 \]
\[ = 54 \text{ cm} \]
3.3.1.3. Kinematics of The Slider - Crank Mechanism

Slider-Crank Mechanism a general slider-crank mechanism is uniquely defined with dimensions m, r, and L. With one degree of freedom, the motion of one link must be specified to drive the other links. Most often the crank is driven. Therefore, knowing $\theta_c$, $\theta_p$, and the position of all the links, the velocities of the other links can be determined.

Conventionally, asymmetric crank system for which the axis of rotation of the crank is perpendicular to the plane of oscillation. Such sliders are required to be operated at some prefixed level equal to offset distance; therefore, the crank center is located at an offset distance of m from the slider center line. (Fig3.3). The ratio $m/L$, where L is the pitman length.
of connecting rod, called eccentricity (ξ), plays an important role in deciding crank radius (r) and stroke length (s) (Amalendu et al, 2003).

Fig3.3 graphical representation of an offset slider crank mechanism

Instantaneous velocity \( v \) of the mechanism is calculated using the equation (Amalendu et al, 2003).

\[
V = \omega (L_{AB} \sin \theta_{AB} - \frac{L_{AB}^2 \sin \theta_{AB} \theta_{AB} - L_{OA} \cos \theta_{AB}}{2(l^2_{BC} - L_{OA})^2}) \tag{3.3}
\]

\[
9.24 \text{ rad/s} \left(0.028 \text{msin}45 - \frac{0.028^2 \sin 90}{2(54^2 - 27)^2} - \frac{27 \times 0.028 \cos 45}{(54^2 - 27)^2}\right)
\]

\[
0.095 \text{ m/s}
\]

The instantaneous acceleration \( a \) is calculated with the following equation for an accelerating system but for our case the mechanism operates with an average speed (Amalendu, et al, 2003).

Therefore:

\[
a = \frac{dv}{dt} = 0 \tag{3.4}
\]

3.3.1.4. Determination of The Forces and Torque

In analyzing the forces on each link an instant of configuration, were the crank arm is at an angle \( \Theta \) of 45° with the horizontal is taken as shown in the fig3.3. At this instant, the link will make an angle \( \beta \) of 32.4° based on the geometric and trigonometric relation. (David, 2012)
From the above assumption, the pressure on a single nozzle is taken to be 1.8 bars. Therefore, the amount of theoretical pressure require for the 6 nozzles will be 1.8 bars X 6 = 10.8 bars.

Considering equilibrium; the force needed at the piston is calculated as follows

Let F be the applied by the connecting rod B on the piston, then from the equilibrium equation

**Fig3.3. Free body diagram of the slider- crank mechanism at 45° position**

\[
\Sigma F_x = 0
\]
\[
F \sin \beta = F_N \quad \text{Eq (3.5)}
\]
\[
\Sigma F_y = 0
\]
\[
F \cos \beta = F_p \quad \text{Eq (3.6)}
\]

Where

F= the force in connecting rod B

F_p= force due to pressure

\( \beta \) = the angle between link B and the vertical, \( \beta = 32.4^0 \) at crank position of 45°

\( F_x \) = the horizontal component of the force in the connecting rod
$F_y =$ the vertical component of the force in the connecting rod

Taking the pressure required to be 10.8 bar and the measured diameter of the piston 3.4 cm

$$F_p = P \times A$$

Taking the fact that 1 bar = 10 N/cm² and base area $A = \frac{\pi d^2}{4}$

$$A = 9cm^2 \text{ and } P = 108 \frac{N}{cm^2}$$

Putting this values in the above equation

$$F_p = P \times A$$

$$= 9cm^2 \times 108 \frac{N}{cm^2}$$

$$= 980N$$

$$F = \text{Force along CR} = F = \frac{F_p}{\cos\beta}$$

$$F = \text{Force along link B} = \frac{980N}{\cos(32.4)}$$

$$= 1166.7N$$

$$F \sin(32.4) = F_N$$

$$= 1166.7N \times 0.53$$

$$= 618.35N$$

Calculating the torque at the crank arm

$$\tau = F \times R_c \sin(\beta + \theta) \quad (\text{David H. 2012})$$

Where

$$\tau = \text{The torque at the crank arm in, Nm}$$

$$F_c = \text{The force at the crank arm in, N}$$

$$R_c = \text{Crank arm radius in, m}$$
\[
\tau = 1166.7N \times 0.028m \sin(77.4) \\
= 31.69Nm
\]

Calculating the tension in the chain,

Both the small sprocket and the crank arm are on the same shaft therefore

\[\tau = T \times R_{ss} \quad (\text{David, 2012})\]

\[Eq(3.10)\]

\[R_{ss} = \text{Radius of the smaller sprocket in, m} = 0.0288m\]

\[T = \text{Torque on the sprocket in, Nm} = \tau = 31.69Nm\]

\[T = \text{the tension in the chain, in N}\]

\[R_{ss} = 0.0288 \text{ and } \tau = 31.69\]

\[T = \frac{31.69Nm}{0.0288m}\]

\[= 1156.3N\]

3.3.1.5. Analysis of The Forces on The Sprayer

The different forces acting on the machine is shown in the free body diagram in fig3.4

The different forces acting on the sprayer are

Free body diagram of the linkages

Each reaction force is described in term of its x and y components, where the direction of each component is assigned arbitrarily. For notational simplification, simple numbered indices are used for all the components.

Forces acting on the sprayer are

\[W_b = \text{weight of boom system}\]

\[W_f = \text{weight of frame}\]

\[W_c = \text{weight of chemical/ water in this case}\]
Ffr = rolling friction

Fx = push force applied by the operator in the horizontal direction

Fy = lift force applied by the operator in the vertical direction

N = the normal force at the wheel contact

Fig3.4. Free body diagram of the forces acting on the sprayer

Considering an equilibrium

\[ \Sigma F_x = 0 \]

\[ Fx - Fr = 0 \] \[ Eq(3.11) \]

\[ \Sigma F_y = 0 \]

\[ Fy + FN - (Wf + Wch + Wbs) = 0 \] \[ Eq(3.12) \]
Taking moment at the wheel contact point B with the ground

\[ 1.88Fy + 0.8Fx - (0.73Wf + 0.82Wch + 0.25Wbs) = 0 \]  \hspace{1cm} Eq(3.13)

\[ Wf = g \rho_{steel} \times Vf \]  \hspace{1cm} Eq(3.14)

\[ Vf = A \times Lf \]  \hspace{1cm} Eq(3.15)

\[ A = \frac{\pi(D1^2-D2^2)}{4} \]  \hspace{1cm} Eq(3.16)

Where

\[ D_1 = 27\text{mm} = 0.027\text{m}, \quad D_2 = 24\text{mm} = 0.024\text{m}, \quad L_f = 214.5\text{cm} = 2.15\text{m} \]

\[ D_1 = \text{External diameter of the pipe} \]

\[ D_2 = \text{Internal diameter of the pipe} \]

\[ L_f = \text{Length of the frame} \]

\[ W_f = \text{Weight of frame} \]

\[ W_{ch} = \text{Weight of chemical/water in this case} \]

\[ W_{bs} = \text{Weight of boom system} \]

Calculating for \( W_f \)

\[ A = \frac{3.14(0.027^2 - 0.024^2)}{4} \]

\[ = 0.00012m^2 \]

\[ Vf = 0.00012m^2 \times 2.15m \]

\[ = 0.000258m^3 \]

Taking \( \rho_{\text{steel}} \) to be 7850 kg/m\(^3\)

\[ Wf = \frac{10m}{s^2} \times 7850 \text{ kg/m}^3 \times 0.000258m^3 \]

\[ = 20 N \]
Considering the symmetric frame the total $W_f$

$$2 \times 20$$

$$= 40 N$$

Calculating for $W_{ch}$

$$W_f = g \rho_{water} \times V_{tank}$$

$$W_f = 10 \text{ m/s}^2 \times 1 \text{ Kg/lit} \times 20\text{lit}$$

$$= 200N$$

Calculating for $W_{bs}$

$$W_{bs} = W_{ch \text{ in the boom}} + W_{boom}$$

Calculating for $W_{ch \text{ in the boom}}$

$$W_{ch \text{ in the boom}} = g \rho_{water} \times V_{water \text{ in the boom}}$$

$$V_{water \text{ in the boom}} = A_{\text{cross section of boom}} \times L_{boom}$$

$D = 20\text{mm} = 0.02\text{m}, L_{boom} = 3\text{m}$

$$A_{boom} = \frac{\pi D^2}{4}$$

$$A_{boom} = \frac{3.14 \times 0.02^2}{4}$$

$$= 0.000314m^2$$

$$V_{water \text{ in the boom}} = 0.000314m^2 \times 3m$$

$$0.000942m^3$$

Calculating $W_{ch \text{ in the boom}}$

$$W_{ch \text{ in the boom}} = 10\text{m/s}^2 \times 0.000942m^3 \times 1000\text{kg/m}^3$$

$$= 9.42N$$
Calculating $W_{\text{boom}}$

$$W_{\text{boom}} = gV \rho_{\text{steel}}$$

$$V_b = AXL_b$$

$L_b = 3m$, $D_1 = 22mm = 0.022m$, $D_2 = 20mm = 0.02m$

$$A = \frac{\pi(D_1^2 - D_2^2)}{4}$$

$$A = \frac{3.14(0.022^2 - 0.02^2)}{4}$$

$$= 0.000066m^2$$

$$V_b = 3m \times 0.000066m^2$$

$$= 0.000197m^3$$

$$W_{\text{ch in the boom}} = 10m/s^2 \times 0.000197m^3 \times 7850 \text{ kg/m}^3$$

$$= 15.53N$$

$$W_b = W_{\text{boom}} + W_{\text{ch in the boom}}$$

$$= 15.53N + 9.42N$$

$$= 24.94N$$

Substituting the values of the weights in Eq.3.24 and Eq.3.23 gives

$$Fy + FN - 264.92N = 0$$

Eq(3.17)

$$1.88Fy + 0.8Fx - 172.8Nm = 0$$

Eq(3.18)

Analyzing the moment and the torque on the driving wheel

The torque due to the chain tension is calculated as

$$\tau_w = T \times R_{Ls}$$

Eq(3.19)

Where

$R_{Ls} = \text{Radius of large sprocket} = 8.7cm$

$T = \text{Tension in the chain} = 1056.3 \text{ N}$
\( \tau_w = \text{Torque on the driving wheel due to the chain tension} \)

\[
\tau_w = 0.087m \times 1056.3 \, N
\]

\[
= 91.89 \, Nm
\]

\( \tau_w \)

\( F_r \)

\( B = \text{wheel deflection measured from the wheel center line to the point of contact} = 0.07m \)

\[
R_r = \sqrt{(R^2 - b^2)}
\]

\[
R_r = \sqrt{(32.5^2 - 0.07^2)}
\]

\[
= 0.317m
\]
Substituting in to the $Eq3.31$

$$91.89Nm + 0.07FN - 0.317Fr = 0$$ \hspace{1cm} .........................Eq(3.21)

Solving $Eq3.11, Eq3.17, Eq3.18$ and $Eq3.21$ simultaneously

$$F_r = F_x = 335.9N, F_y = 60N \text{ and } F_N = 204.9N$$

3.3.1.6. Pump Analysis

Speed

Pump speed, or more correctly, stroke rate, is one of the most critical selection criteria for piston pumps. The rotating and reciprocating parts of the power end, as designated, are often capable of speeds twice that of the actual pump rating. The maximum pump speed is determined by the design of the fluid end, the hydraulic capability of the anticipated suction system, and the required life of the plungers, packing, and valves. Most piston pump standards limit the plunger speed from (0.71 to 1.42 m/s) (William K. et al, 2001).

Determination of the RPM of the pump

Taking the radius of the driving wheel to be 32.5cm and an average speed of operation of the machine 4km/hr, the machine covers a distance of 100m in 100 seconds.

Taking the distance covered 100m the number of revolution of the wheel can be calculated as

$$\text{number of revolution} = \frac{\text{distance traveled}}{\text{circumference of the wheel}}$$ \hspace{1cm} .........................Eq(3.22)

$$= \frac{100 \text{ m}}{2\pi r}$$

$$\text{number of revolution} = 48.9$$

Therefore the number of revolution per minute will be

$$\frac{\text{number of revolution}}{\text{time in minute}} = \frac{48.9}{1.66} = 29.5 \text{ rpm}$$
Taking the driving sprocket rpm the same as that of the driving wheel rpm the angular velocity of the smaller sprocket can be calculated using the formula

\[ N_1 \omega_1 = N_2 \omega_2 \]  
\[ \frac{N_1}{N_2} = \frac{\omega_2}{\omega_1} \]

\[ \frac{14}{42} = \frac{29.5}{\omega_1} \]

\[ = 88.5 \text{ rpm} \]

Where

\( \omega_1 \) = angular velocity of the smaller sprocket

\( \omega_2 \) = angular velocity of the larger sprocket

\( N_1 \) = number of teeth on smaller sprocket

\( N_2 \) = number of teeth on larger sprocket

The plunger speed is

**Capacity**

The capacity \( Q \) is the total volume of fluid delivered per unit of time. This fluid includes liquid, entrained gases, and solids at the specified conditions (William. K et al, 2001).

**Displacement**

Displacement \( D \), (m\(^3\)/h), is the calculated capacity of the pump with no slip losses. For single-acting plunger or piston pumps, this is

\[ D = A \times n \times s \times 6 \times 10^{-8} \]

\[ A = \pi r^2 \]

\[ = 3.14 \times (17 \text{mm})^2 \]

\[ = 907.46 \text{mm}^2 \]

\[ D = 907.46 \text{mm}^2 \times 88.3 \text{rpm} \times 65 \text{mm} \times 6 \times 10^{-8} \]

\[ = 0.312 \frac{\text{m}^3}{\text{h}} \]
Where

\[ A = \text{cross-sectional area of plunger or piston, (mm}^2\) \]
\[ r = \text{radius of the piston (17mm)} \]
\[ m = \text{number of plungers or pistons} \]
\[ n = \text{rpm of pump} \]
\[ s = \text{stroke of pump, in (mm)} \]

**Mechanical Efficiency**

The mechanical efficiency of a single-acting piston pump without internal gears is typically 90 to 92\%. Over half of the mechanical losses are due to the frictional drag of the plungers through the packing. The remaining losses are from the bearings, the crosshead-to-crosshead guide friction, and the extension rod-to-gland seal friction. If these components are properly lubricated, very few power-end design options are available that will produce a measurable increase in mechanical efficiency. Decreasing the diameter of the plungers and minimizing the number of packing rings will result in a small efficiency increase (William, et al, 2001).

**Pump Power Requirement**

Power Brake horsepower is a function of a pump’s capacity, differential pressure, and mechanical efficiency. It is an essential criterion for selecting the drive components but is not valuable for pump selection. A large pump operating well below its design rating can meet the same horsepower requirements as a smaller pump running at a higher speed. Unless the application requires a rated pump, it is usually more economical to select a pump at the upper end of its design rating (William, et al, 2001).

The brake horsepower for the pump is

\[
Kw = \frac{QXPdr}{36 \times ME} \nonumber \text{\textbf{Eq(3.25)}}
\]

\[
= \frac{0.312 \times 10.8}{36 \times 0.91}
\]

\[
= 0.104 \text{ KW}
\]
Where

\( P = \) power, kW
\( Q = \) flow rate, L/min
\( p = \) pressure, kPa
\( \eta_m = \) mechanical efficiency, decimal 0.91

**Input Power (\( W_s \))**

The input power is given as (Miller J, 1987).

\[
W_s = T_s \times N_s \times 10^{-4}
\]

\( W_s = T \times N \times 1.05 \times 10^{-4} (Kw)\)

\[
31.69 \times 88.3 \times 1.05 \times 10^{-4} = 0.29 kW
\]

Where

\( T_s = \) Input shaft torque, 31.69Nm

\( N_s = \) Shaft speed rpm, 102.17rpm

**Determination of Required Piston Transitional Speed**

Taking \( Q = 0.312 m^3/h = 8.6 \times 10^{-5} m^3/s\)

\[
Q_p = V \times A \]

\( 8.6 \times 10^{-5} \frac{m^3}{s} = V \times 9 \times 10^{-4} m^2 \)

\[
V = 0.095 \frac{m}{s}
\]

Where

\( Q_p = \) Pump discharge rate m\(^3\)/sec

\( V_p = \) Piston speed in m/sec
\[ A_p = \text{Cross sectional area of the piston, in } m^2 = 9cm^2 = 0.0009m^2 \]

### 3.3.1.7. Chain Design

Power transmission in the sprayer from the wheel to the slider crank mechanism shaft was through chain drive. Therefore, the velocity of the chain, its strength and length had to be determined.

**Chain velocity**

Average chain velocity (Ajit, et.al, 2012)

\[ V_{av} = N \times P \times RPM \]  \[ Eq(3.28) \]

Where

N= the driving sprocket no. of teeth, 42  
RPM= revolution of the driving sprocket, 30.8 rev/min  
P= commercially available chain pitch, 12.7mm

\[ V_{av} = 30.8 \times \left( \frac{12.7}{1000} \right) \times 42 \]

\[ = 16.43m/min \]

\[ = 0.273 \text{ m/sec} \]

Maximum chain velocity (Ajit K. et.al, 2012)

\[ V_{max} = \frac{d}{2} \times \omega = \frac{p \times n}{19100 \sin \left( \frac{180}{N} \right)} \]  \[ Eq(3.29) \]

\[ V_{max} = \frac{d}{2} \times \omega = \frac{12.7 \times 30.8}{19100 \sin \left( \frac{180}{42} \right)} \]

\[ = 0.274 \text{ m/sec} \]

Minimum chain velocity (Ajit K. et.al, 2012)

\[ V_{min} = \frac{d_n}{2} \times \omega = \frac{p \times n}{19100 \tan \left( \frac{180}{N} \right)} \]  \[ Eq(3.30) \]
\[ V_{\text{min}} = \frac{d_n}{2} \omega = \frac{12.7 \times 30.8}{19100 \tan \left( \frac{180}{42} \right)} \]

\[ = 0.273 \text{ m/sec} \]

Chordal speed variation (Ajit K. et.al, 2012)

\[
\delta = \frac{\Delta V}{V_{\text{max}}} = \frac{V_{\text{max}} - V_{\text{min}}}{V_{\text{av}}} \]

\[ = \frac{0.274 - 0.273}{0.273} \]

\[ = 0.0036 \]

Thus, since the chordal speed variation is so small, the vibration that would be existed in the system is negligible.

**Chain force** (Sharma et.al, 2010)

The total load (force) on the driving side of the chain is given by

\[ F_T = F + F_c + F_f \]

\[ \text{Eq(3.32)} \]

Where

\[ F_T = \text{the total force, N} \]

\[ F = \text{the force due to power transmission, 1156.3N} \]

\[ F_f = \text{frictional force, N} \]

\[ P = \text{power at the sprayer wheel/ power to be transmitted/, 0.054Hp} \]

\[ F_c = \frac{w(V_{av})^2}{g} \]

\[ \text{Eq(3.33)} \]

Where:

\[ F_c = \text{centrifugal force on the chain, N} \]

\[ V_{av} = \text{average chain velocity, m/sec} \]

\[ g = \text{gravitational attraction, m/s}^2 \]
\[ W = \text{weight per meter of the chain, 4.40N/m} \]

\[
\frac{4.4 \left(0.273 \frac{m}{s}\right)^2}{9.8} = 0.033 \, N
\]

\[ F_f = W \times K_f \times C_c \]

Where:

\[ W = \text{weight per meter of the chain, 4.40N/m} \]

\[ C_c = \text{nominal center to center distance between the sprockets, 0.4m} \]

\[ F_f = \text{frictional force, N} \]

\[ K_f = \text{friction factor= 4 for horizontal drive, 2 for inclined drive and 1 for vertical drive} \]

\[ 4.4 \times 2 \times 0.4 = 3.52 \, N \]

So,

\[ F_T = 1156.3 \, N + 0.033 \, N + 3.52 \, N = 1159.85 \, N \]

American National Standard Institute (ANSI) test reveals that chain number ANSI 40 has minimum tensile strength \( T_{sm} \) of 14,100N. So, the safety factor must be more than unity in order the chain to be safe from failure or breakage.

Checking for the safety factor, \( S_f \) (Sharma et.al. 2010)

\[ S_f = \frac{T_{sm}}{F_T} \]

Where:

\[ T_{sm} = \text{Minimum tensile strength} \]

\[ S_f = \text{Safety factor} \]

\[ \frac{14100}{1159.85} = 12.15 \]
Therefore, we can conclude that the chain is safe from breakage.

**Chain Length** (Sharma, et.al, 2010)

The length of the chain is calculated based on the following formula.

\[ L_c = P \times L_p \] \hspace{1cm} Eq(3.36)

And,

\[ L_p = 2C_p + \frac{N_1+N_2}{2} + \frac{(N_2-N_1)^2}{2 \pi C_p} \] \hspace{1cm} Eq(3.37)

\[ C_p = \frac{C_c}{p} \] \hspace{1cm} Eq(3.38)

Where

\( L_c \)= chain length, m

\( L_p \)=chain length, in pitches

\( C_p \)=center to center distance between the sprockets, in pitches

\( C_c \)=approximate center to center distance between the sprockets, 400 mm

\( P \)= chain pitch, 6.35mm

\( N_1 \)=number of teeth of the driven sprocket, 14

\( N_2 \)=number of teeth of the driving sprocket, 42

Solving for \( C_p \)

\[ C_p = \frac{400}{12.7} \]

\[ = 31.49 \]
Solving for $L_p$

\[ L_p = 2C_p + \frac{N1 + N2}{2} + \frac{(N2 - N1)^2}{2\pi C_p} \]

\[ L_p = 2(31.49)_p + \frac{14 + 42}{2} + \frac{(42 - 14)^2}{31.49} \]

= 91.611 pitches

We should take the pitches in whole number, 91 pitches

Solving for $L_c$

\[ L_c = P \times L_p \]

\[ L_c = 92 \times 12.7 \]

1168.4mm = 1.168m

Now, since the chain length in pitches is changed from 91.611 to 92 pitches, the center to center distance of the sprockets has to be corrected. (Sharma, et.al. 2010)

\[ C_c = \left( e + \frac{(e^2-8m^{0.5})XP}{4} \right) \] ..........Eq(3.39)

\[ And \]

\[ e = L_p - \frac{N1+N2}{2} \] ..........Eq(3.40)

\[ e = 91.611 - \frac{14 + 42}{2} \]

= 63.6

\[ m = \left( \frac{N2-N1}{2\pi} \right)^2 \] ..........Eq(3.41)

\[ m = \left( \frac{42 - 14}{2\pi} \right)^2 \]

= 19.87
Therefore, the corrected center to center distance between the sprockets is

\[
C_c = (63.6 + \frac{(63.6^2 - 8 \times 19.87^{0.5}) \times 12.7}{4})
\]

\[
399.83 \text{mm} \sim 400 \text{mm}
\]

**Final Selection of Sprocket Teeth Ratio**

Taking the minimum required transitional piston velocity to be 0.11m/sec the required crank arm angular velocity was calculated using

\[
V = \omega \left( L_{AB} \sin \theta_{AB} - \frac{L_{AB}^2 \sin 2\theta_{AB}}{2(t_{BC}^2 - L_{OA})^2} - \frac{L_{OA} L_{AB} \cos \theta_{AB}}{(t_{BC}^2 - L_{OA}^2)^2} \right)
\]

\[
\text{Eq(3.42)}
\]

\[
V = \omega \left( 0.028 \sin 45 - \frac{0.028^2 \sin 90}{2(54^2 - 27)^2} - \frac{27 \times 0.028 \cos 45}{(54^2 - 27^2)^2} \right)
\]

\[
\omega = 9.24 \frac{\text{rad}}{s} = 29.5 \text{rpm}
\]

Considering the angular velocity required by the crank arm and the angular velocity that the wheels produce the teeth ratio on the smaller and bigger sprockets can be determined using the following formula

\[
N_1 \omega_1 = N_2 \omega_2
\]

\[
\frac{N_1}{N_2} = \frac{\omega_2}{\omega_1}
\]

\[
\frac{N_1}{N_2} = \frac{88.3}{29.5}
\]

Teeth ratio of 3

Based on the teeth ratio it is possible to select smaller and bigger sprockets combination with teeth ratio of 3, found in the mark of smaller teeth 14 and larger teeth 42, is selected

**Crank Shaft Design**

A shaft is a rotating member, usually of circular cross section, used to transmit power or motion. It provides the axis of rotation, or oscillation, of elements such as gears, pulleys, sprockets etc. The diameter and bearings of the shaft that drives the slider crank mechanism was determined based on the load diagram below. *(Ajit, et al.2012)*
Fig.3.6 3D free body diagram of the shaft

Where

F = force due crank arm, 1100.34N

Fx = force due crank force in horizontal direction, N

Fy = force due crank force in horizontal direction, N

F_{DH} = bearing at C reaction in horizontal direction, N

F_{DV} = bearing at C reaction in vertical direction, N

F_{AH} = bearing at A reaction in horizontal direction, N

F_{AV} = bearing at A reaction in vertical direction, N

T_{C} = tension in the chain at an angle \theta, N

**Step1** determining the angle the chain force acting on the shaft (see fig.3.8)

Fig.3.7 chain force application angle
Where

\( C_C = \) center to center distance between the sprockets, 400mm

\( r = \) pitch radius of the driving sprocket, cm

\( r_o = \) pitch radius of the driven sprockets, cm

\( N_1 = \) number of teeth of the driving sprocket, 42

\( N_2 = \) number of teeth of the driven sprocket, 14

\( P = \) chain pitch, 1.27cm

\[ L = (15.8 - r) + r_o \]

And,

\[ r = 0.1 \times \frac{p}{2sin\left(\frac{180}{N_1}\right)} \]

\[ r = 0.1 \times \frac{12.7}{2sin\left(\frac{180}{42}\right)} \]

\[ = 8.5\text{cm} \]

\[ r_o = 0.1 \times \frac{12.7}{2sin\left(\frac{180}{14}\right)} \]

\[ = 2.85\text{cm} \]

\( and, \)

\[ L = (15.8 - 8.5) + 2.85 \]

\[ = 10.6\text{cm} \]

Therefore, the angle through which the chain force acts from the horizontal,

\[ tan\theta = \frac{L}{39} \]

\[ = \frac{10.6}{39} \]

\[ = 0.267 \]

42
\[ \theta = \tan^{-1}(0.67) \]
\[ = 15^0 \]

**Step 2** loads on the shaft (fig. 3.9) in the horizontal plane (XZ)

![Fig. 3.8 load diagram of the main shaft in XZ plane]

Where

\[ F_x = F \cos \theta \]
\[ = 1156.3 N \cos 15^0 \]
\[ = 1116.94 N \]

Here the unknown forces are \( F_{AH} \) and \( F_{CH} \) (equilibrium)

\[ F_{AH} + F_x - F_{CH} - F_{CX} = 0 \]

Taking moment at C,

\[ 8 \times 618.35 N - 9.5 F_{AH} + 4.5 \times 1156.3 N \times \cos 15^0 = 0 \]

\[ F_{AH} = 1049.76 N \]

Substituting the Eq3.56,

\[ F_{CH} = 618.35 N + 1049.76 N - 1116.94 N \]
\[ = 551.17 N \]
Shear and bending moment diagrams of the loads on the shaft in XZ plane (see fig. 3.10)

Fig. 3.9 shear and bending moment diagrams of the main shaft in XZ plane

**Step 3** loads on the shaft (fig. 3.11) in the vertical plane (YZ)

Fig. 3.10 load diagram of the main shaft in YZ plane

\[ F_y = F \sin \theta \]

\[ = 1156.3 \times \sin 15^\circ \]

\[ = 89.4 \text{N} \]

Here the unknown forces are \( F_{AV} \) and \( F_{CV} \)

\[ F_y + F_y - F_{CV} - T_{CY} = 0 \]

\[ F_{AV} + F_{CV} = 1283.96 \]

Eq(3.45)
Taking moment at A,

\[ 9.5F_Y + 5 \times 299.27 - 19.5 \times 984.69 = 0 \]

\[ F_{CV} = 1863.69N \]

Substituting this value in eq.1 yields,

\[ F_{AV} + F_Y - FCY - TCY = 0 \]

\[ F_{AV} = 1863.69N + 299.27N - 984.69N \]

\[ 1178.27N \]

Shear and bending moment diagrams of the loads on the shaft in YZ plane (see fig.3.10)

![Shear and bending moment diagrams of the main shaft in YZ plane](image)

**Step 4** Determination of the maximum bending moment (Richard, et.al, 2011)

\[ M_{max} = (M_H^2 + M_Y^2)^{0.5} \]

\[ M_A = (0^2 + 0^2)^{0.5} = 0 \]

\[ M_B = (52.48^2 + 58.9^2)^{0.5} = 78.89 \text{ NM} \]

\[ M_C = (49.46^2 + 78.77^2)^{0.5} = 93 \text{ NM} \]

\[ M_D = (0^2 + 0^2)^{0.5} = 0 \]

Thus, the maximum bending moment occurs at bearing C.
Step 5 Determination of torque on the shaft (Richard et al., 2011)

\[ P = T_1N_1 = T_2N_2 \] \( Eq(3.47) \)

\[ T_2 = T_1 \frac{N_1}{N_2} \]

\[ 14.34 \left( \frac{29.5}{88.3} \right) \]

\[ = 4.78 \text{ Nm} \]

Where

P = power transmitted

T_1 = torque produced at the wheel, 14.34 Nm

T_2 = torque produced at the crank shaft, Nm

N_1 = angular rotation of the wheel or the driving sprocket, 15.8 RPM

N_2 = angular rotation of the main shaft or the driven sprocket, 3 \times 15.8 = 47.4 RPM

Step 6 Determination of diameter of the shaft …………………… (Richard et al., 2011)

The diameter of the shaft was determined based on the maximum shear stress theory.

\[ \tau_{max} = \frac{0.5\sigma_y}{F_s} = \left( \frac{C_m\sigma}{2} \right)^2 + (C_t\tau)^2 \]

\[ \frac{16}{\pi d^3} \left( (C_m M_{max})^2 + (C_t T)^2 \right)^{0.5} \]

Where

\( \tau_{max} \) = maximum shear stress

\( \sigma_y \) = yield stress of the shaft, 150 MPa

\( F_s \) = factor of safety, 1.5

\( C_m \) = numerical combined shock and fatigue factor for bending moment, 1.5

\( C_t \) = numerical combined shock and fatigue factor for torque, 1

\( M_{max} \) = maximum bending moment on the shaft, 93 Nm
T= net torque on the shaft, 4.78Nm

d= diameter of the shaft (40 C 8), cm

Thus, the diameter of the shaft,

\[
d^3 = \frac{32F_s}{\pi \sigma_y} \left( (C_m M_{max})^2 + (C_T T)^2 \right)^{0.5}
\]

\[
d^3 = \frac{32 \times 1.5}{3.14 \times 150} \left( (1.5 \times 93 \text{ NM})^2 + (1 \times 4.78)^2 \right)^{0.5}
\]

\[
d = 2.4 \text{ cm}
\]

Step 7 Ball bearing size determination (Richard et.al, 2011)

The figure below shows a ball bearing identical to the one used in the designed planter.

![Fig.3.12. Standard designation of ball bearing](image)

The bearing size was determined using the maximum resultant reaction force acting on one of the bearings on the shaft and the desired maximum lifespan of the bearings.

Resultant reaction forces on bearing A and bearing C (see fig.3.7)

\[
R = (R_x^2 + R_y^2)^{0.5}
\]

\[
R_A = (1049.76^2 + 1178.27^2)^{0.5} = 1.57 \text{ KN}
\]

\[
R_C = (1863.69^2 + 551.17^2)^{0.5} = 1.94 \text{ KN}
\]
Basic dynamic load rating

\[ C = R \left( \frac{L}{10^6} \right)^{1/3} \] \hspace{1cm} Eq(3.50)

\[
C = 1.94KN \left( \frac{47.4 \times 30000}{10^6} \right)^{1/3} \]

\[ = 6.68 \text{KN} \]

Where

C= basic dynamic load rating

R= the maximum resultant reaction force on the bearings, \( R_C = 1.943\text{KN} \)

L= total number of revolution of the bearings, rev

\( N_2 = \) revolution of the shaft, \( 3 \times N_1 = 47.4\text{RPM} \)

\( H_m = \) the desired maximum lifespan of the bearings, let it be 30000hrs

Therefore, from the O3 series/medium series/ table of bearings (Richard et.al, 2011), two bearings having bore size of 18mm and outside diameter of 38mm were selected for the shaft.

3.3.2. Manufacturing of Main Parts

3.3.2.1. Manufacturing of Tank And Pump

In manufacturing the sprayer a standard 20 litter knapsack sprayer were dismantled and the tank with piston pump integration is used.

Fig3.13. knapsack tank with piston of 20 litter capacity
3.3.2.2. Manufacturing of Main Frame

To manufacture the main frame 3/4" galvanized pipe is used. The pipe is bent in different three points at different angle and direction to form a wheel barrow shape. Two symmetric wheel barrow shaped pipes were joined using RHS forming a wheel barrow profile.

![Fig3.14. Front, top, side, isometric views and assembled main frame](image1)

3.3.2.3. Wheel

The rear wheel of a mountain bicycle is used with some modification on the sprocket hub. The sprocket hub is modified to hold a circular plate to attach a sprocket using bolts and nuts.

![Fig.3.15. Front, top, side, isometric views and assembled wheel](image2)
3.3.2.4. Manufacturing of Boom

The boom is manufactured from two separate galvanized pipes. The two booms have three nozzles with a nozzle adapter attached with screwed clamp. The booms are fixed to a common holder with clamps so that both booms move together. The boom holder in turn is fixed on a vertical frame using screw clamp, which the boom holder can slide up and down, that helps to adjust the spray height.

Fig3.16. Boom, nozzle and boom holder view and assembly

3.3.2.5. Complete Sprayer Assembly

The sprayer main parts were assembled with bolt and nuts, straps to secure in place and some clamps. In the assembling the main parts only three different sized bolts and nuts were used. This makes easy maintenance, lower spare part need and only few tools are needed.

Fig3.17. Front, top, side, isometric views and sprayer assembly
**Fig3.18. complete sprayer assembly parts**

**Fig3.19. CAD model of the wheel barrow boom sprayer**
3.3.3. Performance Evaluation of The Sprayer

The performance of the sprayer will be evaluated both in laboratory and field and accordingly the following parameters will be considered to be evaluated both in laboratory and field. The relevant data will be collected on both evaluations and analyzed.

3.3.3.1. Laboratory Test

Nozzle Discharge rate

Nozzle discharge test was done to evaluate the amount of liquid discharge from each nozzle and to check the variation between the discharge rates of each nozzle within 5 meter intervals of 20m. Liquid was pumped as the sprayer moves and in each 5m interval the time taken and discharge data was collected and recorded for each nozzle by tying a plastic bag on each nozzle. After each 5m interval the sprayer were stopped, as the sprayer stops moving the pump stops pumping and each liquid was collected from the nozzle and measured using measuring cylinder. Coefficient of variation was used to analyze variation of discharge rate among the nozzles for each 5m interval.

**Fig3.20. sprayer track test**

Uniformity of Coverage

Liquid chemical application Sprayer performance is evaluated by the uniformity of coverage and spray patterns, droplet size and size distribution, and target deposition and drift.
**Spray Overlap**

The overlap is defined as the width covered by two adjacent nozzles divided by the width covered by a single nozzle, expressed in percent. It mainly affects spray pattern of the sprayer it depends on the boom height and nozzle spacing.

The test was done on a test track using a dye. First the test track was painted with a whitish blue color then a black dye was used as a water solution to get a good contrast between the track and the spray solution. The sprayer was tested for 50cm boom height and 50cm nozzle spacing spray. The measurement was taken within 20m distance at an interval of 5 m.

![Fig3.21. spray overlap test](image)

**3.3.3.2. Field Test**

The field size, test duration, spray pressure, swath, discharge, speed of operation, field capacity and other relevant information were taken. The field test were conducted on 0.25 ha trial field at Melkasa.

Nozzle discharge test was done to evaluate the amount of liquid discharge from each nozzle and to check the variation between the discharge rates of each nozzle within 30 meter. Liquid was pumped as the sprayer moves and discharge data was collected and time taken for each trial was recorded. Three replications were undertaken for this test. The discharge from each nozzle was collected by tying a plastic bag on each nozzle. After each 30 m interval the sprayer were stopped, as the sprayer stops the pump stops pumping and each liquid collected from the nozzle was measured using measuring cylinder. Coefficient of variation was used to analyze variation of discharge rate among the nozzles for each trial.
Field Capacity of the Sprayer

Actual field capacity: (Sharma, et al 2010).

For calculating actual field capacity the time consumed for real work and that lost for other activities such as turning, filling of tank were taken into consideration. The time required for actual operation and time lost measured by stopwatch.

Actual capacity field was calculated by

\[ \text{Actual field capacity} = \frac{A}{T_{\text{total}}} \]  
\[ T_{\text{total}} = \text{time for turning} + \text{time for refilling} + \text{time for actual work} \]

\[ = 6 \text{ sec} \times 17 + 25 \text{ sec} \times 4 + 15 \text{ min} \]

\[ = 0.30 \text{ hr} \]

Where

\[ A = \text{area covered, 0.25 ha} \]

\[ T_{\text{total}} = \text{total time taken, } 0.3 \text{ h} \]

\[ = \frac{0.25 \text{ ha}}{0.30 \text{ hr}} \]

\[ 0.83 \frac{\text{ha}}{\text{hr}} \]
Theoretical Field Capacity

Theoretical field capacity was calculated by (Sahay, 2008).

\[ TFC = \frac{\text{Speed} \times \text{boom Width}}{10} \] .......................... \text{Eq}(3.52)

\[ = \frac{4 \text{ km/hr} \times 3 \text{ m}}{10} \]

\[ = 1.04 \]

Field Efficiency

Field efficiency is the ratio of actual field capacity to the theoretical field capacity; field efficiency is expressed in %, (Sahay, 2008).

\[ \text{Field efficiency} = \left( \frac{\text{Actual field capacity}}{\text{Theoretical field capacity}} \right) \times 100 \] .......................... \text{Eq}(3.53)

\[ \frac{0.83}{1.04} \times 100 \]

\[ (0.83/1.04) \times 100 \]

82.7%
CHAPTER FOUR

4. RESULT AND DISCUSSION

Table 4.1. Performance Parameters of the Prototype Sprayer

<table>
<thead>
<tr>
<th>No</th>
<th>Functional</th>
<th>parameters</th>
<th>values</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td></td>
<td>Number of nozzle and spacing, mm</td>
<td>6 x 500</td>
</tr>
<tr>
<td>2</td>
<td></td>
<td>Swath width, m</td>
<td>3.00</td>
</tr>
<tr>
<td>3</td>
<td></td>
<td>Mean pressure, bar</td>
<td>1.80</td>
</tr>
<tr>
<td>4</td>
<td></td>
<td>Discharge rate, ml/min/nozzle</td>
<td>882.00</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Performance</td>
<td></td>
</tr>
<tr>
<td>5</td>
<td></td>
<td>Quantity of solution sprayed, l</td>
<td>70.25</td>
</tr>
<tr>
<td>6</td>
<td></td>
<td>Effective time, (min)</td>
<td>14.75</td>
</tr>
<tr>
<td>7</td>
<td></td>
<td>Lost time, min</td>
<td>1.67</td>
</tr>
<tr>
<td>8</td>
<td></td>
<td>Total time, min</td>
<td>16.42</td>
</tr>
<tr>
<td>9</td>
<td></td>
<td>Field size or Area treated, ha</td>
<td>0.25</td>
</tr>
<tr>
<td>10</td>
<td></td>
<td>Forward speed, km/hr</td>
<td>4.00</td>
</tr>
<tr>
<td>11</td>
<td></td>
<td>Field capacity, h/hr</td>
<td>0.99</td>
</tr>
<tr>
<td>12</td>
<td></td>
<td>Field efficiency, %</td>
<td>95.20</td>
</tr>
</tbody>
</table>

Table 4.2. Discharge rate of individual nozzle in the 1st 5m

<table>
<thead>
<tr>
<th>Rep</th>
<th>Discharge, ml/sec</th>
<th>Mean discharge in ml/sec</th>
<th>CV%</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Nozzle No</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>N1</td>
<td>N2</td>
<td>N3</td>
</tr>
<tr>
<td>1</td>
<td>10.00</td>
<td>10.83</td>
<td>10.83</td>
</tr>
<tr>
<td>2</td>
<td>10.83</td>
<td>10.33</td>
<td>11.17</td>
</tr>
<tr>
<td>3</td>
<td>10.83</td>
<td>10.83</td>
<td>10.83</td>
</tr>
<tr>
<td>Average</td>
<td>10.55 &amp; 10.66 &amp; 10.94 &amp; 11 &amp; 10.66 &amp; 10.27 &amp; 10.68 &amp; 3.20%</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

\[N1, N2 \ldots N6\] are six flat fan nozzles fitted on the boom at 50cm spacing
### Table 4.3. Discharge rate of individual nozzle in the 2nd 5m

<table>
<thead>
<tr>
<th>Rep</th>
<th>Discharge, ml/sec</th>
<th>Mean discharge in ml/sec</th>
<th>CV%</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>N1</td>
<td>N2</td>
<td>N3</td>
</tr>
<tr>
<td>3</td>
<td>13.00</td>
<td>13.83</td>
<td>13.33</td>
</tr>
</tbody>
</table>

N1, N2 … N6 are six flat fan nozzles fitted on the boom at 50cm spacing

### Table 4.4. Discharge rate of individual nozzle in the 3rd 5m

<table>
<thead>
<tr>
<th>Rep</th>
<th>Discharge, ml/sec</th>
<th>Mean discharge in ml/sec</th>
<th>CV%</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>N1</td>
<td>N2</td>
<td>N3</td>
</tr>
<tr>
<td>1</td>
<td>14.60</td>
<td>14.00</td>
<td>14.00</td>
</tr>
<tr>
<td>2</td>
<td>14.40</td>
<td>14.00</td>
<td>14.80</td>
</tr>
<tr>
<td>3</td>
<td>14.00</td>
<td>14.00</td>
<td>14.00</td>
</tr>
</tbody>
</table>

N1, N2 … N6 are six flat fan nozzles fitted on the boom at 50cm spacing

### Table 4.5. Discharge rate of individual nozzle in the 4th 5m

<table>
<thead>
<tr>
<th>Rep</th>
<th>Discharge, ml/sec</th>
<th>Mean discharge in ml/sec</th>
<th>CV%</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>N1</td>
<td>N2</td>
<td>N3</td>
</tr>
<tr>
<td>3</td>
<td>14.90</td>
<td>14.75</td>
<td>14.27</td>
</tr>
<tr>
<td>Average</td>
<td>14.88</td>
<td>14.68</td>
<td>14.60</td>
</tr>
</tbody>
</table>

N1, N2, … N6 are six flat fan nozzles fitted on the boom at 50cm spacing

During the discharge rate test in 20m with 5m interval of individual nozzle, the average nozzles discharge varies from 11 to 10.27 ml/sec, 13.72 to 13.33 ml/sec, 14.80 to 14.00 ml/sec and 14.88 to 14.60 with the average discharge of 10.68 ml/sec, 13.55 ml/sec, 14.32 ml/sec and 14.71 ml/sec at the 1st, 2nd, 3rd and 4th 5m intervals respectively at a bar pressure of 1.8 bars. The coefficient of variation for the average of nozzle discharges was 3.20%, 2.19%, 1.31% and 1.32% in 1st 5m, 2nd 5m, 3rd 5m and 4th 5m, respectively, which showed that the variation
in discharges of the nozzle was below acceptable variation of 10% as per the recommendation (Norbdy, 1978) and (Gomez and Gomez, 1984).

Based on the CV’s shown in the above tables the variability of discharge rate of the nozzles along the boom from the center to the extreme end of the boom in both direction decreases. As shown in the tables above the CV’s decreases from 3.20%, in the 1st 5m to 1.32% in the last 4th 5m which shows the variability in discharge rate decreases significantly within 20m. This is also shown in fig3.23 in graph. As shown in the figure the average discharge rate from all the six nozzles was 14.71ml/sec in the last 5m.

As the prayer advances the average discharge increases from 11 ml/sec in the 1st 5m to 14.32 ml/sec in the 3rd 5m interval which more or less remains constant in the last 5m of 14.71 ml/sec at the 20m mark. This variation in the average discharge along the intervals is due to the fact that the pump gets power from the ground wheel as it advances; therefore it takes some distance for the sprayer to attain optimal pressure at each nozzle along the boom.
Table 4.6. Spray overlaps between nozzles

<table>
<thead>
<tr>
<th>Data points</th>
<th>Overlap in cm</th>
<th>Mean average overlap</th>
<th>CV%</th>
</tr>
</thead>
<tbody>
<tr>
<td>at 5 m</td>
<td>N1-N2 24.7</td>
<td>N2-N3 25</td>
<td>N4-N5 23.5</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>at 10m</td>
<td>N2-N3 24.5</td>
<td>N3-N4 24.8</td>
<td>N4-N5 24.5</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>at 15m</td>
<td>N3-N4 25</td>
<td>N4-N5 23.5</td>
<td>N5-N6 22.8</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>at 20m</td>
<td>N4-N5 23.5</td>
<td>N5-N6 23.5</td>
<td>N6 23.6</td>
</tr>
<tr>
<td>average overlap</td>
<td>24.425</td>
<td>24.45</td>
<td>23.575</td>
</tr>
<tr>
<td>average % overlap</td>
<td>32.83</td>
<td>32.86</td>
<td>31.69</td>
</tr>
</tbody>
</table>

N1-N2, N2-N3 ... are the adjacent six flat fan nozzles fitted on the boom at 50cm spacing

During the spray overlap test in the average% overlap between the nozzles varies from 31.69% to 32.86 at boom height of 50cm and nozzle spacing of 50cm, which showed that the variation in overlap of the nozzle was within the range of 30 – 100%. The coefficient of variation (CV %) of percentage overlap between the nozzles were 1.73% which shows very small variability of overlap between consecutive adjacent nozzles. These mean that the uniformity and coverage of the spray was good.

Table 4.7. Discharge rate of individual nozzle in the field test

<table>
<thead>
<tr>
<th>Rep</th>
<th>Discharge in ml/sec</th>
<th>Mean discharge in ml/sec</th>
<th>CV%</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Nozzle No</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>N1 12.90</td>
<td>12.90</td>
<td>12.71</td>
</tr>
<tr>
<td></td>
<td>N2 13.03</td>
<td>13.03</td>
<td>13.13</td>
</tr>
<tr>
<td></td>
<td>N3 13.18</td>
<td>13.18</td>
<td>13.28</td>
</tr>
<tr>
<td></td>
<td>N4 13.48</td>
<td>13.48</td>
<td></td>
</tr>
<tr>
<td></td>
<td>N5 13.88</td>
<td>13.88</td>
<td></td>
</tr>
<tr>
<td></td>
<td>N6 13.93</td>
<td>13.93</td>
<td></td>
</tr>
<tr>
<td>Average</td>
<td>13.34</td>
<td>13.13</td>
<td>12.93</td>
</tr>
</tbody>
</table>

N1, N2 ... N6 are six flat fan nozzles fitted on the boom at 50cm spacing

The field size, test duration, spray pressure, swath, discharge, speed of operation, field capacity and other relevant information were taken. The average discharge rate varies from 13.34 to 12.67 ml/sec with the average discharge of 13.04ml/sec it was observed that at an operating pressure of 1.8bars at the forward speed of 4 km/hr. The operating pressure was maintained constant as the sprayer advances getting the traction force from the ground. It is observed that the quantity of fluid sprayed was 70.32 litters on 0.25 hectares of land, which gave an application rate of 281.3 l/ha. The actual field capacity of 0.83 ha/hr was calculated
with field efficiency of 82.7% and good uniformity of spray. Comparing the application rate of the prototype with that of the conventional knapsack sprayer, as given by (Malik et al 2012), shows that the 281.3 l/ha application rate of the sprayer with 200 l/ha of the knapsack sprayer represents a 28.9% increase in application rate. Also a field capacity of 0.83 ha/hr and 82.7% field efficiency when compared with 0.4 ha/day and 56% of the knapsack sprayer.

During both the field and laboratory discharge rate test of individual nozzle, the discharge rate is lower for the nozzles mounted at both ends of boom as compared to nozzle mounted in the middle of boom. It was due to increase in frictional force during flow of liquid to reach both ends of the boom and the pressure in the boom builds gradually as the sprayer advance so it took 15 to 20 m to build a uniform pressure throughout the boom. The coefficient of variation of the average nozzle discharges was 2.80%, which showed that the variation in discharges of the nozzle was particularly good when it is less than 10% for field operation (Gomez and Gomez, 1984).
CHAPTER FIVE

5. CONCLUSION AND RECOMMENDATION

5.1. Conclusion

Based on results above the sprayer has managed to maintain the nozzle pressure of 1.8 bar during both the field and laboratory test with an average speed of 4km/hr. The average nozzles discharge variation along travel distance reduced and attained a uniform discharge among the nozzles within 20m distance. The sprayer forward speed and spray application are synchronized, so that once the sprayer attained the optimal uniform discharge along all the nozzles it will maintain its uniformity till the next tank filling. The sprayer is designed with tank capacity of 20 litters, a full tank can cover an area of 0.07ha, which needs a refilling of 14 times to cover one hectare with an application rate of 281 lit/ha and an average discharge rate 13.04ml/s.

The coefficient of variation (CV %) of the percentage overlap between the nozzles were 1.73% which shows a good uniformity of coverage of the sprayer as compared with that of the knapsack sprayer and animal drawn sprayers, which in these sprayers the forward speed and application rate not synchronized as that of the prototype. The prototype has an effective field capacity of 0.83 ha/hr, which is very significant as compared with conventional knapsack sprayer. Moreover the sprayer applies the pesticide at about 3m away from the operator, which minimized the chances of exposure of chemical to the operator and alleviates carrying the chemical tank at the back of the operator’s shoulder.

5.2. Recommendation

Though the sprayer has all the above advantages further improvements could be done to improve its capacity by improving the capacity of tanker and pump, which could reduce the frequency of refilling and increasing the boom length and number of nozzles.

The operation speed could also be enhanced by installing pressure relief valve, which help to regulate the pressure at the nozzles. Since the machine gets the power from the ground wheel, as the operation speed increases so do the pressure at the nozzles. So the relief valve could solve this problem and allows the operator to operate even above the average human speed.
REFERENCE


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Carroll E. (2012). Engineering Principles of Agricultural machines, 2nd ed. American Society of Agricultural and Biological Engineers 2950 Niles Road, St. Joseph, MI 49085-9659 USA


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## A. The designed sprayer test result summary

<table>
<thead>
<tr>
<th>TEST PARAMETER</th>
<th>UNIT</th>
<th>RESULT</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Discharge</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>A. laboratory test</td>
<td>l/min</td>
<td>5.9</td>
</tr>
<tr>
<td>B. Field test</td>
<td></td>
<td>5.34</td>
</tr>
<tr>
<td><strong>Discharge uniformity</strong></td>
<td>CV%</td>
<td>1.32</td>
</tr>
<tr>
<td>Laboratory test</td>
<td></td>
<td></td>
</tr>
<tr>
<td><strong>Plot size completed, M</strong></td>
<td>M²</td>
<td>625</td>
</tr>
<tr>
<td><strong>Cone index, 0-10cm soil depth</strong></td>
<td>KN/m²</td>
<td>328.34</td>
</tr>
<tr>
<td><strong>Soil moisture</strong></td>
<td>%</td>
<td>18.3</td>
</tr>
<tr>
<td><strong>Sprayer speed, S</strong></td>
<td>m/s</td>
<td>1</td>
</tr>
<tr>
<td><strong>Total time to complete the plot, T</strong></td>
<td>Hr</td>
<td>0.30</td>
</tr>
<tr>
<td><strong>Number of nozzles N</strong></td>
<td>No.</td>
<td>6</td>
</tr>
<tr>
<td><strong>Average turning time per row, t</strong></td>
<td>Sec</td>
<td>6.28</td>
</tr>
<tr>
<td><strong>Average refilling time, t</strong></td>
<td>Sec</td>
<td>15.23</td>
</tr>
<tr>
<td><strong>Actual field capacity, A= M/(10000T)</strong></td>
<td>Ha/Hr</td>
<td>0.99</td>
</tr>
<tr>
<td>**Field efficiency, f= [T-[R-1]t/3600] *100</td>
<td>%</td>
<td>80</td>
</tr>
<tr>
<td><strong>Horizontal push, F= B cosθ</strong></td>
<td>Kgf</td>
<td>35.3</td>
</tr>
<tr>
<td><strong>Pump power input = FS/101.64</strong></td>
<td>KW</td>
<td>0.24</td>
</tr>
</tbody>
</table>
### A. Sprayer dimension

<table>
<thead>
<tr>
<th>No.</th>
<th>DESCRIPTION</th>
<th>VALUE</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Overall dimension of the planter</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Length, mm</td>
<td>830</td>
</tr>
<tr>
<td></td>
<td>Width, mm</td>
<td>1000</td>
</tr>
<tr>
<td></td>
<td>Height, mm</td>
<td>700</td>
</tr>
<tr>
<td>2</td>
<td>Wheel, tier</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Number, No.</td>
<td>1</td>
</tr>
<tr>
<td></td>
<td>Diameter, mm</td>
<td>650</td>
</tr>
<tr>
<td></td>
<td>Width, mm</td>
<td>50</td>
</tr>
<tr>
<td></td>
<td>Thickness, mm</td>
<td>50</td>
</tr>
<tr>
<td>5</td>
<td>Chain</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Pitch, mm</td>
<td>12.7</td>
</tr>
<tr>
<td></td>
<td>Length, mm</td>
<td>1155.7</td>
</tr>
<tr>
<td></td>
<td>Angle, degree</td>
<td>15</td>
</tr>
<tr>
<td>6</td>
<td>Sprocket teeth</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Driving, No.</td>
<td>42</td>
</tr>
<tr>
<td></td>
<td>Pinion, No.</td>
<td>14</td>
</tr>
<tr>
<td>7</td>
<td>Tanker, plastic</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Capacity, litter</td>
<td>20</td>
</tr>
<tr>
<td></td>
<td>width, mm</td>
<td>150</td>
</tr>
<tr>
<td></td>
<td>breadth, mm</td>
<td>180</td>
</tr>
<tr>
<td></td>
<td>Height, mm</td>
<td>360</td>
</tr>
<tr>
<td>8</td>
<td>pump, piston pump</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Diameter, mm</td>
<td>34</td>
</tr>
<tr>
<td></td>
<td>Stroke length, mm</td>
<td>65</td>
</tr>
<tr>
<td>9</td>
<td>Metering plate cover, sheet metal</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Diameter, mm</td>
<td>170</td>
</tr>
<tr>
<td></td>
<td>Width, mm</td>
<td>40</td>
</tr>
<tr>
<td>10</td>
<td>Shaft</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Main shaft, solid</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Length, mm</td>
<td>200</td>
</tr>
<tr>
<td></td>
<td>Diameter, mm</td>
<td>24</td>
</tr>
<tr>
<td>11</td>
<td>Bearing, ball bearing</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Main shaft</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Bore diameter, mm</td>
<td>17</td>
</tr>
<tr>
<td></td>
<td>Outside diameter, mm</td>
<td>38</td>
</tr>
<tr>
<td>12</td>
<td>Reinforcing bar, stainless steel</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Length, mm</td>
<td>22</td>
</tr>
<tr>
<td></td>
<td>Diameter, mm</td>
<td>12</td>
</tr>
<tr>
<td>13</td>
<td>flexible Connecting hose</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Length, mm</td>
<td>5750</td>
</tr>
<tr>
<td></td>
<td>Diameter, inch</td>
<td>$\frac{1}{2}$</td>
</tr>
</tbody>
</table>
B. Force that an average human can apply

<table>
<thead>
<tr>
<th>Exertable horizontal force</th>
<th>Applied with</th>
<th>Condition</th>
</tr>
</thead>
<tbody>
<tr>
<td>110 N (24.7 lbf) push or pull</td>
<td>both hands or one shoulder or the back</td>
<td>with low traction $0.2 &lt; \mu &lt; 0.3$</td>
</tr>
<tr>
<td>200 N (45.0 lbf) push or pull</td>
<td>both hands or one shoulder or the back</td>
<td>with medium traction $\mu = 0.6$</td>
</tr>
<tr>
<td>240 N (54.0 lbf) push</td>
<td>one hand</td>
<td>if braced against a vertical wall 510-1520 mm (20.08-59.84 in) from and parallel to the push panel</td>
</tr>
<tr>
<td>310 N (70.0 lbf) push or pull</td>
<td>both hands or one shoulder or the back</td>
<td>with high traction $\mu &gt; 0.9$</td>
</tr>
<tr>
<td>490 N (110.2 lbf) push or pull</td>
<td>both hands or one shoulder or the back</td>
<td>if braced against a vertical wall 510-1780 mm (20.08-70.08 in) from and parallel to the panel or if anchoring the feet on a perfectly non-slip ground (like a footrest)</td>
</tr>
<tr>
<td>730 N (164.1 lbf) push</td>
<td>the back</td>
<td>if braced against a vertical wall 580-1090 mm (22.83-42.91 in) from and parallel to the push panel or if the anchoring the feet on a perfectly non-slip ground (like a footrest)</td>
</tr>
</tbody>
</table>

Horizontal push and pull forces that can be exerted

Source: Human engineering design data digest department of defense human factors engineering technical advisory group APRIL 2000 Independence Avenue, S.W. Washington, DC 20591
C. Density of different materials

<table>
<thead>
<tr>
<th>Polymers</th>
<th>( \rho ) (Mg/m(^3))</th>
</tr>
</thead>
<tbody>
<tr>
<td>EVA</td>
<td>0.95</td>
</tr>
<tr>
<td>Natural Rubber (NR)</td>
<td>0.94</td>
</tr>
<tr>
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1 For full names and acronyms of polymers – see Section V
(Data courtesy of Grant Design Ltd.)
### Young’s modules of different materials

<table>
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<th>Material Type</th>
<th>Tensile Strain (GPa)</th>
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<tr>
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<td>Carbon Steel</td>
<td>224.5 6.2</td>
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<tr>
<td>Aluminum</td>
<td>224.5 5.7</td>
</tr>
<tr>
<td><strong>Polymer Foams</strong></td>
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<tr>
<td>Polyurethane (PU)</td>
<td>224.5 6.2</td>
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<tr>
<td>Polyethylene (PE)</td>
<td>224.5 5.7</td>
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<td><strong>Thermoplastics</strong></td>
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<tr>
<td>Polyethylene (PE)</td>
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<td>Polypropylene (PP)</td>
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1 For full names and acronyms of polymers – see Section V
(Data courtesy of Grantz Design Ltd.)

**Note:**
- The values for Tensile Strain (GPa) are approximate and may vary depending on the specific material and conditions.
- Polyurethane (PU) and Polyethylene (PE) are typical examples of polymer foams.
- Polyethylene (PE) and Polypropylene (PP) are examples of thermoplastics.

**Table 1:**

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<td>224.5 5.7</td>
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**Note:**
- The values for Tensile Modulus (GPa) are approximate and may vary depending on the specific material and conditions.
- Polyurethane (PU) and Polyethylene (PE) are typical examples of polymer foams.
- Polyethylene (PE) and Polypropylene (PP) are examples of thermoplastics.
### E. Yield stress of different materials

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<td>Rigid Polyurethane Foam (HC)</td>
<td>0.8 - 12</td>
<td>1.2 - 12.4</td>
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</table>

1 For full names and acronyms of polymers – see Section V.

(*) NB: For ceramics, yield stress is replaced by compressive strength, which is more relevant in ceramic design. Note that ceramics are of the order of 10 times stronger in compression than in tension.
F. Statistical analysis of discharge on field test

Statistix 8.0 11/22/2017, 9:21:52 PM

Descriptive Statistics

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<td>SE Mean</td>
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<td>Maximum</td>
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(Data courtesy of Grant Design Ltd)
G. Statistical analysis of discharge uniformity among the nozzles in laboratory test

Result from the 1\textsuperscript{st} 5m test of the six nozzles

Statistix 8.0  
11/22/2017, 9:27:55 PM

Descriptive Statistics

\begin{verbatim}
V001
N 18
Missing 0
Mean 10.684
SD 0.3427
Variance 0.1175
SE Mean 0.0808
C.V. 3.2080
Minimum 10.000
Maximum 11.170
\end{verbatim}

Result from the 2\textsuperscript{nd} 5m test of the six nozzles

Statistix 8.0  
11/22/2017, 9:31:41 PM

Descriptive Statistics

\begin{verbatim}
V001
N 18
Missing 0
Mean 13.554
SD 0.2979
Variance 0.0888
SE Mean 0.0702
C.V. 2.1982
Minimum 13.000
Maximum 14.170
\end{verbatim}

Result from the 3\textsuperscript{rd} 5m test of the six nozzles

Statistix 8.0  
11/22/2017, 9:33:37 PM

Descriptive Statistics

\begin{verbatim}
V001
N 18
Missing 0
Mean 14.150
SD 0.1855
Variance 0.0344
SE Mean 0.0437
C.V. 1.3110
\end{verbatim}
Minimum           14.000  
Maximum           14.500  

**Result from the 4th 5m test of the six nozzles**

Statistix 8.0                                            11/22/2017, 9:34:48 PM

**Descriptive Statistics**

| V001 | 
|------|-------|-------|-------|-------|-------|-------|-------|-------|-------|-------|-------|-------|-------|
| N    | 18    |       |       |       |       |       |       |       |       |       |       |       |       |
| Missing | 0    |       |       |       |       |       |       |       |       |       |       |       |       |
| Mean  | 14.713|       |       |       |       |       |       |       |       |       |       |       |       |
| SD    | 0.1944|       |       |       |       |       |       |       |       |       |       |       |       |
| Variance | 0.0378|       |       |       |       |       |       |       |       |       |       |       |       |
| SE Mean | 0.0458|       |       |       |       |       |       |       |       |       |       |       |       |
| C.V.  | 1.3212|       |       |       |       |       |       |       |       |       |       |       |       |
| Minimum | 14.270|       |       |       |       |       |       |       |       |       |       |       |       |
| Maximum | 14.900|       |       |       |       |       |       |       |       |       |       |       |       |

**H. Discharge pattern of each nozzles with in 20m in each 5m interval**