Regional Review Workshop on Completed Research Activities

Proceedings of Review Workshop on Completed Research Activities of Agricultural Engineering Research Directorate held at Adami Tulu Agricultural Research Center, Adami Tulu, Ethiopia, 17-21 November 2015.

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Oromia Agricultural Research Institute

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Comparative Performance Evaluation of BAMRC Donkey Plough and FARC Single Animal Selected purpose

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Abstract

Comparative performance of two single donkey ploughs types (BAMRC and FARC) were compared on two soil (sandy loam and Silty Clay) type conditions at Haromaya and kombolcha districts. Under the parameters studied Ploughing width, Ploughing depth, effective field capacity(EEC), percentage of weed inversion were higher for FARC plough type(30.35cm,12.54cm, 34.42h/ha and 89.73%) than BAMRC(19.31cm, 10.54cm,32.39h/ha and 61.83%) respectively under sandy loam soil condition. Working speed and drought force are not significant (P<0.05) difference between two ploughs type. plough type was also depend on soil condition and the two ploughs performs better under sandy loam soil type and this is due to lower bulk density and moisture content and less weed occurrence. Ploughing depth was significantly (P<0.05) deeper on sandy loam soils compared to silty clay soils although than difference no significance differences (P<0.05) were observed in the drought force requirements between the soil types in this instance. Generally; from the result obtained Modified EEC Fadis Agricultural Research Center plough type performs higher result under two soil conditions and lower results obtained from Bako Agricultural Mechanization research Center plough type.

Key words: *drought force, Ploughing width, Ploughing depth, effective field capacity(EEC)*

INTRODUCTION

Statement of the problem

The shortage animal power (DAP), following a succession of drought has increased the use of donkeys in crop production activities, including Ploughing. For smallholder farmers, particularly those in semi-arid areas, the donkey is the only alternative to cattle for DAP. Donkeys are highly popular as pack-animals in tropical and sub-tropical countries. Recently, the use of donkeys has gone beyond transportation and donkeys are becoming popular as draught-animals in tillage systems (Nengomashal, Pearson, & Smithlt, 1999). It had been emphasized earlier (Fielding, 1987) that donkeys are efficient in tillage, especially in fields with light soils.

In the recent past, draught animal power has increased in importance in Africa, particularly in Ethiopia, as higher level technology, such as tractors, has become less appropriate economically and due to in appropriate topography of the farming land. Therefore animal power is becoming increasingly economically viable for maintaining sustainable production. The quality of a technological innovation may be measured on the basis of its: (a) suitability for family ownership; (b) requirement of lower energy; (c) generation of self-sufficiency and (d) promotion of soil and water conservation in the tillage system.

Preliminary previous studies (Crossley and Kilgour, 1983) reported that the field work rate of a man is about 75- 125 h ha^{-1} and this could be reduced to 25 h ha^{-1} if a pair of oxen was used. However, it was argued that the latter figure could be further reduced to 14 h ha^{-1} for a field of relatively loose soil. It has also been reported (Giles, 1975; Starkey, 1986; Starkey et al., 1989) that a pair of draught oxen has the potential to work about 4ha per season. Animal-drawn ploughs are known and used on small scale farms in Ethiopia for both primary and secondary tillage. The Ethiopian *maresha* is not only old, it is highly persistent. To the present day, it is almost universally used by smallholder farmers, for the cultivation of the tef grain crop. Although a variety of development programmers has attempted to introduce short-beamed steel plows for the past fifty years, there has been almost no adoption of these. The plowing animals are generally oxen, yoked in pairs with withers yokes, and controlled by a single person. Where oxen are in short supply, horses, donkeys or cows may be used, but oxen are the work animals of choice. Camels are occasionally used for cultivation

There are various factors that may be responsible for the late adoption of plows in Hararghe area. In much of the area, different tribal groups have specialized in animal-rearing and in crop production. Thus many crop-growing farmers did not own potential work animals. Moreover, many traditional farming systems have been based on bush-fallow rotations and inner chat mixed farming.

The chat plantation was above perennial aged or stationary plant that planted in the ridge. Therefore they do not need to till their land with a plow or they prefers to plough their farm in between their chat plant by hand or 'Akafa". because the chat plant at the time of Ploughing the oxen and the long pulling beam was broken their chat crop. Hence it is difficult in case of turning in between this plant. for this reason a simple digging implement is appropriate. Provided the fallow periods are long, such systems can be quite productive in terms of yield per unit of human labor. So the simple and standardized single animal drawn that fit the farmers need and, that becomes justified to the farmers' problem was not adopted yet.

So based on the farmer's opinion during the demonstration of ARDU and AIRIC plough of previous activities, which is appropriate to plough in between their cash crop that is chat. Beside the above justification most of the Hararghe farmers have no pair of oxen to plough their farm lands and use their ox for fattening. So this single animal drawn plough can solve their problem of Ploughing in between chat and normal farming of the land. To solve this problem stated above Bako Agricultural Mechanization research center and Fadis Agricultural research center improved and developed single animal plough in the center and promoted to their area. However the performance evaluation of these two ploughs was not evaluated under different soil condition scientifically. Therefore; this activity was initiated in order to compare and evaluate the performance of Bako type and Fadis type single animal drawn plough using donkey animal.

Objective

- To compare, and evaluate of BAMRC donkey plough and FARC single animal drawn plough for selected purposed

Material and Method

Description of Animal used

Male donkeys with age of 4 and 5 years and live weight of 170kg and 175kg and health were used. Donkeys were trained to pull the plough on neighboring field for about 2days in Ploughing prior to the start of the study to ensure they were accustomed to work.

Ploughs used in the field test

The ploughs used for the field tests were BAMRC donkey mouldboard plough type manufactured by Bako Agricultural Mechanization research center and modified EEC type of FARC single animal plough type manufactured by Fadis Agricultural research center developed by the Department of Agricultural mechanization research process of the Agricultural Research Institute of Oromia. The BAMRC donkey plough type had a mass of 5.2kg and modified EEC type of FARC single animal plough type had a mass of 4.5 kg. And some structural difference between the two ploughs and shown in the figure below

Figure 1: BAMRC type plough type   Figure 2: modified EEC type FARC plough type

Harnessing used

Breastband harnesses (Barwell and Ayre,1982) was used to attach the donkey to the plough. The back straps on the harnesses was adjustable, to ensure a googd fit on any donkeys regardless of size. Donkey was attached to the plough via traces from each side of the breast band harness to swingle 30x40mm rectangular metal beam. And the beam attached to the harness through 12mm diameter and 1.80m long round bar (tondeno). The type of breastband harness was manfuctured and produced by FARC.

Figure 3. FARC single donkey breast band harness and during operation

Soil data

Soil samples were collected from the study area at a depth of 0 – 15 cm and 15 – 30 cm at the experimental site to determine selected soil properties including soil texture, moisture content, and bulk density. The collected samples were transferred to plastic bags and weighed by sensitive balance and taken to soil laboratory. For determination of bulk density undisturbed soil samples was used whereas disturbed soil samples by using auger were taken for analyzing the rest of soil properties mentioned above. The donkey ploughed on two soil types: silty clay and sandy loam soil.

Experimental Sites

The experiment was conducted at Damota kebele of Harmaya districts and xiya kebele of kombolcha districts on farmers' field for two consecutive years of 2006-2007 E.C. From each district two experimental sites (farmer's field) were selected.

Measurements: an ergometer was used to measure the average drought force exerted by the animals to pull the plough and the distance covered while working. The force measured using suspended and two sided hook spring balance. The two spring balance was attached between the implement and the animals so that the force exerted by the animals passed through it in a straight line. Speed of working, Ploughing depth (depth of cut of ploughshare), Ploughing width (width of cut from landslide to furrow edge), total area ploughed, effective field capacity (EFC; h/ha) and soil moisture content were also recorded. EEC (h/ha) was calculated from the total area worked in ha and the total work time taken from the start of the first furrow to the end of the last furrow.

Procedures and measurements

The area ploughed and the time spent working by the donkey depended on the farmers' Ploughing programme on the day of recording. Breastband FARC type harnesses was used. In both districts, the farmers' handled the animals at all times and were encouraged to follow their normal Ploughing and handling procedures.

Experimental design and treatments

The treatments used during the experiments were two plough type and soil type.

1: BAMRC plough type and Modified EEC FARC plough type

2: Soil texture: silty clay and sandy loam and this treatment layout in completely randomized design and replicated three times

Table 1. Treatment combination

	Plough type	
Soil type	BAMRC(P1)	FARC(P2)
Sandy loam(S1)	S1xP1	P2XS1
Silty clay(S2)	P1xS2	P2XS2

Note that: where S1: designated for Sandy loam soil type; S2: designated for Silty Clay soil type P1: designated for Bako Agricultural Mechanization Research Center plough type; P2: designated for Fadis Agricultural Research Center modified EEC plough type

Data collected during performance evaluation of single animal plough

- Soil moisture
- Soil texture
- Ploughing depth
- Ploughing width
- Total area of Ploughing
- Travelling speed
- Draft force
- Time spent turning at the head lands
- Time spent for any reason
- Total operating time
- Bulk density
- Weeds in m*m before and after test
- Easy of handling
- Easy of adjustment
- Visible deformation

Data analysis

- Analysis of variance was carried out using Genostat software and factorial experiment with Randomized complete block design was used.

Results and Discussion

Table 2: Soil physical characteristics of the study area

Place of Study area	Soil texture classes (0-30cm depth)	Soil bulk density (g/cm3)	Moisture content of the soil (% in volume basis)
Harmaya districts, Damota kebele	Silty clay	1.25	5.1
Kombolcha districts, xiya kebele	Sandy loam	1.12	4.33

There were no significant differences ($P>0.05$) between the donkey of 170kg and 175kg in the draught forces exerted to pull either plough type.

Plough types.

The drought force required to pull the ploughs which was higher ($P<0.05$) for Bako type plough than for Fadis modified EEC single donkey type, the parameters (Ploughing width, Ploughing depth, percentage of weed inversion and effective field capacity) for FARC plough type were higher than BAMRC plough type whereas working speed of FARC plough type was lower than BAMRC plough type, and there were highly significant differences (P> 0.05) in Ploughing depth, percentage of weed inversion and effective field capacity for FARC plough type than BAMRC plough type and detail results were presented in table below.

Table 3: Performance result of two plough type under sandy loam soil conditions at study area

Sandy loam	**BAMRC type plough**	**Modified EEC FARC type plough**
Ploughing depth(cm)	10.54	12.38
Ploughing width(cm)	19.31	30.35
Drought force(N)	209	199.7
Working speed(m/s)	0.72	0.69
Effective field capacity(h/ha)	32.39	24.43
Percentage of weed inversion	61.83	89.73

Table 4: Performance result of two plough type under Silty clay soil conditions at study area

Silty clay	BAMRC type plough	Modified EEC FARC type plough
Ploughing depth(cm)	9.96	12.25
Ploughing width(cm)	17.3	29.38
Drought force(N)	204.7	200.7
Working speed(m/s)	0.72	0.71
Effective field capacity(h/ha)	34.42	26.56
Percentage of weed inversion	58.17	85.13

Table 5. Comparative performance of two plough type under two soil condition

Treatments	Ploughing width (cm)	Ploughing depth (cm)	Working speed (m/s)	Drought force (N)	Effective field capacity (h/ha)	Percentage of weed inversion (%)
Sandy loam x BAMRC plough type	19.31[a]	10.54[a]	0.72[a]	209.00[a]	32.39[a]	61.83[a]
Silty clay x BAMRC plough type	17.30[b]	9.96[b]	0.72[a]	204.67[b]	24.43[b]	58.17[b]
Sandy loam x FARC plough type	30.35[c]	12.38[c]	0.69[b]	199.67[c]	34.42[c]	89.73[c]
Silty clay x FARC plough type	29.38[d]	12.25[c]	0.71[ab]	200.67[c]	26.56[d]	85.13[d]
mean	24.09	11.28	0.71	203.50	29.45	73.72
CV	5.14	2.55	4.48	2.55	1.71	2.86
LSD0.05%	0.70	0.16	0.02	2.79	0.27	1.13

Note: Treatment means followed by the same letter(s) a. b: c and d are not significantly different at a given level of significance.

Comparative performance evaluation of two ploughs on two soil type condition

Ploughing Width

From the result and data recorded; Maximum Ploughing width (30.35cm) was recorded on FARC plough type under sandy loam soil type whereas; minimum Ploughing width (17.30cm) was recorded on BAMRC plough type under Silty clay soil type. There is highly significance difference ($P<0.05$) between the treatments on Ploughing width and higher Ploughing width were recorded on FARC plough type under two soil conditions.

Ploughing Depth

From the analysis of data recorded; Maximum Ploughing depth (12.38cm) was recorded on FARC plough type under sandy loam soil type whereas; minimum Ploughing depth (9.96cm) was recorded on BAMRC plough type under Silty clay soil type. There is highly significance difference in Ploughing depth ($P<0.05$) between the treatments except on FARC plough type under two soil conditions and higher Ploughing depth were recorded on FARC plough type under two soil conditions.

Working Speed

There is no significance difference ($P<0.05$) in working speed between treatments. Higher working speed (0.72m/s) was recorded under BAMRC plough type and minimum working speed (0.69m/s) was recorded under FARC plough type.

Drought force

From the analysis of data recorded; Maximum Drought force (209N) was recorded on BAMRC plough type under sandy loam soil type whereas; Minimum Drought force (199.67N) was recorded on FARC plough type under Sandy loam soil type. There is no significance difference in Drought force ($P<0.05$) between the treatments except on BAMRC plough type under two soil conditions and higher Drought force were recorded on BAMRC plough type under two soil conditions.

Effective Field Capacity

From the analysis of data recorded; Maximum Effective Field Capacity (34.42h/ha) was recorded on FARC plough type under sandy loam soil type whereas; minimum Effective Field Capacity (24.43h/ha) was recorded on BAMRC plough type under Silty clay soil type. There is highly significance difference in Effective Field Capacity ($P<0.05$) between the treatments under two soil conditions and higher Effective Field Capacity was recorded on FARC plough type under two soil conditions.

Percentage of weed inversion

From the analysis of data recorded; Maximum Percentage of weed inversion (89.73%) was recorded on FARC plough type under sandy loam soil type whereas; minimum Percentage of weed inversion (58.17%) was recorded on BAMRC plough type under Silty clay soil type. There is highly significance difference in Percentage of weed inversion ($P<0.05$) between the treatments under two soil conditions and higher Percentage of weed inversion was recorded on FARC plough type under two soil conditions.

Discussion

From the result obtained Modified EEC Fadis Agricultural Research Center plough type performs higher result under two soil conditions and lower results obtained from Bako

Agricultural Mechanization research Center plough type and may be this is due to different design of the plough type; and plough type was also depend on soil condition and the two ploughs performs better under sandy loam soil type and this is due to lower bulk density and moisture content and less weed occurrence.

The lighter FARC plough required less drought force to operate it. Whether the differences in drought force between the two plough types can be attributed to differences in the weights alone, is not known. Implement manufacturers do not always regard the weight of the plough as the most important factor in drought requirement(Inns, Shetto, & Mkomwa, 1840). However in the present study lightening the weight seems to have been an effective means of reducing the drought force of the implement bringing it within the donkeys.

The predominant soil type in the smallholder farming areas in study areas are sandy loam and silty clay. Theoretically, the physical structure of sandy loam soils should allow deeper penetration of the plough than on silty clay soils. This was the case in present study where Ploughing depth was significantly ($P<0.05$) deeper on sandy loam soils compared to silty clay soils although than difference no significance differences ($P<0.05$) were observed in the drought force requirements between the soil types in this instance.

Farmers (and their visiting neighbors) were initially skeptical of the work potential of such a small and light plow but, at the end of some hours work, during which the plow was tried by a number of people, they were satisfied (and somewhat surprised) with the quality and quantity of its output.

Ease of adjustment and use

All farmers appreciated the lightness of the two plow types which made it easy to control. This was an added advantage when turning as the plow could be easily held out of work. The adjustments on the plow were few and the farmers felt that they could easily manage them. The handle allowed operators of various heights to work comfortably without too much bending. The smaller handles enabled a better hand grip making the plow even more controllable, as it could be held firmly and comfortably.

Many farmers observed that BAMRC plough type a little bit heavier than FARC plough type and width of cut of the soil and tilt angle from the landslide is smaller in BAMRC plough due to this the performance is lower than FARC plough type and farmers select FARC plough type

Performance

The quality of work was satisfactory, with good inversion of the soil and trash coverage, especially where the soil was relatively moist. However, where operators were unskilled and donkeys were not well trained there was poor overlapping of the furrow slices. Training and experience of donkeys and farmers should overcome this problem.

Many farmers observed that BAMRC plough type the field capacity (rate of work) was rather lower because of the small width of cut. This necessitated more passes and hence more time to plow an acre compared to than FARC plough type

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Development and Evaluation of Animal drawn Wheat Row Planter

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Abstract

The planting operation is one of the most important cultural practices associated with crop production. Increases in crop yield, cropping reliability, cropping frequency and crop returns all depend on the uniform and timely establishment of optimum plant populations. The basic objective of planting operation is to put the seed in rows at desired depth and spacing, cover the seeds with soil and provide proper compaction over the seed. Three different Animal drawn wheat row planter version has been developed and evaluated under farmers condition. The last version (recommended one) has four rows with fertilizer application system. The test was conducted in two seasonswith three replications with different soil types. The planter performance were: average seeding rate 111kg/ha, average time required to plant one hector 8.33hours, depth of planting obtained 5cm, plant population 1.2million plant per hectorand average yield obtained was 45kuntal/hector.The labor required per operation is one person. Compared to traditional wheat row planting practice, the technology shows advantages of labor saving, time and seed savings.

Keywords: *Planting time, Plant population, Uniformity, Depth of planting, Yield, Seed rate*

Introduction

Agriculture is the oldest and most important avocation of the world. Tillage and planting methods are one activity of agriculture that reduce soil loss through wind and water erosion as well as water runoff produce or tolerate non-erodable obstacles on the soil surface such as stable soil aggregates (clods) and organic material (mulch). But rough soil surfaces with imperfectly incorporated plant residues place great demands on planting techniques. Also Labour requirements with regard to hand planting increases proportionally to the planting rate. For timely planting, farmers tolerate low (erosion-susceptible) plant densities and improper seed embedding. This results in low yields, which in turn affects farmers' willingness to adopt innovations. Because of agricultural production is mainly dependent upon the maintenance and improvement of soil productivity. So farmers should be educated to use lands according to their capability and to adopt proper soil conservation measure.

In agricultural activity the problem of mechanizing the planting process in small- scale agriculture remains largely unsolved worldwide. It is assumed that in Africa on average not even one farmer out of 1,000 is using a planter on the animal traction or tractor mechanization levels. From that Ethiopia is one country. Our country is one of the developing country with large population that depending on agriculture. Agriculture plays an important role in the country's economy. Though the sector is the main income generating and the largest labor intensive sector it is the least productive which it could not even support the food demand of the farmers who are engaged in the sector .Because of this sector affecting crop production system, internal factors (genetic or hereditary) such as seeds varieties and external factors for instance precipitation, temperature, atmospheric humidity solar radiation and so on.

Among the agricultural disciplines agricultural mechanization is the one which needs special attention. It is because to exploited the maximum productivity of the different crops the seed need to get the appropriate germination and growing conditions so that the crop can perform best giving maximum yield. It is not only in pre-harvest activities but also in post harvest activities the production of the crop should not be lost because of inefficient harvesting and post harvest handling. Therefore developing, evaluating inter-mediate agricultural mechanization technologies should be adapted to sector.

Problem statements

Wheat row planting is becoming common practice since four to five years back in Ethiopia. It is believed that seed row planting facilitates further managements (such as weeding, cultivating) of crops to increase productivity. Practically farmers are witnessed the productivity increment achieved so far. But in all over the country, small farmers are using traditional practice of row planting where they use their figure and other locally available material for metering. In doing so, the person who place seed in row suffers heavy back-ache/pain which hinders farmers to adopt this practice as needed though it has significant benefit.

To ease the hardship in wheat row planting practice, it is fundamental to develop and evaluate appropriate animal drawn wheat row planter for small scale farmers.

Objectives:

1. To design and Develop animal drawn wheat row planter
2. To evaluate performance of the machine under farmers condition

Experimental Details

Metering system selection

Phase I: Gravity system

Wheat row planter which has eight rows with gravity system was developed. Agitator mechanism was mounted on shaft inside the hopper where the shaft joins the two wheels together. As the oxen pulls, force is transferred through ground wheel. The agitators are intended to facilitate flow of seed through extension of the hopper.

This wheat row planter was tested on field. While on test, seed flow continuity problem and lack of control mechanism was observed. Because of these problems this version of technology was rejected by the team.

Phase II: Fluted roller system

In this phase of mechanism selection, the design concept was taken from single axil tractors' planting implement. In this phase of technology development two prototypes of wheat row planters were manufactured. The first without fertilizer applicator and the second with fertilizer applicator was developed. The mechanisms were the same except difference in fertilizer application. This mechanism resolves the previous problems. And this type of planter is known as seed drill machine.

Technology description

Power source: a pair of animal
Weight: 64kgkg
Overall Width:
Number of rows: Four with fertilizer application
Row spacing: 25cm

Components of the technology.

It is a seed drill with mechanical seed metering device which consists of: (i) Frame (ii) Seed box (iii) Seed metering mechanism (iv)drive transmission system v) Furrow openers (vi) Covering device and ix) Transport wheels.

Frame

The frame is made of mild steel angle section and flats. It is made in such a way that strong enough to withstand all types of loads in working condition. All other parts of a seed drill are fitted to the frame.

Seed box

It is a box like structure made up of steel sheet metal with eight compartments for seed and fertilizer. Seed metering mechanism is placed at the bottom of the box.

Seed metering mechanism

The mechanism which picks up seeds from the seed box and delivers them in tothe seed tube is called *seed metering mechanism*. It is Fluted roller feed type and provided at the bottom of the box.The fluted roller is driven by a circular shaft. There are horizontal groves provided along the outer periphery of the rollers and rollers can be shifted sideways depending upon the seed rate. These rollers are mounted at the bottom of the seed box. They receive the seeds in the longitudinal groves and pass on to the seed tube through the seed hole. The fluted rollers are placed inside housing made up of Aluminum and galvanized water pipe respectively, the shaft is steel shaft.

Drive transmission system

The drive transmission mechanism consists of a sprocket-chain assembly and a driven shaft thatcarry the seed picking discs. The chain connects the drive shaft sprocket and the driven shaft sprocket. As the drive shaft which is connected to transport wheel rotates, the driven shaft which carries the seed metering discs rotates and picks up.

Furrow openers

These are the parts which open up furrows in the soil for placing the seeds. It is shovel structure made up of sheet metal and flat iron.

Covering device or furrow closer

It is a device which closes the furrow with soil after the seed has been dropped in it. The covering device is made in straight bar mode which is connected to frame at the back.

Transport wheel

There are two wheels fitted on an axle for transporting the drill on roads. Flat iron wheels and or pneumatic tire are used as transport wheels. The transport wheels are fitted with chain and sprocket attachment to transmit the motion of the wheel to the seed metering mechanism when the drill is in operation.

The first prototype

The second prototype

As seen from the picture, the first planter is smaller than the second planter because of space requirement to include fertilizer application.

Parameter Description and Data Collection

The data were collected from the parameters as described below:

1. **Planting time**: The time taken for sowing by animal drawn planter and manual hr/ plot
2. **Plant population**: The number of plants in plot area
3. **Uniformity**: The percentage of even distribution of plant /plot. The row spacing and plant population in a row are treated here.
4. **Depth of planting**: The depth of the hole was measured by ruler (cm)

5. **Yield**: The wheat on each plot were harvested, dried, cleaned and weighted then adjusted to kg per hectare.
6. **Seed rate**: amount of seed required to cover one hectare in kg per hectare
7. **Labor Requirement**: it is the number of person required on operation/during planting time (in person per operation)

The data collected at different sites are presented in the following table. The experiment was done in one season.

Table 1: Data taken from each site

No	Parameters	Site 1	Site 2	Site 3
1	Seed rate (kg/ha)	118.52	106.7	109
2	Planting time (hrs/ha)	10	6.67	8.33
3	Plants per meter of row (plants/meter)	Sr1:p1=52, p2=42	Sr1:p1=50, p2=58	Sr1:p1=53, p2=59
		Sr2:p1=63,p2=81	Sr2:p1=48,p2=43	Sr2:p1=36,p2=33
		Sr3:p1=49,p2=49	Sr3:p1=60,p2=79	Sr3:p1=44,p2=43
		Sr4: p1=66,p2=65	Sr4: p1=47,p2=43	Sr4: p1=61,p2=47
4	Yield kg/hectare	4140	4550	3500
5	Depth of planting	3-5cms	4.5-5cms	3-5cms
6	Soil type	Medium	Light soil	Medium

Srn (where n is 1, 2, 3, 4)-is sample from row n and p1 and p2 are plants per meter of a row at position 1 and 2. On this data between row distance was taken constant as per the design (i.e 25cm).

Result and Discussion

Plant population

Chances of optimal yields are improved by plant population. Even though the optimum is dependent on variety, there is recommended number of plant per hectare of land (i.e. 700000-1000000).

As presented in table 1; plant population is between 72 and 47 plants per meter of row for site 1, between 69.5 and 45 plants per meter of row for site 2 and between 56 and 34.5 for site 3. Plant population is not comparable with standard of row planted wheat as per information from http://www.daf.gov.au/wheat/planting-information. From this data it is vital to modify the technology for these requirements.

Distribution Uniformity

Distribution uniformity indicates variation in delivery betweenopeners. The coefficient of variation (CV) is a mathematical termused to describe distribution uniformity.

$$\mathbf{CV = (StDEV\ sample) * \frac{100}{Average\ sample}}$$

Where: CV is Coefficient of Variation

StDEV-is Standard Deviation of Sample data and Average sample is arithmetic average of the sample data taken.

The interpretation of coefficient of variation is as characterized by PAMI (Prairie Agricultural Machinery Institute it is Canadian Company working on machinery research) has accepted the following scale as its basis for rating distribution uniformityof seeding implements for wheat crop:

CV greater than 15% -- unacceptable

CV between 10 and 15% -- acceptable

CV less than 10% -- very good

CV less than 5% -- excellent

Table 2: shows calculated Coefficient of variation (CV) for the three sites and for all data too:

	Site 1	**Site 2**	**Site 3**	**Of all data**
StDEV	12.45	12.06	10.03	12.1
Sample Avg	58.375	53.5	47	52.96
CV (%)	**21.7**	**22.5**	**21.5**	**22.9**

As per PAMI, distribution uniformity obtained by this technology is unacceptable for wheat crop.

Planting time

The planting time is one of the requirements of mechanization technologies. As indicated in the table, the projected planting time per hectare for each sites are 10h. 6.67 and 8.33. While in traditional wheat row planting time is between 32 and 36hr/ha based on the farmer and oxen. Comparing the two methods (row planter and traditionally row planting), row planting technology saves up to three folds of planting time. As seen it saves significant amount of time for farmers to use for other productive work.

Labor requirement

Using anima drawn planter requires one person for operation. In case of traditional method of wheat row planting totally three persons are required per single operation (one for furrow opening, one for dropping seed and one for dropping fertilizer in the opened furrow). Comparing man power required for specific operation animal drawn planter reduces man power from three to one which is very significant for a single operation. Beside the number, the operation hard ship related to traditional row planting is totally removed which will facilitate the adoption of the practice.

Depth of sowing/Planting depth

Optimum planting depth varies with planting moisture, soil type, seasonal conditions, climatic conditions, and the rate at which the seedbed dries. The general rule is plant as shallow as possible provided the seed is placed in the moisture zone but deep enough so that the drying front will not reach the seedling roots before leaf emergence. Optimum planting depth is for wheat is around 50-70 mm. The technology is designed for depth upto 5cm and compared to the design the data is promising. As to the standard, modification of design idea is required.

Seed rate (kg/ha)

Because seed sizes may vary depending on production years and variety type, a fixed quote for the weight of seed needed to plant one hectare is not always a true or accurate measure of obtaining a desired plant population per hectare. An actual seed count is required to calculate a more accurate planting rate. But in this experiment, the characterization was done based on the weight of seed. Taking the average of the three sites, the seed rate of the technology is 111.5kg/ha. Comparing local practice seed rate which is 150kg/ha this technology is relatively advantageous.

Yield (kg/ha)

Yield is a factor variety, weather condition, soil type, management and the like, comparatively, animal drawn planter shows yield improvement. In the season of the experiment yield obtained from area was less than or equal to 4000kg/ha. But using animal drawn and full management package the yield obtained was 41kg/ha. Though it is no significant, still there is yield increment. This yield was more absolute if it was repeated over seasons.

Conclusion

As discussed in result and discussion session above, the technology shows significant advantages on some evaluation parameters and on the others needs more work. Animal drawn wheat row planters are important in time saving, labor saving and improves operation difficulties. Therefore it is advantages for small scale farmers to use this technology.

Recommendation

Even though, this technology is promising in the future to facilitate the land and labor productivity plan of Government the following points needs attention.

- Improvement is required to upgrade precision that is moderating plant population and distribution uniformity.
- Modification of design and change material selection to decrease weight of the technology.

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Performance Evaluation of Animal Drawn Row Planter for Maize, Haricot bean and Sorghum Seeds

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ABSTRACT

This study was undertaken to evaluate the performance of a prototype planter capable of planting maize, haricot beans and sorghum seeds at predetermined seed spacing's and planting depths. The test results indicate good performance for maize, haricot bean and sorghum. The effects of three main factors, three different types of seed, three levels of planter forward speeds, three levels of hopper filling were investigated in a split- split plot design with 3 x 3 x 3 factorial experiment on performance of the metering mechanism to establish the average percent of seed spacing indices distribution by the planter. The pattern of distribution of the seed in the row and the percentage of seed spacing indices were examined. The treatments that were examined consisted of three types of seeds, three levels of planter forward speeds and three levels of hopper filling. The results indicate that the effects of planter forward speeds on seed spacing indices were statistically significant ($p < 0.05$).The mean field capacity, field efficiency, depth were 0.12 ha/hr (8.33 hr/ha), 71% and 4.94cm with coefficient of variation of 0.042 (4.20%), respectively. The planter established acceptable plant population; 7.33, 11 and 15.33 plants within rows of 2 m length for maize, haricot beans, and sorghum, respectively, when compared with design or desired number of 8, 12 and 14 plants with maize, haricot beans and sorghum, respectively. Based on the performance evaluation results, it is concluded that the prototype planter can be efficiently, effectively and economically used by the majority of Ethiopian farmers.

INTRODUCTION

Agricultural mechanization is the use of mechanical devices or systems to replace human muscle in all forms and at any level of sophistication in agricultural production, processing storage and so on in order to reduce tedium and drudgery, improve timeliness and efficiency of various farm operations, bring more land under cultivation, preserve the quality of agricultural produce, provide better rural living condition and markedly advance the economic growth of the rural sector (Anazodo, 1986; Onwualu et al., 2006).

About 69% of the Ethiopian farmers have farmlands less than or equal to one hectare in size and average grain yield for various crop is less than one metric ton per hectare (CSA, 2013). It is very difficult for these farmers to own and operate costly agricultural machinery and equipments that can establish the optimum plant population. Hence, in most part of the country, manual broadcasting method of sowing is still in use. This method of crop establishment adversely affects the seed requirement and production per unit area. The broadcasting method of crop establishment results in improper placement of seed, fails to put the seeds firmly in the soil, leads to uneven placement of seeds at correct interval and exposes seed for consumption by rodents and birds.

As our population continues to increase, it is necessary that we must produce more food within smaller land, but this can only be achieved through some level of mechanization. However, planting machine or planter that is normally required to produce more food is beyond the buying capacity of small holder farmers. These small holder farmers still continue to plant manually, the result of which is low productivity especially in the production of grains. It is therefore necessary to develop a low cost planter that will reduce tedium and drudgery and enable small holder farmer to produce more foods.

Thus, it is important to improve the planting operation by reducing human effort, and increasing stand accuracy and field capacity. Furthermore, developing agricultural sector requires modern techniques and suitable agricultural mechanization which will increase the agricultural production and reduce the level of manual labour, which represents 85% of the total labor force, involved in agricultural sector (CSA, 2013). Nonetheless, Ethiopia as a nation and the Ethiopian farmers, as the major section of the population, lacks the means to establish optimum plant population of almost all crops timely.

The objective of this study therefore was to evaluate the performance of animal drawn two-row planter and recommend for smallholder farmers.

1. MATERIALS AND METHODS

1.1. Description of the planter

The row planter (figure1) was developed at Asella Agricultural Mechanization Research center, Asella Ethiopia. The planter consisted of frame, seed hoppers, metering mechanisms, furrow openers, furrow covering devices and drive wheels

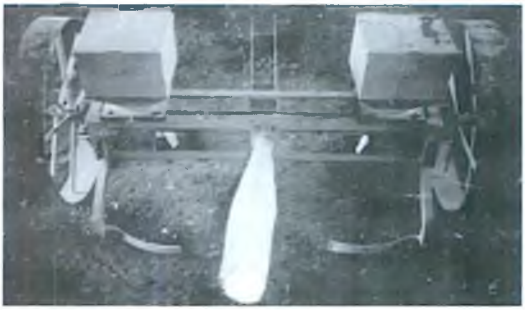

Figure1. Photograph of evaluated planting machine

1.2.Performance Test and Evaluation

Two sets of tests were performed; laboratory investigation to calibrate the machine in terms of seed rate, seed damage, and seed spacing, and field tests carried out to obtain actual overall performance of the machine. The seed variety used for the testing maize (Melkassa-2), haricot bean (Awash-1) and sorghum (Gubiye) seeds were obtained from Ethiopian Seed Enterprise, Asella Branch.

2.2.1 Seed damage test

The test for percentage seed damaged was done with the machine held in a similar position to that described above, with 2.5 kg of seeds loaded into each of the hoppers. A paint mark was made on the drive wheel to act as a reference point to count the number of revolutions when turned, and a polythene bag was placed on the discharge spout to collect the seeds discharged. The wheel was rotated 20 times in turns and the time taken to complete the revolution was recorded with the aid of stop watch. The seeds discharged from the spout were observed for any external damage. Germination tests were also conducted to assess the level of internal damage by the metering mechanism.

2.2.2 Evaluation of Seed spacing/distribution

To assess uniformity of seed placement and seed spacing distribution, the planter was pulled by 15 hp mini tractor at predetermined forward speed and hopper loading capacity on manually dug, leveled, fine sand applied, gently packed and watered soil surface. At the end of each test run, measurement of successive seed spacing was taken to calculate the mean seed spacing, seed miss index, seed multiple index, quality of feed index and precision in spacing were estimated based on the following equations (Kachman and Smith, 1995).

$$MISI(\%) = \frac{n_{III} + n_{II} + n_{I}}{N} \times 100 \qquad 1$$

$$MULI(\%) = \frac{n_I}{N} \times 100 \qquad 2$$

$$QTFI(\%) = \frac{n_{II}}{N} \times 100 \qquad 3$$

$$PREC(\%) = \frac{S_{II}}{X_{ref}} \times 100 \qquad 4$$

Where: MISI = miss index

MILI = multiple index

QTFI = quality of feed index

PREC = precision index

x_{ref} = Theoretical (desired) seed spacing

n_I = number of seed spacing's less than or equal to half of the desired spacing (< 0.5 x_{ref}).

n_{II} = number of seed spacing's close to the theoretical seed spacing (0.5 to 1.5$x_{ref.}$).

n_{III} =number of seed spacing's greater than 1.5 times the theoretical seed spacing (>1.5 $x_{ref.}$).

2.2.2. Field Test

A plot of 180m^{2}area was marked out on the field. The plot was prepared under a conventional tillage system to obtain a fairly flat field. Field tests were conducted to determine the effective field capacity and efficiency and average depth of placement of seeds. Investigation into the field efficiency and effective field capacity of the planter involved continuous observation and timing of each activity involved in the planting operation. Two stop watches were used to estimate activity time, while four people were involved in the determination of field efficiency. One stop watch was used to record the time losses such as those for turning at field ends, removal of clogs, and adjustment. The other stop watch was used to continuously measure the time during which the planter actually performed the intended operation i.e. time for actual planting operation. Field efficiency and effective field capacity was determined using the relationship by Kepner *et al* (1987)

$$V = \frac{D}{t_a} \qquad 5$$

Where: - V = Working speed,

D = distance of run (m)

t_a = average time of each pass (second)

$$e = 100 \times \frac{T_e}{T_t} \qquad 6$$

Where: - e = field efficiency (%).

T_e= effective operating time (sec.)

T_t=total time= (effective operating time + time lost for turning)

$$C_e = \frac{W_e \times S_{mf} \times e}{10} \qquad 7$$

Where: - Ce = effective field capacity (ha/hr)

W_e = implement effective width/inter row spacing (m)

S_{mf} = mean forward speed (km/h)

e = field efficiency (decimal)

3.Results and Discussions

3.1 Seed damage test

The numbers of mechanically damaged, i.e. bruised, skin removed or crushed seeds were counted and their percentage was computed. The mean percent seed damaged for maize, haricot bean and sorghum seeds were found to be 1.51±0.52%, 1.28±0.34% and 1.95±0.40%, respectively. Seed damage increased with increasing metering roller speed and this was attributed to shearing and jumping of seeds against the wall of the hopper at high speeds and the magnitude of damage was depended on the strength of the seeds.

The results of germination test revealed that mean percentage germination of 97.38±0.44, 83.61±0.37 and 80.46±1.36 for maize, haricot bean, and sorghum, respectively. A difference between the percentage germination rate before and after metering the seeds can be related to the metering rollers quality, the variability of the seeds and the friction between the metering device and seeds. The reduction in percent germination in maize, haricot beans and sorghum were 0.68, 1.39 and 0.54%, respectively, indicating that the mechanical damages caused by the metering devices on the respective seeds were within acceptable level, less than one percent, except that of haricot beans which was 1.39 % indicating the need for improvement of the metering device to safely handle haricot bean seeds.

3.1. Seed spacing distributions

Performance indicators such as spacing indexes that include miss index (MISI), multiple index (MULI), quality of feed index (QTFI) and precision (PREC) were used to assesses functional fulfillment of the prototype planter.

3.1.1. The seed miss index

Experimental results show that operational speed had significant effect on percent miss index. However, the level of effect varied with crop type. Increasing speed of operation from 3 km/hr to 7 km/hr had increased the percent miss index when planting maize, while the effect of speed increase from 3 km/hr to 5 km/hr when planting haricot beans and sorghum did not have significant effect on percent miss index. In general, increase in operational speed tended to increase percent miss index when planting maize, haricot bean, and sorghum seed using the prototype planter. As figure 2 clearly indicated forward speed greater than 5 km/hr would result in percent miss index of approximately equal to ten and above which exceeds the acceptable level of percent seed miss index.

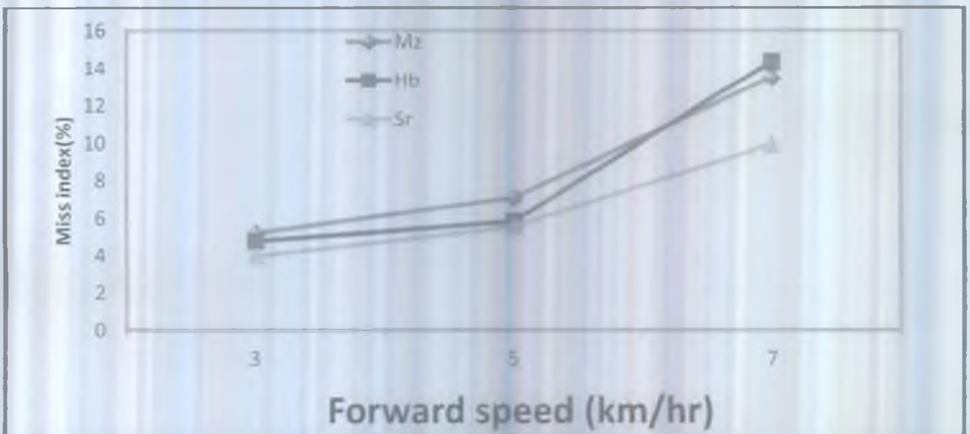

Figure 2.Effects of planter forward speeds on seeds miss index.

3.1.2. The seed multiple index

Planter forward speed, level of hopper fill and the combined effect of the two had no significant effect on mean percent multiple indexes of maize and sorghum, but the effect on mean percent multiple index was significant difference for haricot bean between speeds of 3 and 5 km/hr and that of 7 km/hr. The combined effect of level of hopper filling and forward speed on percent seed multiple indexwas also had significant effect ($P < 0.05$) for all seeds at forward speed of 7 km/hr. The highest percent seed multiple index occurred at planter forward speed of 5 km/hr for maize, haricot bean and sorghum. The lowest values, of the same, were recorded at planter forward speed of 7 km/h.

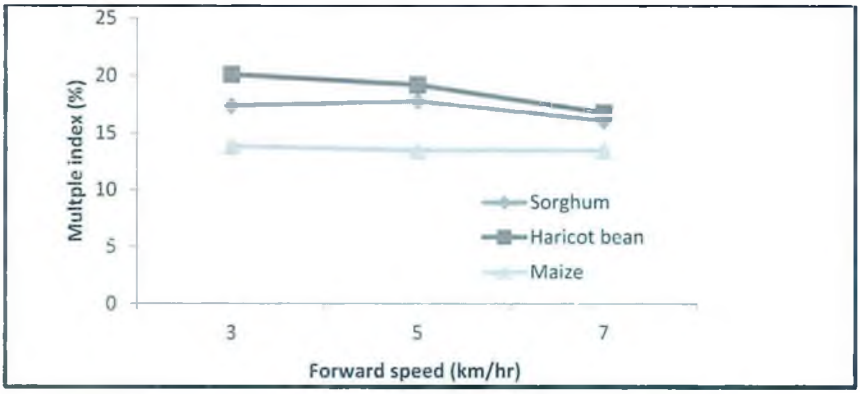

Figure 3. Effects of planter forward speeds on seeds multiple indices.

3.1.3. The quality of seed feed index

Forward operational speed of the planter and seed types had significant ($P < 0.05$) effect on the quality of seed feed index (percent cell fill) at speed of 7 km/hr regardless of the type of seeds used. The effect of percent or level of seed fill in the hopper was not statistically significant ($P > 0.05$) on the quality of seed feed index. This clearly indicated that the forward speed of the planter, i.e. the speed of the metering mechanism, had detrimental effect on the percent quality of seed feed index (percent cell fill).

As we can see from figure 4, the highest percentquality of seed feed indexes of 80.91, 75.09, and 78.65% were observed with maize, haricot beans and sorghum, respectively, when the planter was operated at the forward speed of 3 km/hr. The lowest percent quality of seed feed index of 73.01, 68.86 and 73.00 were observed with maize, haricot beans and sorghum, respectively; when the planter was operated at linear speed of 7 km/hr. From these, it could be concluded that operating the planter at speeds greater than 5 km/hr would reduce the plant population/ ha, hence could lead to reduced harvest at the end of the day.

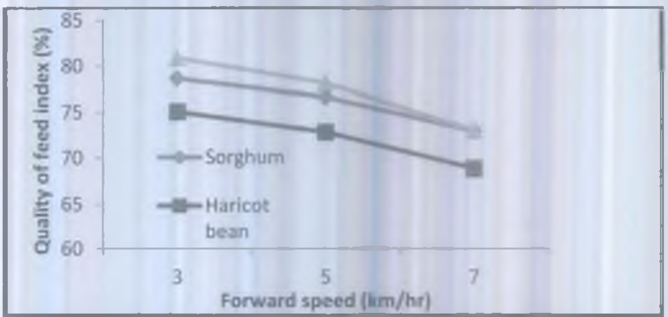

Figure 4. Effects of planter forward speeds on quality of seed feed indexes.

3.1.4. The seed precision index

Analysis of the effect of operational speed of the planter, hopper fill level and seed types on percent seed precision index (percent spacing variation) indicated that only planter forward speed had significant (P < 0.05) effect on the percent seed precision index; means that variation, in seed spacing within a row, increases as planter linear forward speed increases, i.e. as the speed of operation is increased, one should expect high variability in seed spacing, which is not a desired trait. Nonetheless, level of seed in the hopper did not have significant (P > 0.05) effect on the percent seed precision index. The combined effect of seed level in the hopper and forward speed of the planter, on the percent of seed precision index, was significantly difference (P < 0.05) at the planter forward speed of 7 km/hr only. Figure 5. clearly indicated that planter linear forward speeds greater than 5 km/hr would result in seed spacing variations of over 13%, though the variations for maize and sorghum were greater than 15 and 17%, respectively at plant speed of 5 km/hr.

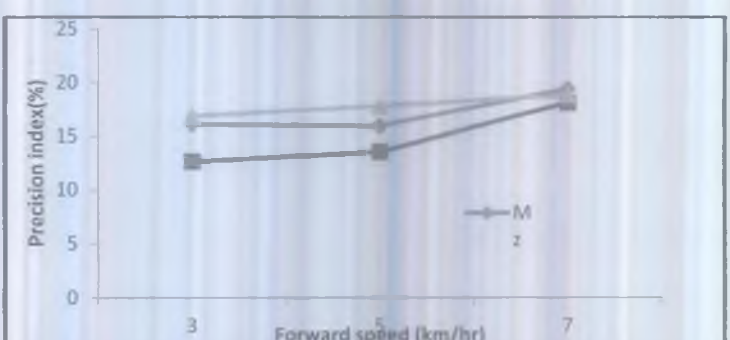

Figure 5. Effects of planter forward speeds on percent seed precision index.

4.2 Field efficiency and field capacity of the planter

The mean field capacity and efficiency of the prototype planter were 0.12 ha/hr (8.33 hr/ha) and 70.57%, respectively. This shows that the prototype planter can plant a hectare of land in slightly over eight working hours. In another words, the planter can best suit majority of the Ethiopian farmers. The field efficiency of the prototype planter agreed, as recommended Kepner*et al.* (1978) for planters, is within the acceptable level.

5.2 Conclusion

From the forgoing summary, one can see that the performance of the prototype planter in terms of percent seed miss index (MISI, % seed skip), percent seed multiple index (MULI, % redundancy), percent quality seed feed index (QTFI, % cell fill), percent seed precision index (PREC, percent spacing viability), depth of planting, plant count/stand (achievement of the optimum plant population), field capacity, field efficiency, labour cost and economics of owning and operating the machine is acceptable. Above all, the prototype planter successfully handled seeds of three different crops, maize, haricot beans and sorghum seeds. Hence, it can be concluded that the prototype planter can be efficiently, effectively and economically used by the majority of Ethiopian farmers. However, the speed of the planter should not exceed 5 km/hr in order not to seriously and negatively affect the percent cell fill or the recommended plant population.

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DESIGN AND DEVELOPMENT OF TEFF CLEANING MACHINE

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Abstract

This paper deals with the design and development of a tef grain and chaff separating and cleaning machine and presents a review of former needs/approaches. The performance was evaluated in terms of separating efficiency, cleaning efficiency, separation loss and cleaning loss at different sieve slopes (0,5 and 10°), sieve oscillations (5,10,15 and 20Hz) and feed rates (3,6,9 and 12 kg/min). The separation and cleaning losses increased with increasing sieve oscillation and feed rate for all sieve slopes.

Background and justification

Agricultural production, productivity and its sustained development depend on the advancement of science and technology that will enhance production, processing, handling, transportation, distribution and marketing processes of strategically important/economical crops. One of the economical cereal crops in Ethiopia is tef. It is indigenous to the country, and is a fundamental part of the culture, tradition, and food security of the people. This crop is gaining international recognition and acceptance, and is a means of foreign currency earning in addition to its value as food and industrial crop at home. Currently, tef is grown on approximately 2.80 million hectares of land which is 27% of the land area under cereal production. Tef accounts for about a quarter of the total cereal production and is highly economical food grain in Ethiopia. Approximately, 6 million households grow tef and it is the dominant cereal crop in 30 of the 83 high-potential agricultural woredas (Bekabil *et al.*, 2011).

The traditional methods of harvesting, threshing and postharvest handling of tef usually lead into contamination of the product with stones, sticks, chaff, dirt and dust. Materials obtained after threshing include long straws, chaff, small fragments of spikes, leaves and grains. Therefore tef grain, after threshing cannot be stored or used for consumption or as planting material due to the very fact that the presence of long straws, chaff, small fragments of spikes, leaves, dust, dirt and other foreign materials in the grain will accelerate deterioration, thus lead to poor physical condition and quality of grain becomes eminent. As a result, farmers are compelled to do additional work of separating and cleaning grains from undesirable materials that will otherwise reduce the quality and the value of the product prior to storage, marketing, distribution and subsequent processing. Tef winnowing, separation and cleaning i.e. removal of undesirable materials, is accomplished manually by tossing the grain into air and letting the wind do the separation and cleaning, removal of lightest impurities, leaves and large amount of debris with certain amount of grains or grains. For further cleaning is usually done using sieves to remove the heavy particles and dirt larger than that of tef grain or grain. The traditional methods of cleaning are labor-intensive and time consuming. In certain circumstances, the velocity of the wind may be too low so that heavier impurities (gravel, ear, chaff, etc.) remain mixed with the grains.

Against all the odds, the Ethiopian farmers prefer to grow tef because it tolerance to low moisture stress, waterlogged and anoxic conditions being better than maize, wheat, and sorghum. Cattle prefer to feed on tef straw rather than any other cereal straws. Moreover, tef as grain has highest market prices than the other cereals; this includes both grain and straw, and the grain is not attacked by weevils during storage (Seyfu, 1993). Doubling both the grain and the straw yield of tef would significantly contribute to Ethiopia's economy and food security. Techniques being explored to achieve these include not only increasing area under tef cultivation but also using appropriate mechanization technologies. To alleviate problems associated with chaff, straw, dirt and dust separation and cleaning tef grain it was felt appropriate to develop a machine that can separate undesired materials and clean tef grain after threshing using either manual, animal or mechanical means. Although this research endeavor is mainly concerned with tef grains, the techniques and the equipment will applicable to all types of grains by only replacing the sieve.

The general objective of the study was to design and fabricate a prototype machine for separation tef chaff from grain and to do cleaning.

2. Material and methodology

2.1 Experimental Site Description

The experiment was conducted at Asella Agricultural Mechanization Research Center (AAMRC), Oromia Agricultural Research Institute (OARI). Asella is located at 6° 59' to 8° 49' N latitudes and 38° 41' to 40° 44' E longitudes, having an elevation of 2430 meters above sea level (masl).

2.2 Design Considerations

The tef separating and cleaning machine was designed and developed based on the following considerations.

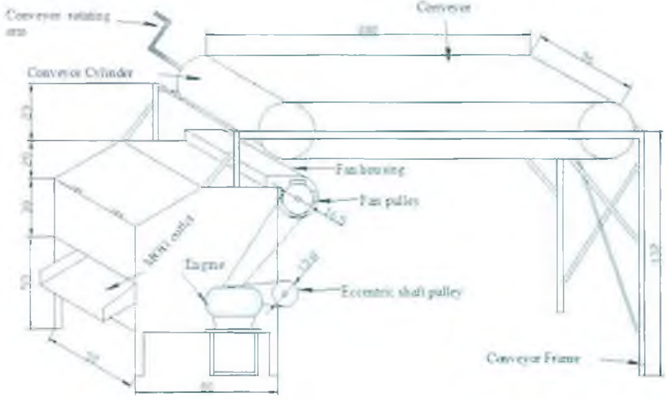
I. The availability of materials locally to reduce cost of production and maintenance.

II. Due to small size of tef grains and their low terminal velocity, separation of grains from chaff and straw and complete cleaning was thought to be practically impossible. Hence, it was decided to use combination of fan (pneumatic) and mechanical (sieves).

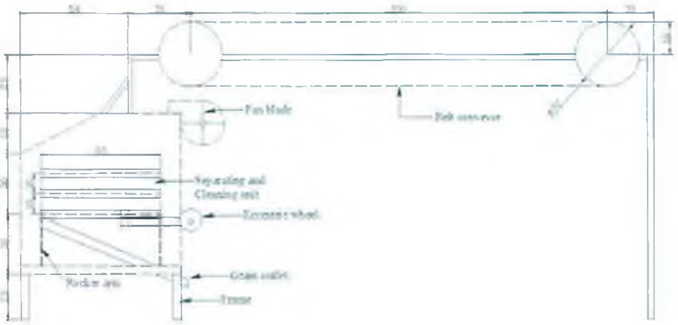
III. To avoid clogging of the machine during the experiment, manual operated constant and uniform feeding mechanism, for each feeding rate, was designed and used.

2.3 Description of the Machine

The major components of the tef and chaff separating and cleaning machine are the frame, belt conveyor, pulleys and V-belts, separating and cleaning unit, diesel engine, eccentric wheel, fan and fan housing. Figure 1 gives details of machine designed, constructed and used in the experiment.



(A)



(B)

Figure 1. Details of the Experimental Tef Chaff Separating and Grain Cleaning Machine (A = isometric view, B = section view; all dimensions are in cm)

A manual driven belt conveyor was used to control feed rate of chaff and tef grain into the separating and cleaning unit. The belt conveyor was horizontal and had been provided with rectangular guards (262 x 50 cm) on its both sides for easy flow. The fan house was made from 1.50 mm thick sheet metal and meant to accommodate the fan blade and shaft. The separating and cleaning unit had three sieves, sieve frame, rocker arm and chaff and grain outlet pan. A 5 hp

HONDA diesel engine was used as source of power. Drive and driven assembly consisted of a shafts, belts and pulleys. The frame of the experimental machine was made from 40 x 40 mm angle iron.

2.4 Working Principles of the Machine

A mixture tef grains and chaff was uniformly spread on the belt conveyer along its entire length and fed into the separating and clean unit at predetermined rate of feeding. As the materials were on their way into the separating unit, they were subjected to air current generated by the fan. As a result, light chaffs with low terminal velocities were blown away before they reached separating and cleaning units. Heavier materials, with higher terminal velocity, managed to land on the top sieve that permitted passage of tef grains and retained chaffs with sizes greater than the grains. The final separation and cleaning was effected by sieves oscillated. The top sieve was designed and selected to retain very coarse materials and to convey them to the MOG outlet. The middle sieve was selected in such way that it could scalp all materials larger and heavier than grains passed through the top sieve. The purpose of the bottom sieve was to carry out further cleaning of tef grain from trash, sand, dust, dirt and broken grains.

2.5 Design Analysis

2.5.1 Determination of sieve slope

The sieve slopes used in the experiment were determined in the basis of the condition that the angle α be $\leq$ angle φ (Fouda, 2009). Where α is the inclination of the sieve with the horizontal while φ is the friction angle between the mixture grain and chaff and the sieve surface. According to Zewdu and Solomon (2006), the mean value φ (angle of internal resistance) of tef grain on mild steel is 20°. Hence, 0°, 5° and 10° sieve slopes (Figure 3) were selected.

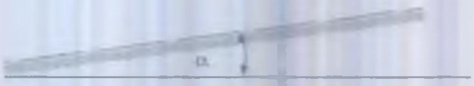


Figure 2. Inclination of sieve with the horizontal.

2.5.2 Determination of sieve oscillation

The data on optimum sieve oscillation frequency of sieves to make separation between tef grain and chaff and to effect cleaning was not available. However Khawaja (2009) recommended sieve oscillation frequencies between 6 and 20 Hz to be used in cleaning cereal grains. Accordingly, sieve oscillation frequencies of 5, 10, 15 and 20 Hz were used.

2.5.3 Construction of belt conveyor

The conveyor, meant to feed the prototype machine at uniform and continuous feeding rate and avoid over loading of the sieves and/or intermittent flow of materials into the separation and cleaning unit, was designed and made from cloth (*abujedi*). The conveyor was made to run over two cylinders made out of sheet metal, placed at center distance of 2 m and having diameter and width of 32 cm and 36 cm, respectively. According to Conveyor Belting Australia (CBA) (2009), the capacity of belt C (Eq. 1) was determined to be as 32.34 kg/s. The width of the conveyor cylinders and the cloth used (36 cm) were 4 cm narrower than the sieve width in order to make sure that all conveyed materials land on the sieve without spilling over and out of sieve area.

$$C = AV\rho \qquad (1)$$

Where: A = load cross section area perpendicular to the belt computed as Width x length = 0.36 m x 2 m = 0.72 m^2. P = density of material (1361 kg/m^3) and V = velocity of conveyor (0.033 m/s).

2.5.4 Design and fabrication of the prototype separating and cleaning unit

The major components of separating and cleaning unit included sieves, sieve frame and rocker arm. The sieving unit consisted of three sieves for separating and cleaning simultaneously. The three sieves had length of 60 cm and width of 40 cm.

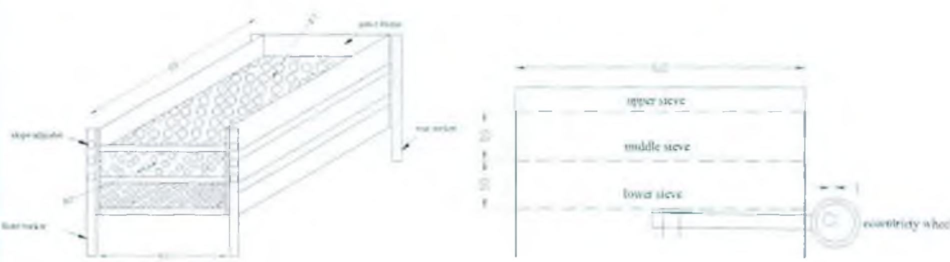


Figure 3. Schematic presentation of sieves arrangement and oscillatory motion generation system (all dimensions are in cm)

The top sieve had round holes, 3 mm in diameter, the middle sieve had round holes 2 mm in diameter while the bottom sieve had 1 mm slotted holes. The vertical intervals between sieves were 10 cm (Fig. 3). The eccentricity was responsible to cause oscillations on the sieves.

2.5.5 Design and construction of fan and housing

A centrifugal fan was constructed from a sheet metal of 1.50 mm thickness. It was mounted on a shaft with a diameter of 25 mm and supported on roller bearings below the feed conveyor. The fan assembly had four radial blades. The fan blades were 38 cm in length and 8 cm in width. The fan was operated at three different angular velocities where the maximum being 1000 rpm. The fan speeds were adjusted and set at desired level by adjusting engine speed rather than employing pulleys or gears. A 5 hp HONDA diesel engine with V- belt and pulley arrangements was used as power source. The fan housing was provided with eight rectangular inlet holes, four on each side. The sizes of the holes were four 6 x 5 cm^2 and four 8 x 5 cm^2 in order to control the air flow rate in into the fan housing. One rectangular hole with size of 6x40 cm^2, as an outlet of the air from the fan house, was provided. This hole directed air at recommended velocity towards materials falling from the conveyor into the separating and cleaning unit (Fig.4).

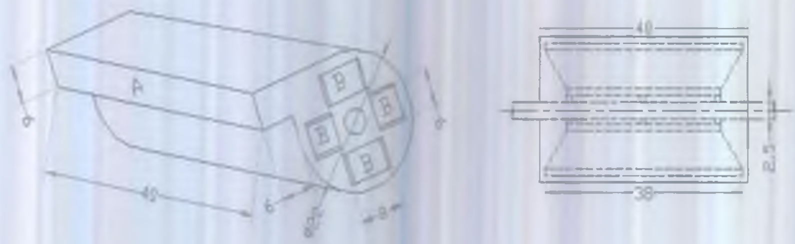


Figure 5. Fan housing and fan blade; A is outlet opening and B is inlet opening (all dimensions are in cm).

The fan discharged air blast under the feeding conveyor with constant velocity of 3.2 m/s (the minimum terminal velocity of tef) through the exit. This constant air speed was adjusted using a digital anemometer and either by decreasing or increasing the air inlet openings on the sides of the fan housing. The inlet openings could be closed and opened using the shutters provided to maintain the desired air velocity. The volume flow rate of air (V) at fan exit was calculated to be 0.077 m^3/s (velocity of exit (3.2 m/s) x cross sectional area at exit (0.024 m^2) = 0.077 m^3/s). Using the equation given below, quantity of material removed (G_m) by the air stream per unit time (kg/s) was determined to be 0.023kg/s.

$$V = \frac{G_a}{\rho} = \frac{G_m}{\mu\rho} \tag{2}$$

Where: V = volume air flow m^3/s, ρ = density of air (1.2 kg/m^3), G_a= the mass flow rate of air at exit which equals to the flow rate fan exit multiplied by the density of air (Vx ρ = 0.077 x 1.2 = 0.092 kg/s), and μ = a coefficient 0.25.

The dynamic head (h_d) of the fan was calculated using Eq. (3) and assuming C_{mean} = terminal velocity of tef grain = 3.20 m/s and density of air 1.2 kg/m^3 was found to be 6.14 Pa

$$h_d = \rho \frac{C^2_{mean}}{2}$$

(3)The inlet area (A_{eq}), coefficient (k), static head (h_s) and overall pressure head (H) were computed for different fan rpm and given in Table 2.

Table 1. Adjusted inlet area at different rpm to keep constant 3.2m/s at fan exit.

Fan speed (rpm)	Inlet Area A_{eq} (cm^2)	$k = \frac{A_{eq}}{A}$	$h_s = \frac{(1-k^2)h_d}{k^2}$ (Pa)	c_{mean} (m/s) (anemometer reading at fan exit)	Over all pressure head ($H=h_s+h_d$)
500	210	0.88	1.79	3.2	7.93
740	200	0.83	2.77	3.2	8.91
1000	160	0.67	7.54	3.2	13.68

2.5.6 Selection of pulleys and belts

The machine required three pulleys; one pulley mounted on the crank shaft of the diesel engine as main drive, and on fan shaft and the eccentric wheel shaft one each. Two belts were used to transmit power from engine to the fan shaft and eccentric wheel shaft. The driving pulley was mounted on the crank shaft of the engine and the driven pulleys were mounted on fan shaft and eccentricity wheel shafts. The diameter of the pulley used on the crank shaft of the engine was 120 mm. The power, from the engine shaft to the fan shaft and eccentric wheel shaft, running at different angular speeds, was transmitted through V-belts. Since the selected engine was 5 hp, slightly less than 4kw, type A V-belt was selected and used.

According to Sharma and Aggarwal (2006) the diameter of driven pulleys, center distance, belt length and belt speeds were calculated as follows:

$$D_1 N_1 = D_2 N_2 \qquad (4)$$

$$C = \frac{D_1 + D_2}{2} + D_1 \qquad (5)$$

$$L = 2C + 1.57(D_1 + D_2) + \frac{(D_2 - D_1)^2}{4C} \qquad (6)$$

$$V = \frac{N_2 \pi D_2}{60000} \qquad (7)$$

Where: D_1 and D_2 = diameters of driving and driven pulleys (mm),N_1 and N_2 = rpm of driving and driven pulleys, C = center distance between two adjacent pulleys (mm),L = length of belt (mm) V = speed of belts (m/s)

The diameter of driven pulleys was arrived based on the following values;

- The speed of diesel engine shaft is 1375 rpm (measured)
- Diameter of the engine shaft pulley is 120 mm (measured)
- Maximum fan speed is 1000 rpm (Adane, 2004)
- Maximum speed of eccentric wheel is 1200 rpm, since the maximum oscillation is 20 Hz.

Therefore, the diameters of pulleys used on the fan shaft and eccentric drive wheel shaft were calculated using Eq. (4) and were found to be 165 mm and 138 mm respectively. The centre distances between the diving pulley and driven pulleys were determined using Eq. (5), and found to be 263 mm and 249 mm the engine and fan and engine and eccentric wheel shafts, respectively. However, to maintain stability and to minimize vibration 500 mm center distance between fan shaft and engine crank shaft pulleys was used since it lies between $\frac{(D_1 + D_2)}{2} + D_1 \leq C \leq 2(D_1 + D_2)$ as recommended by Maciejczyk and Zbigniew (2000). The lengths of belts were calculated using Eq. (6) and found to be 1448 mm and 903 mm to connect pulleys on engine crank shaft and fan shaft and engine crank shaft and eccentric drive wheel shaft, respectively. Based on the calculations made A-57 and A-36 belts were selected to connect pulleys fan shaft and engine crank shaft and eccentric drive wheel shaft and engine crank shaft, respectively. The linear speed of the belt connecting the fan pulley to the engine pulley was 8.64 m/s with Eq. (7).

2.5.7 Determination of shaft diameter

The diameters of the fan shaft and eccentric drive wheel shaft were determined using maximum shear stress theory. Figure 6 shows forces acting on fan shaft.

Figure 6. Forces acting on fan shaft and their locations.

Where:

R_{AH} = horizontal bearing reaction force at A

R_{AV} = vertical bearing reaction force at A

W_{FB} = weight of fan blade

R_{CH} = horizontal bearing reaction force at C

R_{CV} = vertical bearing reaction force at C

T_B = total belt tension (T_i+T_j) at D
Wp = weight of fan pulley

Step1: Determining the direction belt pulls on the fan shaft (Fig. 7)

$$\sin\alpha = \frac{opposite}{hyphotness} = \frac{26}{50} = 0.433\text{, then } \alpha = 31.33^{o}$$

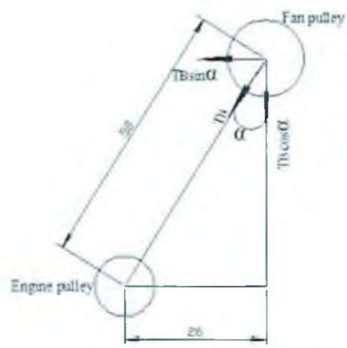


Figure 7. Direction belt pulls on the fan shaft.

Step 2: Determination of belt tensions (T_i and T_j) and torsional moment (M_t)

Using Eq. (8):

$$T_{\max} = \sigma a = 170.1 \text{ N}$$

Where; σ and a = 2.1 N/mm^2 and 81mm^2 respectively from (standard table)

Using Eq. (9):

$T_c = mV^2$ = 8.06 N, for both belts since they have almost similar velocity.

Where; m = 0.108 kg/m (standard table), V = 8.64 m/s

Then using Eq. (9) the tight side tension (T_i) of both belts was calculated to be;

$T_i = T_{max} - T_c$ = 162.04 N

Slack side tension T_j was determined as 19.12 N and 18.17 N for fan/engine and eccentric wheel/engine belts; using Eq. (10):

$$\frac{T_i - T_c}{T_j - T_c} = e^{\mu\theta \operatorname{cosec}\frac{\alpha}{2}}$$

Where: μ = coefficient of friction between belt and pulley = 0.3(tandard Table), α = groove angle = 40° (standard Table), θ = angle of wrap = 3.05 and 3.07 rad for fan-engine and eccentric wheel-engine belts respectively and were determined using Eq. (18) (Khurmi and Gupta, 2005)

$$\theta = 180 - 2\left[\sin^{-1}\left(\frac{D_2 - D_1}{2C}\right)\right] \qquad (11)$$

Where: D_1 = 120 mm for engine shaft pulley,D_2 =165 mm for fan shaft pulley and 138 mm for eccentric drive wheel pulley, C = 500 mm for between shaft and engine pulley and 249 mm for between eccentric drive wheel shaft and engine pulley.

Torsional moments (M_t) were calculated using Eq. (12).

$$M_t = (T_i - T_j)\frac{D_2}{2}$$

(12)

And found to be 11791 N-mm and 9927 N-mm for fan and eccentric wheel shaft respectively.

Where: T_i = 162.04 N for both fan/engine and eccentric drive wheel/engine belts,T_j=19.12 N for fan and 18.17 N for eccentric and D_2= 165 mm for fan 138 mm for eccentric.

Step 3: Analysis horizontal and vertical forces on fan shafts

To calculate bending moments on shafts it was necessary to know the horizontal and vertical forces acting on shafts.

- **Forces acting on fan driving shaft - vertical (YZ) plane**

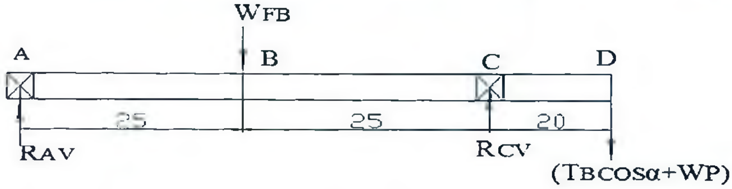


Figure 8. Free body diagram of the fan shaft on vertical (YZ) plane.

$T_B = T_i + T_j$ = 162.04 +19.12=181.16 N

W_{FB} =11 N, W_p =15 N, α = 31.33°, ($T_B \cos\alpha + W_P$) = 170 N

In order to calculate reaction forces R_{AV} and R_{CV}, it was considered that

$\sum BM_A = 0$

50 *R_{CV} = 25*W_{FB} + 70 ($T_B \cos\alpha + W_P$)

R_{CV} = 243.50 N

$\sum F_y = 0$

R_{AV} + W_{FB} + R_{CV} + ($T_B \cos\alpha + W_P$) = 0

R_{AV} = 62.50 N downward

- **Forces acting on fan driving shaft on horizontal (XZ) plane**

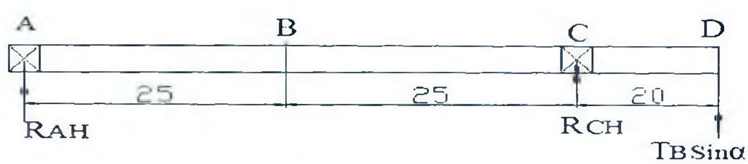


Figure 9. Free body diagram of fan shaft on horizontal (XZ) plane.

$T_B \sin\alpha$ =181.16 x sin31.33 = 94.2 N

$\sum BM_A = 0$

50 *R_{CH} = 70 $T_B$sin a

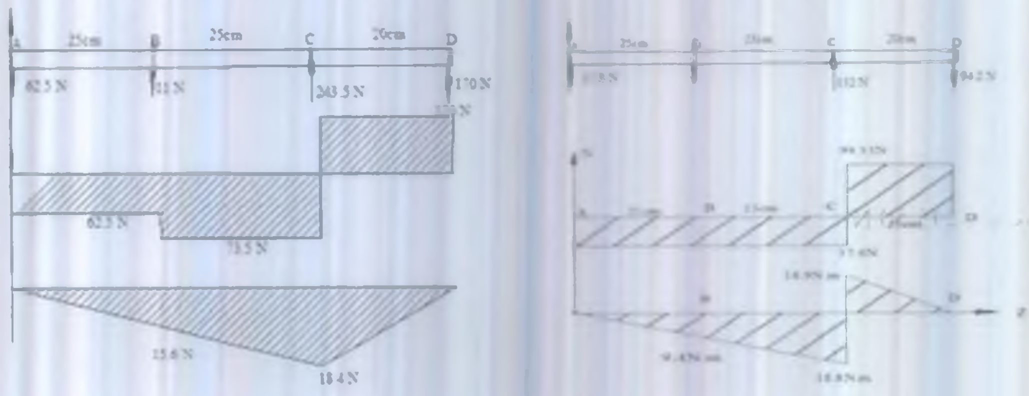
$R_{CH} = 132$ N

$\sum F_h = 0$

$R_{AH} + R_{CH} + T_B \sin a = 0$

$R_{AH} = 37.8$ N downward.

Based on the magnitude and location of all forces acting on the fan shaft shear force and bending moment on the horizontal and vertical plans containing the fan shaft were computed and plotted as follows:



a) Vertical (YZ) plane   b) Horizontal (XZ) plane

Figure 10. Shear force and bending moment diagrams for fan shaft

Step4: Determination of the maximum bending moment for fan shaft

The maximum bending moment was found to be at point C of the fan shaft.

$$M_{\max} = \left(M^2{}_V + M^2{}_H\right)^{1/2} = (18.4^2+18.9^2)^{1/2} = 26.38 \text{ N-m}$$

Finally the diameter of fan shaft calculated as equation below

$$d^3 = \frac{16}{\pi\tau_{\max}}\sqrt{(k_b M_b)^2 + (k_t M_t)^2}$$

M_t=11.79 N-m, M_b = 26.38 N-m, k_t= k_b = 1.5 and τ_{max} = 55 Mpa then d = 15.89 mm. Assuming a safety factor of 3, d = 15.87 x $\sqrt[3]{3}$ = 22.89 mm

$\therefore d = 25$ mm

Following the same procedure torsinal and bending moment and diameter of eccentric wheel driving shaft can be:

$M_t = 9.93$ N-m, $M_b = 27.20$ Nm, $k_t = k_b = 1.5$, $\tau_{max} = 55$MPa then d = 15.9 mm. Assuming a safety factor of 3, $d = 15.9 \times \sqrt[3]{3} = 22.93$ mm

$\therefore d = 25$ mm

2.5.8 Determination of power

The power required to operating the separating and cleaning machine was considered to be the sum of powers required to drive the fan assembly, the eccentric drive assemble, the sieving system including the grain and chaff on the sieves, power required to oscillate the sieves and the loads on them and power required to overcome frictional resistance. The total power (P_t) required for the cleaning and separating processes was determined by using the Equation given by Nduka *et al*, (2012).

$P_t = P + 10\%P$ ***(10% is possible power loss due to friction drive)*** (13)

Where: P_t = total power required to drive the machine, P = the sum of $(T_i - T_j)\,V$ for fan and sieve oscillation, T_i = tight side tension of belt, 162.04 N for both fan and eccentric drive wheel belts and T_j = slack side tension of belts = 19.12 and 18.17 N for fan and eccentric drive wheel belts, respectively, and V = speed of belts = 8.64 m/s for both fan and eccentric drive wheel belt.

On the basis of the above, power required to drive the fan assembly, P_f = (162.04 N - 19.12 N) 8.64 m/s = 1234.83 W

Accordingly, the power required to drive eccentric drive wheel assembly, Pe = (162.04 N – 18.17 N) 8.64 m/s =1243.04 W

The total power to operate fan and eccentric drive assembly $P = P_f + P_w = 1234.43\text{W} + 1243.04\text{W} = 2477.87$ W.

Overall total power P_t = P + 10% of P = 2725.66W = 3.63 hp.

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Performance Evaluation of Asella Multi crop thresher for Soybean

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ABSTRACT

The objective of this paper is to study the feasibility of using Asella multi cropthreshing machine for soybean and selecting the optimum conditions for operation. Threshing drum speeds tested were 500,600 and 700 rpm and feed rates of 5, 10 and 15kg/min and soya bean seed was used at moisture contents of 9%. Performance parameters were threshingefficiency, cleaning efficiency, threshing capacity and total loss.The test results indicated a maximum of 99.87% threshing efficiency, 81% cleaningefficiency, 156.84kg/hrthreshing capacityand a minimum of 4.07% total seed loss. The performance was found to be influencedby all the study variables.

Key Words: thresher, performance, soya bean, threshing efficiency, cleaning efficiency, threshing capacity and total grain loss.

1 . Introduction

Soybean is grown in varied agro-climatic conditions. It has emerged as one of the important commercial crop in many countries (Ogoke et al. 2004). It is an annual plant that may vary in growth, habit and height. It may grow up to 2 meters in height. The pods, stems, and leaves are covered with fine brown or gray color. The fruit is a hairy pod that grows in clusters of 3–5, with each pod 3–8 cm long and usually containing 2–4 (rarely more) seeds 5–11 mm in diameter (Nieuwenhuis, 2002).

Soybean is a favorite commodity for export and local consumption. The successful production and good marketability ofthis crop depend on both quantity and quality of the crop. Threshing is one of the important practices which can affect the quantitative and qualitative losses of soybean.

To increaseseed production with good quality feasibility of threshing machine is a must.

Up till now, the manual threshing of soybean crop is still the common practice followed by the majority of the farmers in Ethiopia.The traditional methods of threshing soya bean pods involve manual rupturing of the pods and the separation of the seed from the chaff. The process is tedious and time consuming; it also results in losses as well as low quality product.To facilitate speedy threshing of soybean is necessary, in order to reduce post-harvest deterioration, mechanical threshers are recommended, because manual threshing methods cannot support commercialized threshing.

Threshing efficiency increased with the increase of cylinder speed but decreased with the increase of feeding rate and concave clearance. An optimum speed is desirable to get an optimum performance of a thresher as excessive speed can cause the grain to crack, and too low speed can give unthreshed heads(Kaul and Egbo, 1985).

Hence, the main objectives of this study was to evaluate the existing Asella model 3 multi crop thresher for soybean threshing and to select the optimum conditions of threshing drum speed and feed rate in order to reduce grain damage and producing better quality of grain in post-harvest operation

3 . Materials & Method

Field experiments were carried out for threshing soybean crop in Asella agricultural mechanization research center during season 2013. All experiments were conducted using Asella multi crop threshing machine.The specifications of these machines are shown as follows (fig.1):-

Threshing unit:

Type of drum: loop type
Number of pegs- 32
Drum diameter: 308 mm.
Drum length: 940 mm.
Drum –concave clearance 26mm

Figure 1. Multi crop thresher

The cleaning unit comprises of a screening sieve and a blower.The blower is made of mild steel sheet and enclosed in spiral shaped housing. An inlet for the sprout is provided at the top of the housing along its length and attached directly to the sprout.The blower is made of 3 vanes of 140mm by 60mm and is attached to a 10mm diameter blower shaft mounted on two bearings on each side to allow free rotation. It is connected to the threshing pulley by a belt to its own pulley.

3.1 Performance Evaluation

The threshing machine performance was studied under the following condition:-

-Three different of feeding rates of 5, 10 and 15 kg/min.

- Three different of drum speeds of 500, 600 and 700rpm.

-The seed moisture content of 9 % (d.b) at the optimum maturity of dry soybean plant

-Powersource of an ATIMA motor engine (10hp) maximum speed of 3000rpm.

-The threshing efficiency, cleaning efficiency, percentage total loss and germination were evaluated at each combination of variables.

3.1.1 Determination of total threshing losses

The total threshing losses percentage (TGL) including both unthreshed grain losses (UGL) and damage grain losses (DGL) was calculated using the following equation, (Mishram and Desta 1990):-

$$TGL,\% = \frac{D_G + Un_G}{T_G} \times 100 \qquad 1$$

Where: DG = Weight of damaged grains collected at all outlets per unit time, kg.

Un_G= Weight of unthreshed grains per unit time, kg.

TG = Weight of total grains input per unit time, kg (Total grain input (TG) = feed rate x grain-straw ratio).

Unthreshed grain losses (UGL) percentage was calculated as follows:-

$$UGL,\% = \frac{Un_G}{T_G} \times 100 \qquad 2$$

Damaged grain losses (DGL) percentage was calculated as follows:-

$$DGL,\% = \frac{D_G}{T_G} \times 100 \qquad 3$$

3.1.2 Determination of efficiencies

i. $Threshing\ efficiency, \% = \frac{T_G - Un_G}{T_G} \times 100$ 4

ii. $Cleaning\ efficiency, \% = \frac{W}{W_O} \times 100$ 5

Where:-W - Weight of grains from the main output opening after cleaning, kg.

Wo- Weight of grains and small chaff from the main output opening, kg.

3.1.3 Determination of threshing capacities

Threshing capacity (Kg /hr),

$$T_C = \frac{W_{TG}}{T} \quad 6$$

Where: -T_C – threshing capacity

W_{TG} -Weight of threshed grains

T – Threshing time

4. RESULTS AND DISCUSSION

The performance of the thresher was evaluated at the various drum speed and feed rate at moisture content of 9% in terms of threshing efficiency, cleaning efficiency, Threshing capacity and percentage total loss. Table 1 gives the results of the performance tests.

4.1 Threshing Efficiency

4.1.1 Effect of threshing drum speed

The results are plotted in Fig. 2. It could be noticed that the lowest values of threshing efficiency was obtained at 500 rpm, however the highest values of threshing efficiency was obtained at 700 rpm at different feeding rates. As the drum speed increased the un-threshed seeds decreased. These results may be due to increasing the impact action of the threshing drum on the threshed material (soybean heads) which is directly proportional with drum speed. Increasing the impact action due to increasing drum speed increased the kinetic energy of the threshing material which differs widely from seeds to soybean heads trashes; consequently the seeds will be quicker and run away from the trashes. Therefore the separating efficiency will be increased and increased the threshing efficiency.

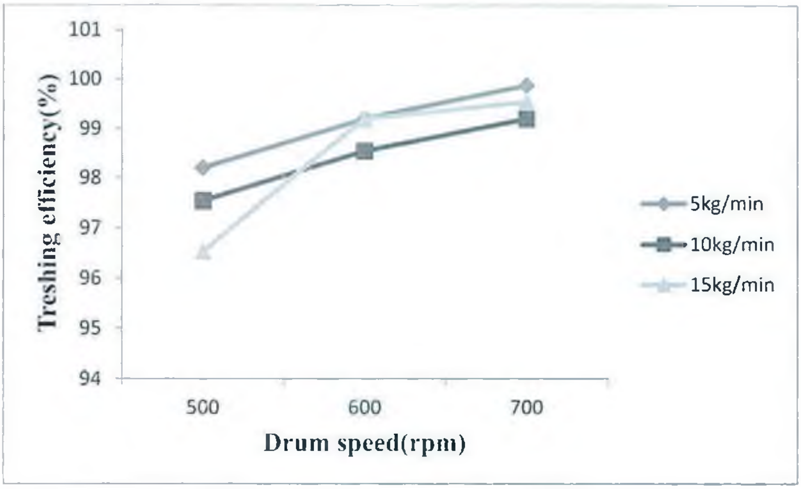


Fig. 2. Effect of drum speed on threshing efficiency

Increasing drum speed from 500 to 700rpm under constant material feed rate of 5, 10 and 15 kg/min increase the percentage of threshing efficiency by 1.67, 1.66, and 3 %, respectively using the threshing machine under the same conditions.The increase in the percentage of threshing efficiencies by increasing drum speed are attributed to the high stripping and impactingforces applied to the soybean materials, that tends to improve the threshing operation and increase threshing efficiencies.

4.1.2 Effect of material feed rate on threshing efficiencies

Concerning the effect of material feed rate on the percentage of threshing efficiency, results obtained shows that increasing feed rate increased the percentage of threshing efficiency when feeding rate increase from 5 to 10kg/min and decreased the percentage of threshing efficiency from 10 to 15kg/min under all experimental conditions. The increase in the percentage of un-threshed grains and the decrease in percentage of threshing efficiency by increasing material feed rate are attributed to the excessive materials in the threshing chamber. Consequently, soybean plants leave the device without complete threshing that tends to increase un-threshed grains and decrease damaged grains.

4.2 Cleaning Efficiency

4.2.1. Effect of threshing drum speed on cleaning efficiency

The results in Fig. 3, illustrated that the lowest values of cleaning efficiency was obtained at drum speed of 500 rpm of threshing machines, however the highest values of cleaning efficiency was obtained at drum speed of 700 rpm at different feeding rate. The high drum speed increased the velocity of cleaning air results in higher capability of air to carry the foreign material and residual of soybean heads from seeds consequently increased cleaning efficiency. On the other words, increased the drum speeds from 500 to 600 rpm increased the cleaning efficiency by 1.33% at feeding ratesof 5kg/min. However by increasing drum speed from 600 to 700 rpm increased the cleaning efficiency by 0.67% at feeding rate of about 5kg/min.

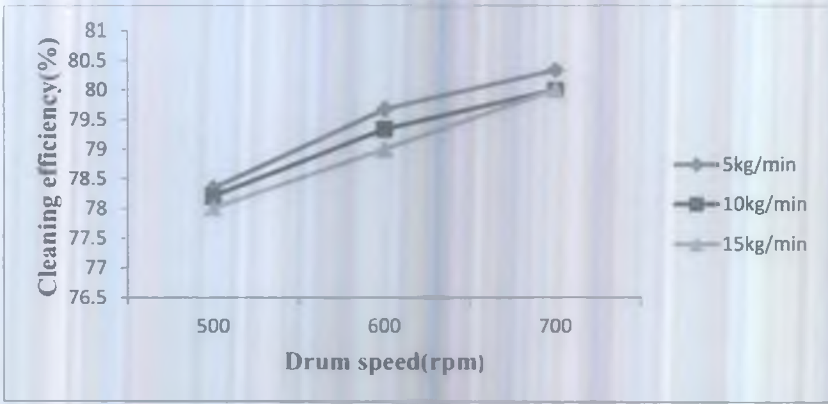


Fig. 3. Effect of drum speed on cleaning efficiency

4.2.2. Effect of feeding rate on cleaning efficiency

The results plotted in Fig. 4, indicated that the increasing feeding rate from 5 to 10 kg/min. the cleaning efficiency was decreased by 0.14, 0.33 and 0.34% at drum speed of 500 , 600and 700rpm respectively. This is true, because the machine have oscillating sieves which greatly increase cleaning efficiency. The maximum cleaning efficiency of 80.34% was obtained when using the machine at feeding rate of 5 kg/min and 700rpm drum speed under 9% of moisture content.

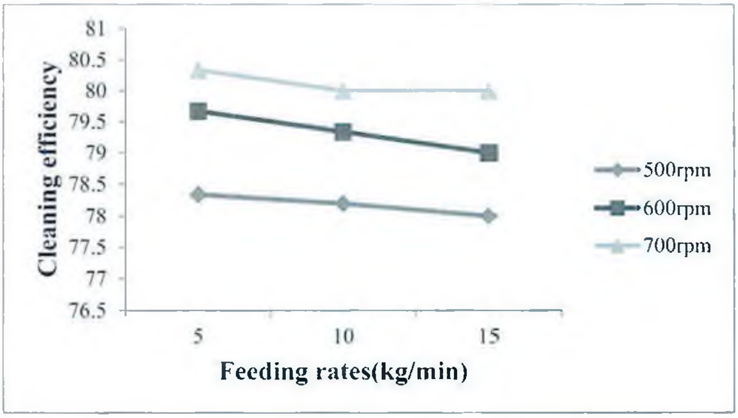
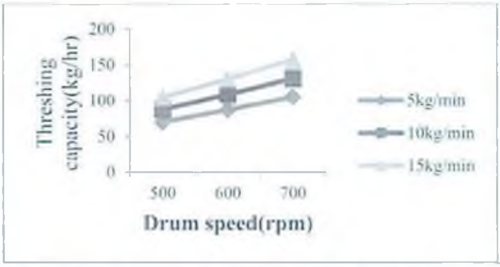


Fig. 4. Effect of feeding rates on cleaning efficiency

4.3 Threshing capacity

Figure 5showsthat, the effect of drum speed on capacity at different feed rates. The results indicated that the capacity increased with an increased in drum speed for all feed rates. And also the capacity increase with an increase of feed rates. The maximum capacity was obtained at the feed rates of 15 kg/min and at drum speed of 700 rpm.



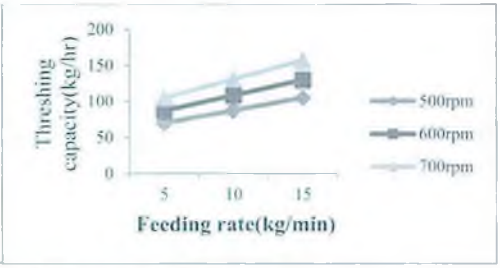


Fig. 5.Effect of drum speed and material feed rate on threshing capacity (kg/hr)

4.4 Total Seed Loss

4.4.1 Effect of drum speed

The total grain losses are the sum of un-threshed grains, damaged grains and seeds thrown out directly by oscillating sieve motion and air velocity during the threshing operation. The total losses were influenced significantly by the cylinder speed. The results in Fig.6, illustrated that the total seed increased with increasing drum speed at different feeding rates. Also, it could be noticed that the lowest values of the total seed losses was obtained at 500 rpm drum speed of threshing machines. However, the highest values of the total seed losses were obtained at 700 rpm drum speed of threshing machines at different feeding rates. The results also cleared that at feeding rate of 5 kg/min., increasing threshing drum speed from 500 to 700 rpm increased the total seed losses from 4.07 to 4.2 %.

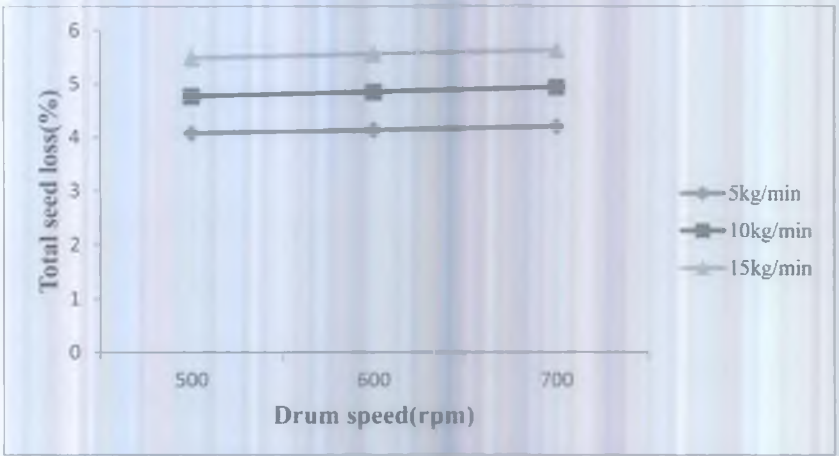


Fig.6. Effect of drum speed on total seed loss

4.4.2 Effect of feeding rate

Fig. 7. illustrated that the maximum values of total seed losses percentage of about 5.48, 5.56 and 5.63 % was obtained at feeding rate of 15 kg/min. On the other hands, increasing feeding rate from 5 to 10 kg/min at drum speed of 500rpm increased the total seed losses from 4.07 to 4.76% under 9% soybean heads moisture content.

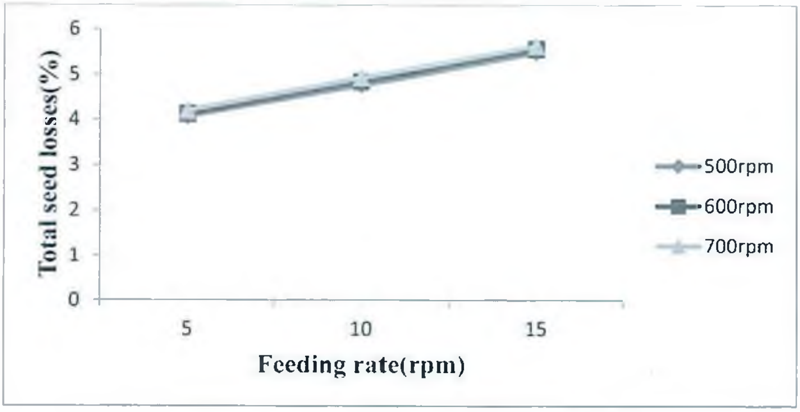


Fig.7. Effect of feeding rates on total seed loss

4.5 Seed breakage

4.5.1 Effect of threshing drum speed

The results in Fig.8.indicated that by increasing drum speed from 500 rpm to 600 rpm gave the lowest increment rate than obtained when increased the drum speed from 600 to 700 rpm. Also, it could be noticed that the lowest values of seed damage was obtained at 500rpm, however the highest values of seed damage was obtained at 700 rpm of drum speed of threshing machines at different feeding rate.

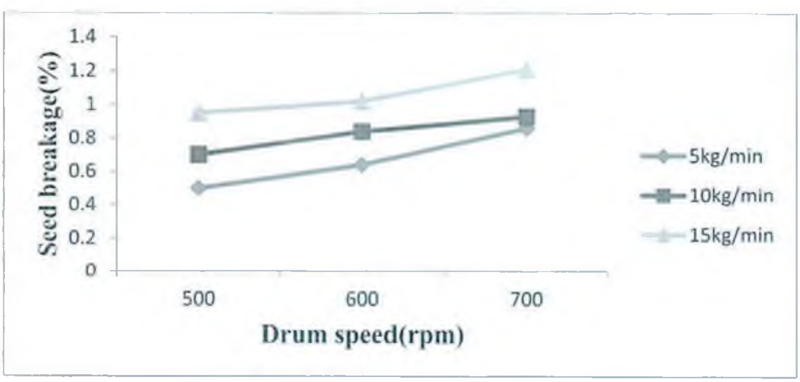


Fig.8. Effect of drum speed on seed breakage

4.5.2 Effect of feeding rate

Results in Fig. 9, illustrated that the higher values of seed damage resulted at 15kg/min of feeding rates of the machine. Increasing feeding rate from 5 to 10 kg/min. increased the seed damage by 0.2%while increasing feeding rates from 10 15kg/min increased the seed damage by 0.25 % at drum speed of 500rpm. The maximum seed damage percentage of 6.79, 5.32 and 1.56 % were occurred at feeding rate of 15 kg/min under different drum speed.

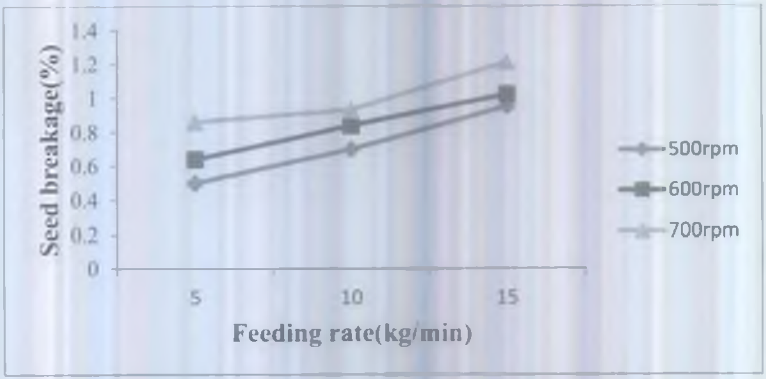


Fig.9. Effect of feeding rate on seed breakage

4.6 Summery

The following conclusions could be drawn from this study:-The performance of the modified soybean thresher was affected by the moisture content, the cylinder speed of the threshing drum as well as the feed rate. The optimum threshing efficiency of 99.87 % was obtained using the drum speed of 600 rpm, feeding rate of 5kg/min and 9 % of grain moisture content. Maximum threshing capacity of 156.84 kg/hr was obtained at drum speed of 700 rpm, 15kg/min feeding rate and 9 % of grain moisture content. Maximum Grain loss was 5.63% at drum speed of 700 rpm, 10/min feeding rate and 9 % of grain moisture content. Minimum grain loss 4.07 % at drum speed of 400 rpm 5 kg/min feeding rate and 9 % of grain moisture content. Maximum cleaning efficiency 81% at drum speed of 600 rpm, 10 kg/min feeding rate and 8 % of grain moisture content. Minimum cleaning efficiency 78.34 % at drum speed of 400 rpm, 5 kg/min feeding rate and 9 % of grain moisture content.

4.7 Recommendation

In order to minimize damaged grain, it is recommended that to investigate the effects of other machine-crop parameters such as concave clearance, hopper inlet opening and crop variety on the performance of the thresher.

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Adaptation and Promotion of Agro Saw Seed Cleaner

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ABSTRACT

The machine has over all dimensions of length, width and height 2400 mm, 1400 mm and 2800 mm respectively. The cleaning system consists of an air blast fan and a reciprocating shaker containing replaceable sieve for different crop seeds. The cleaning operation is powered by electric motor to drive cam shaft, fane and grain elevator. The developed cleaning machine had the ability to clean the premature grains, chaff and leaves, which are lighter than grains. The performance of the adapted prototype was evaluated in terms of percentage cleaning efficiency and cleaning capacity at various levels of feed rates. The test crops used was wheat. The best performance was obtained 99.07% and 19.2quintal/h of cleaning efficiency and capacity respectively for wheat grains.

1. INTRODUCTION

Agriculture is the largest single industry in the world, and seed production is an important segment of this industry. In traditional method, about 40% of the total labor required to produce crop is extended in threshing, cleaning and transportation activities (Johnson, 1992). The average post- harvest losses of food crops such as Teff, Wheat and Maize are annually 12.9%, 13.6% and 10.9% respectively (Derege A. et al 1989). Among this cleaning by rational method has many loss of grain.

Cleaning of grain or winnowing is one of the important postharvest processes done in preparing seeds/grains as food or any industrial raw material. It involves the removal of chaff and other debris from the grain. There are quite a number of factors that affect the performance in terms of cleanliness and grain loss during the operation. Such factors include amount of wind or air velocity, feed rate, shaker frequency, dimension of sieve opening, sieve tilt angle, crop variety and moisture content (Sharma, 1976).

In Ethiopia, grain cleaning i.e. removal of undesirable materials, is accomplished manually by tossing the grain into air and letting the wind do the separation and cleaning, removal of lightest impurities, leaves and large amount of debris with certain amount of grains. In certain circumstances, the velocity of the wind may be too low so that heavier impurities (gravel, ear, chaff, etc.) remain mixed with the grains. These contaminants must be removed, and the clean seed properly handled and stored to provide a high quality planting seed that will increase farm production and supply uniform raw material for industry.

In Ethiopia, seed/grain cleaning is part of women's contribution in processing of grains. A circular tray made from the grass so called 'sindedo' of stalk is used. The long hours associated with the traditional method results in fatigue, loss of concentration and consequently, reduction in separation quality. So often the natural wind condition may not be favorable for the operation and the result is increased time of operation and drudgery. Especially, seed producer cooperatives, governmental seed enterprises, state farms and agricultural research centers are facing problems of seed cleaning and most of them are processing their seed manually and the others import the machine from abroad by expensive hard currency. Even the machine has been installed by the experts coming from the manufacturing company.

Seed cleaning can be accomplished by using pneumatic separators, screen cleaners, or gravity tables. Many commercial cleaners incorporate more than one of these cleaning methods (Hauhouot*et al.*, 2000).In general seed processing has four stages namely, rough cleaning, pre-cleaning, industrial cleaning and fine cleaning. Rough-cleaning,' on the other hand, is a process .in which' material both larger and smaller than the crop seed is removed. Pre-cleaning or rough-cleaning is now regarded as a basic operation by many seeds men, because seed harvested with modern combines are often heavily contaminated with foreign material such as sticks , stems, leaves, trash and weed seeds . This material, which may be as much as 60 to 70 percent of the volume of the combine run seed lot, needs to be removed before seed can be safely stored, effectively cleaned.

This stage is, as the first entrance for further seed processing and grading, is very important and requires huge labor and consumes more time when it is operated manually. Especially, seed producer cooperatives, governmental seed enterprises, state farms and agricultural research centers are facing problems of seed cleaning and most of them are processing their seed manually and the others import the machine from abroad by expensive hard currency. Even the machine has been installed by the engineers coming from the manufacturing company.

Hence, this project was initiated to bridge the existing technology gaps in the area of seed cleaner, with prime objective of adopt and manufacture the AGROSAW (Model-PC-3) seed cleaner and conduct promotion with the following specific objectives:-

1. To adapt AGROSAW (Model-PC-3) pre-cleaner and make performance evaluation of the machine at station.
2. To promote the adopted technology for selected governmental and non-governmental organizations
3. To create awareness among seed enterprise, research centers, etc. to raise their perception on the technology.

2. Materials and Methods

2.1 Description of the Machine

The major components of the machine are the frame, grain elevator bucket, hoppers, dust collectors, pulleys and V-belts, separating and cleaning unit, electrical motors, eccentric wheel, fan and fan housing. Figure 1 gives details of machine developed, constructed and used in the experiment. (Photograph of the cleaning machine).

Figure 1. Grain cleaning machine

2.1.1 Bucket elevator

The bucket elevator consists of buckets attached to an endless belt which runs along a vertical path. The system would be enclosed in separate housings for the lift leg and return leg. The buckets load themselves as they pass through a seed hopper or '*boot" at the bottom of the run, and, depending upon elevator type, the load is discharged by centrifugal force or gravity as the buckets round the top section of the assembly. The centrifugal-discharge elevator is commonly operated at high speeds to obtain high capacities, and the resultant discharge velocity may cause excessive damage to injury sensitive seed. Slow-speed units are better for easily damaged or slow-flowing materials. Another limitation of bucket elevators is that they are difficult to clean. Small seeds tend to lodge in hard-to-clean areas throughout the elevator and remain there to contaminate subsequent lots.

2.1.2 Power source

The following electric motors were installed in the frame of the cleaning unit.

Cam Shaft Drive: - Single face electric motor,1.1 KW, 1.5 HP, 220 V, 1420 rpm, 50Hg, 10.4 A, Anti clockwise rotation.

Fan Shaft Drive: - Single face electric motor,2.2 K.W, 3HP, 220 V, 1450 Rpm, 50Hg, 13 A, Anti clockwise rotation.

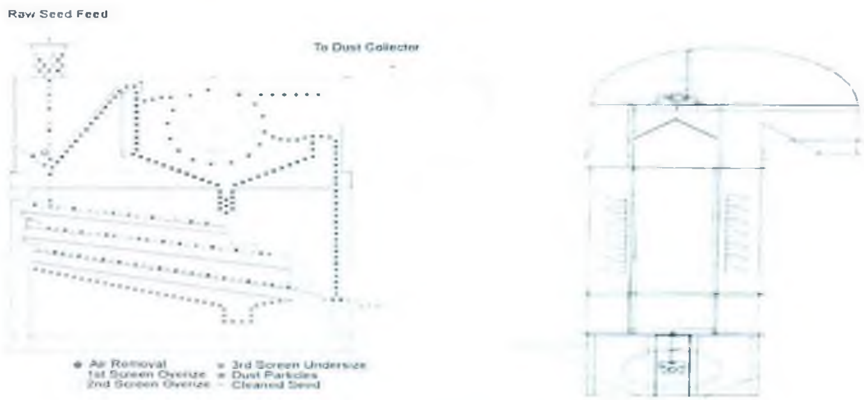
Grain Elevator Drive: - Single face electric motor.1.1 KW, 1.5 HP, 220 V, 1420 Rpm 50Hg, 6.6 A, Anti clockwise rotation.

2.1.3 Selecting Screens

The two basic screens for cleaning round-shaped seed are a round-hole top screen and a slotted bottom screen. The round-hole top screen should be selected so as to drop the round seed through the smallest hole possible, and retain anything larger. The seed drops through the top screen onto the slotted bottom screen, which takes advantage of seed shape and retains the round, good seed while dropping broken crop seed and many weed seed. The basic screens for cleaning elongated seed are a slotted top screen and a slotted bottom screen. In special separations it may be necessary to pass such seed through round hole top screens or over some screen other than a slotted bottom screen, but generally, slotted top and bottom screens are used.

2.1.4 Operation system of the Machine

The grains seed lifted by elevator bucket from the ground vertically upward and discharge it from top to the machine hopper where they are evenly distributed by a feed roller and transferred through a controlled gate on the top sieve. In the process the grains are subjected to primary aspiration by the use of air trunk which drains off chaff, straw, dust or decreased grains. Then the material is passed through sieve layer for separation according to their width and thickness. After the separation, the graded material is subjected to air lifter and aspiration chamber where remaining light particles are sucked off by a strong upward draught of air. The graded material and the impurities are automatically discharged in separate chutes.



Cross sectional view of the machine and grain elevator

2.2 Performance Evaluation

The test involves taking randomly selected three samples which were at the grain outlet and the non-grain (unwanted material) outlet. The weights of grain and other material in each sample were recorded. The procedure was repeated for each throughput. The amount of debris in clean grain outlet samples determined the cleanliness (cleaning efficiency) while the amount of grain found in non-grain outlet samples determines the grain loss. The expressions used for calculating the percent cleaning efficiency and percent grain loss were as follows:

$$\eta = \frac{G_o}{G_O + C_{cg}} x\ 100$$

Where:-

η = cleaning efficiency, %
G_o = weight of pure grain at the outlet, g.
C_{cg}= weight of contaminant in cleaned grain, g.

$$C_L = \frac{G_i}{G_w} x\ 100$$

Where:-

G_i= weight of grain at the chaff outlet, g.
G_w= weight of grain at input, g.
C_L = cleaning loss,

2.2.1 Mechanical Seed Damage

Seeds can suffer mechanical injury in any handling operation that provides a chance for abrasion or impact to occur. The injury may not be evident immediately, either visually or by germination tests, but may appear later in the form of reduced storage life and poor field germination.

2.2.2 Method of promotion

To conduct the promotion of seed cleaning technology to seed producer, research centers, farmers unions' and respective organizations were purposively selected based on their seed producing potential and accessibility. From these selected organizations two respective persons were invited to attend field days organized at Asella Agricultural Mechanization research center by panel discussions for one day.

3. Result and Discussions

The machine was fabricated at AAMRC on the basis of the manufacturer specifications. The physical attributes of grain was used for selection of sieve opening sizes required to effect the cleaning of the grain. Oscillations of the sieves were made by using a four-bar linkage mechanism where the legs of the sieve holding frame were pinned and oscillated about a vertical plane.

3.1 Capacity of the machine

Capacity of the cleaner varies depending upon the crop, contaminants and grade desired. Best results with the samples tested were obtained with incoming feed rates of 19.2 quintals / hour and 99.07 % cleaning efficiency in Wheat crop whereas the cleaning capacity of imported machine was 18.6quintal/hour this may happen duo to sieve difference.

3.2 Effect of Feed rate on Cleaning Efficiency and Cleaning loss

As we can see from test results, increasing the feed rate was caused decreased in cleaning efficiency. The declining cleaning efficiency and increasing cleaning loss with increasing rate of feeding was due to the formation of a thick layer of material on sieves that considerably hindered or limited passage of grains through the sieve perforations. The increasing in cleaning loss and decreasing of cleaning efficiency with increasing feed rate could be attributed the load intensity on the sieve that could result in matting of the sieve with material other than grain blocking sieve holes. Furthermore, whenever there are high feed rate, the current supplied will not be capable of suspending and blowing the materials aerodynamically as multiple particles act as obstruction to airflow.

3.3 Participants views towards promoted technology

Participants were asked how they perceive to the new technology as compared to imported technology of seed cleaningmachine in terms of its given features/attributes like, easiness of utilization, appearance of the machine, import substitution, affordability of technology. Likertscale method was used to measure respondent's opinion/views towards the attributes of the new technology with respect to imported seed cleaning machine. A Likert scale is an ordered scale from which respondents choose one option that best aligns with their view. It is often used to measure respondents' attitudes by asking the extent to which they agree or disagree with a particular question or statement. In this case an odd number of response categories having five responses (strongly disagree, disagree, neither disagree nor agree, agree, and strongly agree) were used. All participants strongly agree with the easinessto utilize, import substitution, affordability of technology with respect to imported seed cleaning machine. Out of total of participants about ninety nine percent of participants strongly agree with necessity of import substitution.

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Computational Fluid Dynamic Simulation and Experimental Testing of a Serpentine Flat Plate Solar Water Heater

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Abstract

The aim of the study is to improve thermal performance of passive serpentine flat plate solar collectors using striped technique. Striped mechanism was applied on absorber plate so as to diminish thermal fusion in the plate and investigation enhancing practice of energy conversion from the collector units to the working fluid. Study was conducted or carried out with numerical simulation and experimental testing to compare results for validation. Demand of domestic hot water has mostly been filling with conventional flat plate solar collectors. Conventional solar collectors are relevant for high flow rate that requires high operational costs. In the past, serpentine solar collector was ignored due to large pumping requirements at higher flow rates. However at low flow rate, serpentine collector is more economical and efficient. Therefore, striped absorber plate of the serpentine solar collector in varoius modes were designed by ANSYS 14.5 relase FLUENT and simulated using computational fluid dynamics. The effect of the configuration parameters of striped serpentine solar collector was investigated and good result was obtained.The anaysis was done by decoupling the last striped from whole system. So that the result of the second stripe became inlet boundary condition for the last of four segments. For the collector mass flow rate of 0.00285kg/s and solar radiation of 650w/m^2*, temperature of absorber plate(*T_p*) and water at collector exit (*T_0*) became 360k and 338k respectively. The same collector model was manufactured and experimental investigation was carried on with similar conditions as did for simulation. Therefore, absorber plate (*T_p*) and water at exit of the collector (*T_0*) during the experimental test attained maximum temperature of 353k and 336.9k respectively. There fore, numerically predicted temperature distribution on the striped absorber plates was agreed with experimental obtained data with little discrepancy. This inconsistency was due to variatiance of solar radiation and data measurement error. Collector heat removal factors obtained with both numerical study and experimental testing was similar in figures and remarkable with other research.*

Keywords: *Thermal breaking, serpentine flat plate solar collector, thermosyphone, CFD & Experiment*

1. Introduction

Solar collectors are special kind of devices that transform solar irradiance into internal energy of the transport medium, and hence increases their thermal effects. They absorb and capture the incoming solar radiation, converts it into heat and transfers the heat to a fluid flowing through the collector[1]. Demand of domestic hot water has been satisfying with various heating application.

However, solar water heating alone reduced domestic water heating costs by as much as 70% [2]. Flat-plate solar collector is the most common for residential water and space heating as well as for industrial application. Most of them currently available on the market are of the parallel tube type known as conventional flat plate solar collector[3]. They are relevants for high flow rate that requires high operational costs. Moreover, conventional flat plate collector had been in service for a long time without any significant changes in their design, shape and operational principles[4].

According to Matrawy & Farkas [5] configuration of a solar collector is an important factor that determines its thermal performance. Serpentine solar collector has the potential to perform better than a conventional parallel tube collector in low-flow systems due to the earlier onset of turbulent flow which enhances heat transfer application. Even for the same collector area, tube spacing and tube diameter, serpentine collector performer better than conventional collector[6]. According to Myrna & Beckman[7] the major reason for the difference in performance between conventional and a serpentine flat plate solar collector was the internal heat transfer coefficient. However in the past, serpentine flat plate solar collector was ignored due to large pumping requirements at higher flow rates.

Serpentine collector has geometry for which collector efficiency factor and heat removal factor cannot easily be expressed in a simple form. If thermal break is made midway between the serpentine tubes, then the collector can be analyzed as a conventional collector. If the break is not provided, reduced performance can be expected and more complicated analysis is necessary[8]. The heat removal factor for a serpentine collector is much more difficult to determine than for a conventional flat-plate collector[7]. Unlike analysis for conventional collector, there is heat transfer between the tubes for a serpentine collector.

Several paper with analytical solutions were published. All analytical solutions were done to treat the differential equations governing the heat transfer in a serpentine-tube absorber. Abdel-Khalik[9] found an analytical solution for heat removal factor of a serpentine tube bonded to the plate with two segments. He concluded that analytical solution for two segments was applicable for any number of segments with small error. Zhang and Lavan[10] showed that this conclusion was lead to much errors than predicted for a certain parameter range by obtaining analytical solutions for N=3 and 4. The solution ignores heat transfer application through its U-bend portion and assumes one-dimensional heat transfer in the plate.

Chiou & Perera[11] also analyzed the serpentine collector for any number of turns. As the number of turns increases, value of heat removal factor (FR) approach the values for turn N=1. In this case, the analysis for a long straight collector with no turns will hold. Therefore, the model is very close to the model for the flat plate collector, with the exception being that the internal heat transfer coefficient will be different [6].
There are only a few publications that report on experimental results of serpentine-flow solar collector. Eisenmann & Wiese[12] had conducted experiment on two serpentine collectors that have the same geometry and shape. In the first collector, serpentine tube was soldered to the absorber plate all through the collector, whereas in the second collector bends of the tubing was not thermally connected with the absorber. They put both over the sun under identical meteorological conditions and measured their performance. The efficiency of the collector that was soldered to absorber plate became about 2 to 2.5% superior in the experiment.

The experiment was conducted on the collector whose absorber plate was soldered to serpentine tube rather than striped plate that attached to tube. In fine and tube absorber collector arrangement, heat is normally transferred through absorber plate to tubes and then working fluids. As result, thermal diffusion due conduction mode caused throughout the system which reduces overall performance in the collector. Moreover, the experiment was unable to make predictions on the required parameters UL & F' of a serpentine-flow collector experimentally. Unlike parallel flat plate collector, there is heat transfer occur between tubes for a serpentine collector resulting in two dimensional heat transfer problem. Thus, it requires thermal break midway between serpentine tubes, and then the collector can be analyzed as a conventional collector. Consequently, coarser approximations need to be made in order to achieve analytical solutions for the absorber and fluid temperatures (Lund, 1989)[13] as cited by Eisenmann & Wiese.

Since computational fluid dynamics analysis of the flow and heat transfer in flat plate solar collectors is computationally quite difficult and cost, number of research works on this subject is quite low[2,14]. However, there was no adequate simulation work has been done so far for a serpentine collector. Thus, it was designed to perform simulation and experimental testing of serpentine flat plate solar collector. It is obvious that there might be certain limitations for experimental results thus data at each and every point but computational fluid dynamics (CFD) handles complex situations where experimental is not applicable because of limitations and cost effectiveness problem[15].

2. Material and Methodology

Geometry model was designed using ANSYS and the model get transferred to mesh. In meshing section, parameters of geometry part was defined. For better thermal and flow analysis, under mesh sizing, fining was selected to discretize flow further in to many elements and updated to recognize the input. Similar model was designed, manufactured and assembled for the experimental test. Thermocouple sensors were provide on serpentine tube and the plate in order to gauge thermal and flow distribution through the collector. Collector site orientation was adjusted and temperature distribution through the plate as well as water in the tube recited with digital multi-meter reader.

2.1 Development of Geometry &Meshing

Basic serpentine flat plate solar water heater was developed and going to be investigated. Unlike conventional collector, there is a complicated heat transfer network present in between serpentine tubes in the collector which results the analysis into 2-D heat transfer problem. Therefore, it is not easy to do the analysis of flow and thermal character inside the tubes and absorber plate. Thus, it requires appropriate thermal breaking application in the system. Breaking in the midway was one of the option but using more breaking system cause further complication on the setup of the boundary condition for decoupling and/or coupling system.

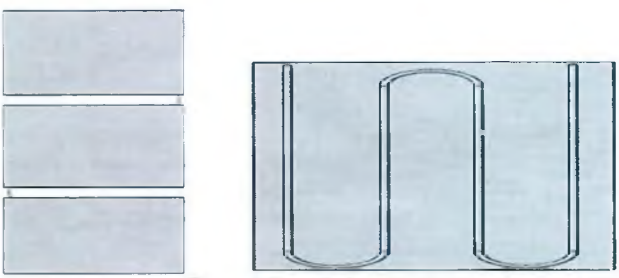

Figure 1.How designed striped geometry was decoupled from main system

Geometry employed for numerical investigation was illustrated as above and two thermal breaking lines were applied to simplify the analysis. The breaking lines are 20mm apart which separate adjacent plates adequately for possible losses occur due to heat conduction and moreover 20mm thick Styrofoam insulation was used in between two consecutive striped plates. Development of an exact computational mesh for the domain under investigation was paramount importance in CFD simulations. The accuracy of numerical results in CFD modelling was mesh dependent that means the finer mesh generally provides better results at the increased computational time [16]. Therefore the size of the mesh in the domain should be gradually increased to such level that the further raise in the number of control volumes does not result in considerable changes in theoretical or imaginary results obtained at the end of the exploration.

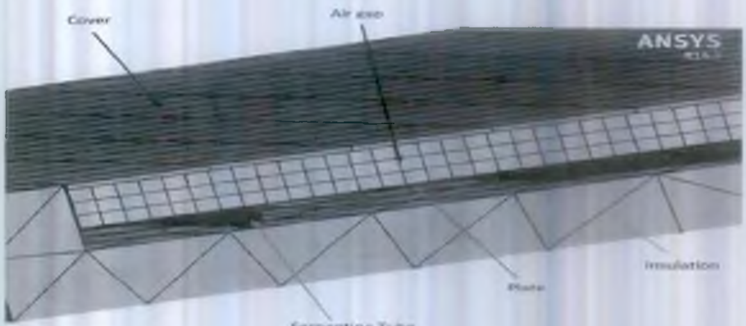

Figure 2. Computational grid of the system

Properties of materials employed for model were activated from the domain. The properties of water was used for a liquid domain in the computational mesh. Piecewise-linear functions were used to make into account the dependence of water properties upon temperature. The effect of gravity was taken into account along the vertical axis by specifying the negative acceleration value. Since flow was very small, laminar flow model with pressure-velocity coupling used.

The radiation heat loss to the sky was included in the boundary condition of the side wall and glass cover. The apparent sky temperature was calculated by using the equation from ASHRE hand book[15]. Heating of surfaces due to radiation and/or heat sources within the fluid phase can be included in this model. There are five radiation models which allow us to include radiation, with or without a participating medium in heat transfer simulations.

Solar load model are used to calculate radiation effects from the sun's rays that enter a computational domain. The model includes a solar calculator utility that can be used to construct the sun's location in the sky for a given time-of-day, date, and position. Solar load is available in the 3D solver only, and can be used to model steady and unsteady flows.

2.2 Prototype Manufacturing

Similar model of serpentine solar collector was manufactured with striped techniques where the plate was attached to serpentine shaped tube for separated segments. Striped mechanism prohibited or minimize intensity of thermal losses through the collector active part without doing the required work. Thermocouple sensor was provide on serpentine tube and the plate in order to gauge thermal and flow distribution through the collector.

Heat losses to neighbor or adjacent side of the collector system with conduction and convection processes. To interact with losses, thermal breaking line was formed between two adjacent plates with about 20mm part. That thermal line was covered with Styrofoam material which serve as insulator for heat to minimize air circulation in the collector. Since the collector rely on natural circulation system, significant amount of heat can be gained due to unique property of low flow rate.

Figure 3. Physical feature, how serpentine tube bend to required shape & kept in collector system

2.3 Experimental setup

Experiment setup was established in Jimma Agricultural Mechanization Research Center Lab and testing was conducted for over one month with outdoor condition. The collector was designed with two thermal breaking lines that divided the collector into three striped real model. And all the collector units were kept over the sun to trap incident radiation. Serpentine tube was soldered to the absorber plate to enhance thermal flow in favor of contact with aluminum sheet that has 0.8mm thickness. They are used to establish good heat transfer application between absorber plate and the tube and also insist heat transfer process from plate to transport fluid.

2.4 Collector Orientation

The collector position was adjusted to best performing angle of orientation. This should be as close as possible to Due South (0°) in the Northern Hemisphere for absorption of maximum solar irradiation. The surface orientation leading to maximum output of a solar energy system may be quite different from the orientation leading to maximum incident energy. The total annual energy received as a function of slope is maximum at approximately $\beta = \emptyset$ where $\emptyset$ is the latitude. For maximum annual energy availability, a collector tilt angle equal to the latitude is considered [8].

The performance of the solar water heater depends on prompt exposure to incoming solar radiation. Therefore, the solar collector should be far from block obstacles like tall buildings, trees or hills positioned in front and back side that hindered them to gain substantial solar energy.

Table 1. Collector Specification

No	Parameter	Symbol	Magnitude
1	Length of one serpentine segment	L	0.96m
2	Distance b/n tubes	W	0.14m
3	Plate thickness		0.8mm
4	Tube outside diameter	D	12.7mm
5	Tube inside diameter	Di	11.28mm
6	Plate thermal conductivity	k	46 w/m0c
7	Mass flow rate of water		0.0028kg/s
8	Collector area	Ac	1.52m2
9	Collector slope angle		100
10	Space b/n plate & glass	-	20mm
11	Thickness of back insulation	L_b	0.046m
12	Back insulation thermal conductivity	Kb	0.02w/m 0c
13	Thickness of collector	-	0.085m

3. Result and Discussion

3.1 Introduction

Under computational fluid dynamics case, solution was found and the required parameters were displayed in post-processing. Here by using CFD software temperature distribution inside flow tube and absorber plate of serpentine collector were predicted thereby to estimate collector efficiency factor and other parameters that express collector performance. Here flow characteristics and thermal performance of the serpentine solar collector was examined by experimental testing. Temperature distribution through the plate and working fluid has been followed with K-type thermocouple sensors. The sensor displayed the input in the form of voltage and the voltage was translated in to temperature using standard table.

Eventually, the result obtained with computational fluid dynamics simulation analysis was compared with experimental testing outcome for validation purpose.

3.2 CFD simulation

The numerical results obtained from CFD modelling of a serpentine solar collector model is presented as follow. As it was tried to explain in earlier, the collector system was divided in to three stripes because of thermal breaking system. Each striped model was engaged with detail operational and boundary condition considered for best simulation purpose. The properties of copper, aluminum and glass were applied for tubes, absorber plate and cover system respectively.
The properties of water were temperature-dependent and piecewise-linear functions were used to take into account dependence of water properties upon its temperature.
On top surface of absorber plate, equivalent heat fluxes of $650W/m^2$ was applied. Providing that the side and the bottom part of the plate was set to be at adiabatic condition. This heat flux was calculated from average based of solar radiation collected on June 30, 2014. For both CFD and experimental testing, heat fluxes of 650W/m2 was used to simulate temperature distribution though out absorber plate and water flow through the tube.

Collector thermal analysis was generally performed in the following ways. The first striped model was simulated with a given boundary conditions and output condition of this model coupled and decoupled to become input for the next stripe. Again the output of the second striped model was coupled and again decoupled to the last striped. In such away the solution for serpentine flat plate collector was analyzed until the result obtained become dissimilar. Figure5a demonstrated the temperature distribution pattern of the water in the tube for the last decoupled striped model.
This methods may works or be easy for fluid flow in the collector tube but might be difficult or may has limitation concerning with air, cover and insulation material that were thermally broken for analysis. For this case, uniform boundary conditions for air, cover and insulation were used for all stripes. For instant, wall boundary conditions with convection heat transfer coefficient of 20 w/m^2k was applied for air in all stripes. Because of the serpentine solar collectors was set to be inclined to horizontal, the effect of gravity was taken into account along the vertical axis by specifying the negative acceleration value of cosine 10^0 multiplied by the gravitational forces constant[16]. Collector angle was established based on recommendation made with literature. Experimental site has latitude of 7.7^0 and collector angle was made to be 100 for best annual solar radiation collection option without applying tracking system.

Figure 4. Temperature contour of the last striped serpentine tube

Figure 4 presents the temperature contours of water in the tube. As it can be seen from the graph, the water entered the serpentine tube of the last striped at an inlet temperature of 334k (which is exit temperature of second striped). Gradually as the flow passes through the tube, the temperature of the fluid at the exit of the tube became 342k.This result showed that there was better heat transfer process in the system. The numerical results obtained clearly demonstrate that the proposed design of the solar collector provides a considerable improvement of the performance.

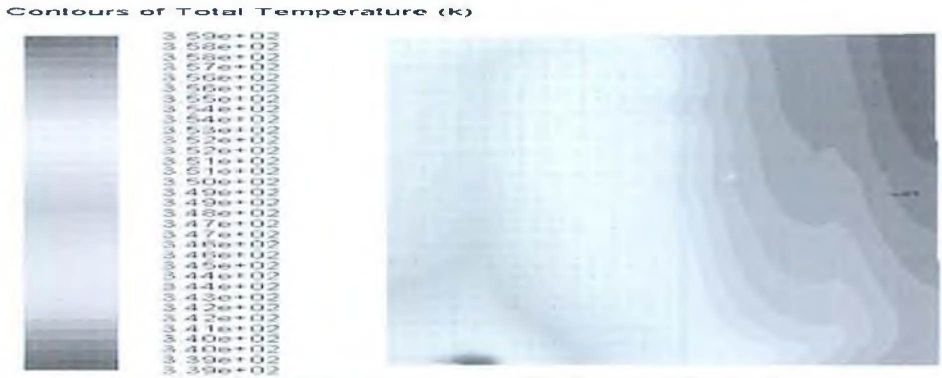

Figure 5. Temperature contour of the last striped absorber plate

Absorber plate has achieved maximum temperature of 360k that was small as compared with some literature. This was because solar radiation available during the time of experimental testing was of the smallest of the all months. In general, the test was conducted during the summer season when availability of constant solar radiation was inconvenient. Meanwhile as you increase the heat flux more and more, thermal distribution through the system became higher.
The same amount of solar heat flux was used to examine collector performance thoroughly for both numerical and experimental test activities. Since the main aim of this research is to investigate thermal performance characteristic of the serpentine flat plate solar collector with CFD simulation and experimental testing, similar condition has to be applied to perform comparison.

Figure 6. Pressure drop of water flow in serpentine tube of the last striped

As it can be seen from the Figure 6, high pressure drop was seen at collector inlet. Water moves down from the storage tank to the collector due to gravity but as soon as it leaves tanker and reach the inlet section, it faces the bend that become obstacle to flow.

As result it losses energy which causes pressure drop at the section. In general, pressure drop seen was an order of 124 Pascal. Only the last decoupled four segmented striped was analyzed here, in fact pressure drop through serpentine tube is higher when compared to conventional one.

The flow in the collector was induced by the density gradient, which in turn was caused by the heating of the working fluid. The figure in general shows the typical variation of the density inside the collector.

Figure 7. Density contours of water in serpentine tube

It can be seen that for the given boundary conditions the density of water changes from 886 kg/m³ at the top after heating process to 994 kg/ m³ at the bottom where the temperature is the lowest. Also from numerical results the water is cold more at the collector inlet section since higher water density results represent a lower water temperature in collector tube.

3.3 Leak test result

On the serpentine tube hole was prepared and the thermocouple sensor was attached with help of epoxy materials. Unless the sensors were attached with drilled holes tightly to tube, leak probably occur. Therefore to minimize the tendency of the leak, leak test was conducted with water-immersion methods. Leak was general omitted from water flow through the tube in whole collector system.
The smallest bubble an operator could detect has 1 mm radius and that the waiting time is 30 seconds [17]. Assuming that the pressure inside the bubble is at atmospheric pressure, it can be stated from the previous equations that the bubble volume is and therefore the minimum detectable leak rate at Jimma atmospheric condition become

3.4 Estimation of Solar Irradiance

The thermal performance of natural circulation solar water heating was tested on June 30, 2014 at Jimma Agricultural Mechanization Research Center using serpentine flat plate solar collector setup with batch type water tank. An estimation of solar insolation of that experimental site was made by employing engineering equation solver (EES) software. A program that compromised important parameters of air with non-dimensional units was developed. Data's were collected on the temperature of air, sky and black body with ten minute intervals. Since once the program was developed, the collected data was inserted in the program and manipulated. Figure 8 shows the profile of solar radiation available that was calculated based on data of the test day. Since season was summer, it has been difficult to get a clear day to do performance test and several trials have been done to get good daily solar curve. This data was one of the best clear days for performance test.

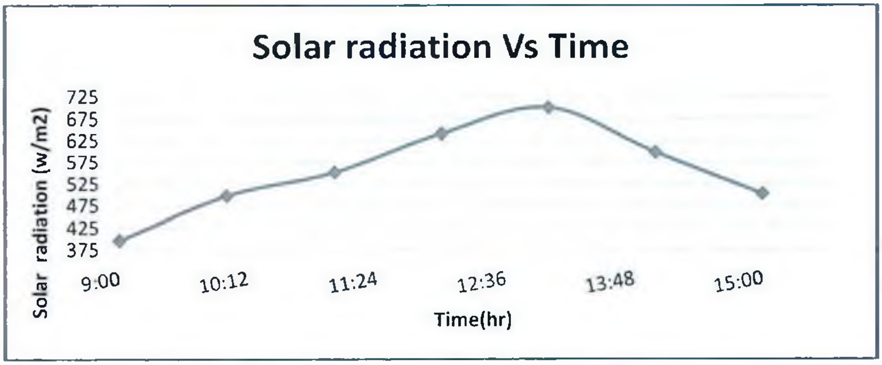

Figure 8. Solar radiation on June 30, 2014 at Jimma Agricultural Mechanization Research Center

Due to decreases in solar radiation, the ambient temperature also starts to decrease. Ambient temperature variation of the experimental site also has the following pattern.

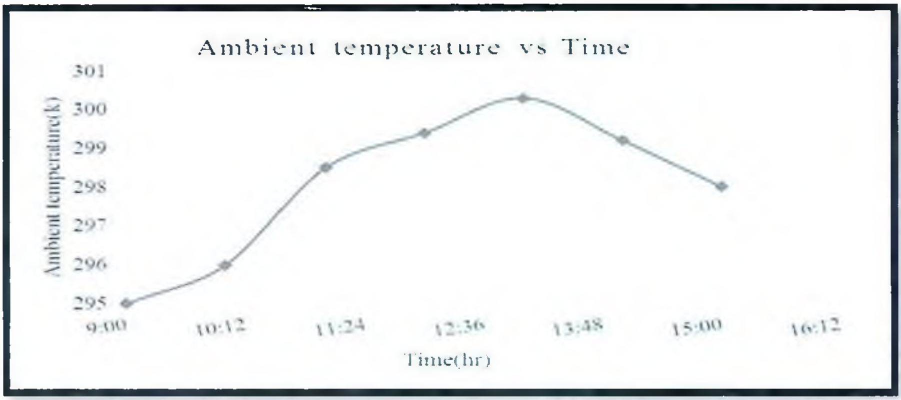

Figure 9. Ambient temperature variation of the experimental site

3.5 Experiment result

Temperature distribution through the plate and water in the tube were measured using k-type thermocouple. Since the thermal breaking system was applied, absorber plate was detached from each other and fixed to tube with soldering techniques. On the serpentine tube, this temperature sensors were attached on holes so as to read temperature distribution through the flow. In addition, the sensor was attached on the surface of plate to measure how hot could absorber surface be.

Table 2 below displays temperature of water flow in the serpentine tube. The collector system was divided into three stripes as result of thermal breaking application and on each strip as per the thermocouple sensor allocated, data has been registered with in 20 minute intervals. Different reading of temperature was recorded at every hour and average values of water temperature are considered. Here temperature was read in terms of voltage induced caused due to temperature difference between conducting wires in sensor. Using this induced voltage, corresponding temperature was read using standard table including ambient temperature.

Table 2. Temperature of water in the tube at various points

Z(m)	0.09	0.16	0.30	0.42	0.56	0.83	0.97	1.12	1.26	1.48	1.47	1.52
X(m)	-	0.48	0.93	0.03	0.93	0.48	0.03	0.48	0.93	0.48	0.03	0.93
Tw (K)	295	298.2	302.5	317.4	317.6	319.3	320.7	322.9	323.3	323.5	336.9	317.4

Table 3 displayed average plate temperature distribution in the absorber plate. Temperature of the plate was computed on average based and displayed here at every ends of the strips. Four temperature values were shown as per the stripes.

Table 3. Absorber plate temperature at various points

Z-axis(m)	0.295	0.68	1.15	1.55
X-axis (m)	0.45	0.57	0.35	0.48
Plate temp(k)	334	345	351.5	348

Heat removal factor of the collector (FR) can also be calculated for the experimental testing. Collector inlet temperature of the water was estimated from experimental testing and become 293k and collector exit temperature is 336.9k as well as mean temperature plate is 351.5k from collector experimental data. Therefore, UL becomes 4.98w/km^2 and F_R becomes 0.780878.

Table 4. Collector removal factor at solar heat flux of 650w/m^2

Parameters	CFD	Experimenta	Analytical
Heat removal (F_R)	0.8783	0.7809	0.8190 (Al)
Efficiency factor (F')	0.7476	0.5879	-

Figure 10 indicated that how temperature of the flow was varied with location. This figure displays temperature distribution of water flow in tube along z-axis. Water was admitted to the inlet of collector at temperature value of 20 ^{0}C. As it passes through the tube, heat transfer takes place so that water start gaining considerable heat from the process. As it can be seen from the graph, the water in the tube extract little amount of heat in first strip since it goes short distance to complete loop.

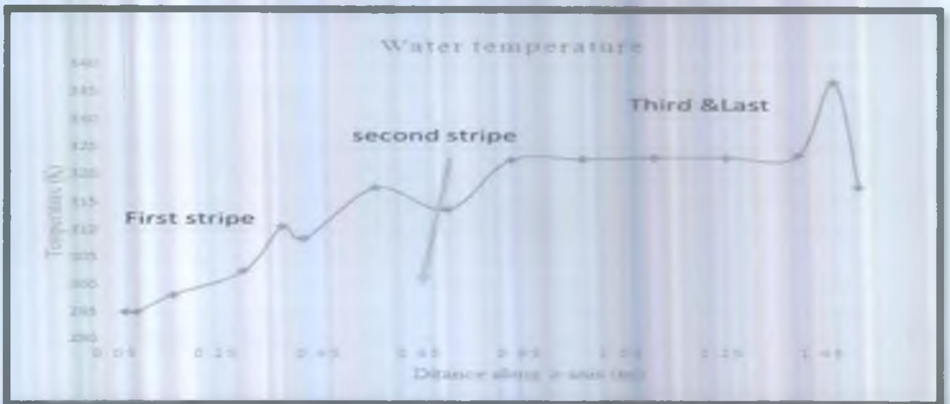

Figure 10. Temperature variation of water in the collector

In the above diagram, the maximum temperature of the water in the first strip was 310k but as it passed to the next strip, considerable amount of heat was enabled to be gained. Due to possibility of long contact time with the tube surface that causes good heat transfer process in the system. As result, water in the tube gained significant amount of heat. The water in the tube attained maximum temperature of 337k throughout the system. Figure11 displayed below demonstrated collector plate temperature distribution in whole strips. Temperature variation is taken in average based at each end of the strips. In second and third strips, collector plate temperature was sharply increasing and attain maximum temperature of 351k at the middle third strips. Here solar radiation directly fallen on the absorber plate and there was no apparent shading effect. For last strips, there is possibility of self-shading effect for time before noon. Even though water storage tank was suited above the collector, it affected thermal performance of the last strip. The graph is drawn using result obtained at 14hr.

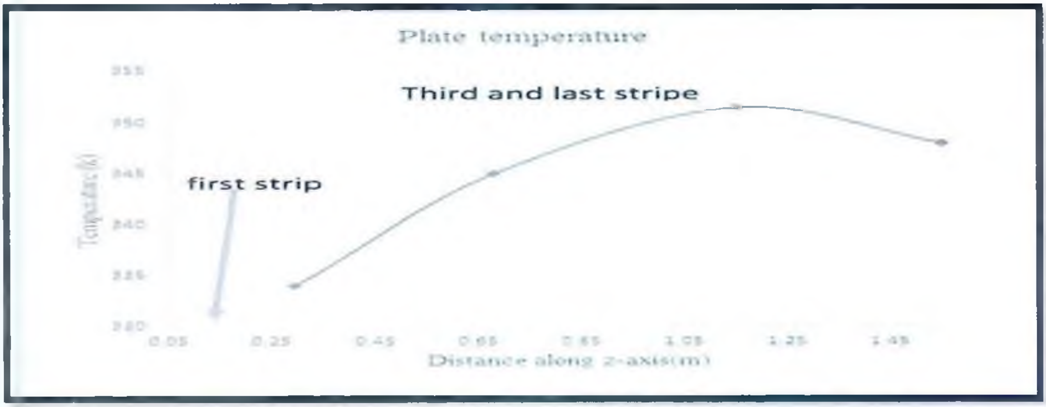

Figure 11. Temperature distribution in absorber plate

Table 5 below has displayed the summary of water of temperature at tube exit obtained by CFD simulation and experimental testing .The data's were kept here to comparison the result obtained with both simulation and testing.

Table5. Summary of comparison of water temperature in tube obtained by Computational Fluid Dynamics & experimental result

Time (hr)	Solar Intensity (w/m2)	Amb Temp(k)	Water temp by CFD(K)	Water temp by experiment(k)
9:00	396	295	315	305
10:00	497.6	296	320	317.73
11:00	552.2	298.5	322	313.73
12:00	641.8	299.4	326	322.77
13:00	648.6	300.3	328	323.38
14:00	700.5	299.2	338	336.9
15:00	562.5	298	320	318.43

Figure 12. Comparison of temperature of water at collector exit with CFD & experimental

Collector instantaneous efficiency was found to vary according to the external conditions i.e. solar radiation, ambient temperature and mean tanker temperature. The efficiency curve of experimental results agreed with model. The collector system showed higher efficiency at low tank temperature and the efficiency decreases as the tank temperature increases. An average tank temperature was 322k and FRτα is the y-intercept in the efficiency line and FRU_L represents the slope of the curve. The higher the slope, the higher is the sensitivity to external conditions. Both experimental and model collector curves have high slope value which indicates the efficiency is very sensitive to external condition. The heat transfer coefficient U_L also depends on the mean plate temperature.

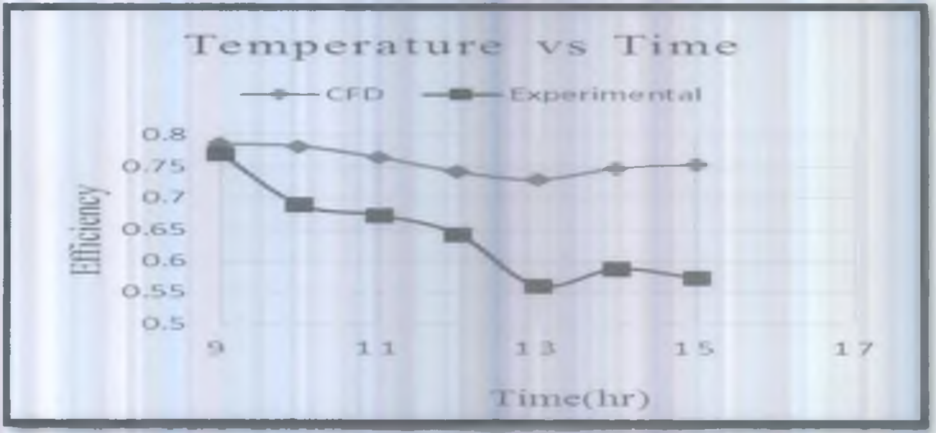

Figure 13. Collector Instantaneous efficiency during test hours

Khalifa[18] experimentally investigated the impact of mean plate temperature on the total and collector top heat transfer coefficient. It was found the total and collector top heat transfer coefficient varies during the test hours due to the increase solar radiation and the mean absorber temperature. Higher collector inlet temperature increases the mean plate temperature and increases the collector total heat transfer coefficient which reduces the efficiency of the collector.

4. Conclusion

Thermal performance of flat plate solar collector can be improved by altering its configuration of heat transport system from solar absorber to heat storage system. Solar collector size, shape and flow rate in general affect system performance. Serpentine flat plate collector was designed based on thermal breaking system that was recommended by different literatures. Unless thermal breaking was applied, it is difficult to model formulae that enable to analysis as conventional collector. The performance of this collector was investigated by numerical simulation methods. Collector model was designed applying ANSYS 14.5-FLUENT software. The software was used to identify the optimum configuration of the most economical serpentine flat plate solar water heating systems.

For the collector mass flow rate of 0.00285kg/s and solar radiation of 650w/m^2, temperature of absorber plate(Tp) and water at collector exit (T_0) became 360k and 338k respectively. The same collector model was manufactured and experimental investigation was carried on with similar conditions as did for simulation. Consequently,absorber plate (Tp') and water at exit of the collector(T_0') during the experimental test attained maximum temperature of 353k and 336.9k respectively.

Collector model has three stripes and while both CFD simulation and experimental test was taking over, all strips exhibited varies temperature distribution in the collector system. For experimental session, solar radiation is directly fallen on the absorber plate and there is no apparent shading effect seen especially for the time before noon in second strips whereas in third and last strips, there is possibility of self-shading effect observed for time before noon. Even though water storage tank was suited above the collector, it affected thermal performance of the last strip. Consequently, such temperature variation was happened in the strip.

In general, numerical and experimental results obtained were found with good agreement with some deviation. In numerical case, serpentine flat plate solar collectors outperformed better when compared to experimental due to some experimental imperfectness during data collection and instant variation of solar insolation. Numerical results obtained demonstrated that striped absorber plate design would improve the heat transfer from the walls of tubes to the liquid which results hot water production of the solar collector good.

5. Recommendation

Serpentine solar collector requires further study to model the science exist behind the collector. So far different studies were conducted on the collector to model collector's important parameters using analytical methods for several turns. Yet modelled collector heat removal factor (FR) formula was invalid for all materials.

As we all know that serpentine solar collector has geometry for which collector efficiency factor and heat removal factor cannot easily be expressed in a simple form. Unlike the analysis for the parallel flat-plate where the fins between the tubes are assumed adiabatic at the center of the tube spacing, there is heat transfer occur between the tubes. However no analytical modelling was formulated for collector efficiency factor for serpentine solar collector. It is unable to make predictions on the thermal output of a serpentine-flow collector with experimental.

Thermally breaking the collector system in to more than two stripes requires serious attention in settling required boundary conditions. In this research, two thermal breaking line was applied and each stripe was analyzed separately and later coupled. Accordingly, output of the first stripe became input for the second stripe and output of the second stripe became input for the third stripe. In such manner, analysis of collector system was done and finally coupled together to become the collector system. This methods may works for fluid flow in the tube but might be difficult with air, cover and insulation material that were thermally broken for analysis. Therefore, more effort is required to model (F') to analyze the thermal behavior of the tube bends. Moreover, detailed numerical investigations should also be taken into consideration.

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