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FARM MACHINERY INDUSTRIAL RESEARCH CORP.

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i. Indicate by footnotes or legends any abbreviations or symbols used in tables or figures.
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k. Round off numbers, if possible, to one or two decimal units, e.g., 45.5 kg/ha instead of 45.4762 kg/ha.
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EDITORIAL

On 11th of March, 2011 2:46 PM, a disastrous earthquake with an epicenter located on the ocean of Tohoku, struck Tohoku and Kanto area of Japan with an unprecedented magnitude of 9.0. Many areas in Japan had experienced tsunamis before and thought they were ready for it. However, the one that struck this time was far more tremendous than anyone in Japan had predicted. It washed away many towns and left them in ruins. The death toll continues to rise and is now more than 30,000.

Also, this huge quake resulted in catastrophic damage to the nuclear plants in Fukushima because the tsunami height was much higher than predicted and struck and broke the refrigeration system inside. This caused a breakdown of the nuclear reactor and resulted in a serious radiation leak. This radiation pollution affected many agricultural products that were suspended from shipment. It was a horrible incident, indeed. The reduced electricity power has resulted in rolling blackouts that has caused many companies to be unable to produce and serve as usual.

However, under such circumstances, many helping hands came from other areas in Japan, and even countries abroad. These warm regards cheered up many Japanese citizens. Our company has also received many heartwarming letters from the readers and co-editors of AMA. We convey thankfulness to all the kind words.

Japan overcame many disasters from the past. It is time to overcome once more, making a brand new and a better country as we cooperate with the people of the world. What we have learned from this catastrophic earthquake is that we should always be ready for unpredictable disasters. We especially have felt that each family should keep firmly in mind that food and water should be saved as a stock for when they are not available, and not buy them up after something has happened. Food-sufficiency in Japan is about 40 % in calorie-base, which is extremely low, and we must do something to increase this number to be ready for another disaster.

As the demand for food has grown rapidly in developing countries, the balance of food supply and consumption has started to collapse. Therefore, people in various countries have started to suffer from the rising cost of food. We should increase the production of food on a global level, but the farming land on earth is limited. We must develop the productivity of the land and increase the effect of timely operations. Promoting agricultural machinery will be very important as related to the growth of food production. Therefore, we should research, develop, and produce agricultural machinery in all areas around the world and not limit this to just some countries. I would like to continue cooperating with the readers of AMA to accomplish this great mission.

Thank you all once again for the kind and caring help from all over the world.

Yoshisuke Kishida
Chief Editor

April, 2011

CONTENTS

AGRICULTURAL MECHANIZATION IN ASIA, AFRICA AND LATIN AMERICA

Vol.42, No.2, Spring 2011

Effect of Selected Tillage Implements on Physical Properties of Two Types of Soils in Khartoum Area (Sudan)

by

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Abstract

The effect of three types of tillage implements (chisel, offset disc harrow and ridger) on bulk density, porosity, aggregate stability and penetration resistance of sandy clay soils at two locations in Khartoum state were investigated. Bulk density of the soil surface layer was reduced by all tillage treatments compared to no-till. For all treatments, bulk density was increased with depth and the values were higher for the clay soil. Soil porosity for all treatments decreased with depth and the highest values, 56.5 % in sandy clay and 53.5 % in clay soil, were at the surface layer (0-15 cm) with the offset disc harrow. Bulk density and porosity were generally interrelated and simple regression analysis showed a high correlation between the two parameters in both soils for the different treatments (R^2 = 88.6 % and 99.1 %). Highly significant difference at the 1 % level was observed between the effects of different tillage treatments on average porosity percentages. Soil aggregate stability was higher for the clay soil compared to sandy clay and the highest value was with the no-till treatment as 54.7 % at a soil depth of 30-45 cm. The offset disc harrow had the lowest aggregate stability percentages for both soils. Penetration resistance of the upper soil depth, 0-15 cm, was significantly reduced by tillage treatments compared to no-till in both soil types. The highest reduction was with the offset disc harrow as 31 % in sandy clay soil and 52 % in clay soil. In general, soil penetration resistance increased with depth and the values were higher in clay soil. Multiple regression analysis showed high correlation between the average penetration resistance and the average bulk density a with aggregate stability percentages (R^2 = 97 %).

Introduction

There are many tillage implements that are used for soil bed preparation to achieve the most useful tillage objectives and improve soil physical properties. Soil manipulation for land preparation using different tillage system and implements is the most expensive among energy input components of agricultural crop production practices (Igbal *et al.*, 1994 and Coates and Thacker, 2001). Selection of effective implements and optimum tillage practices for land preparation may reduce cost of the energy budget and can maintain or increase crop production (Erickson and Larson, 1983). The variations in adapting different tillage implements in Sudan may be due to soil type, crop planted or socioeconomic factors. Selection of the tillage implement should be based on available resources like power source, soil and crop type. The main objectives of using tillage implements are to reduce soil bulk density, increase soil porosity, favor water infiltration and produce good soil tilth (Lal, 1995).

Although the effect of some tillage implements on some soil properties may be clearly observed, it is difficult to predict all effects as its physical and chemical characteristics will be affected by the soil manipulation process (Michael, 1978). The effect of tillage implements as individual or in groups were studied (Karayel and Oznerzi, 2003 and Rahmani and Salokhe, 2001). Cavalien *et al.* (2006) studied the effects of different tillage systems on soil physical properties like bulk density, penetration resistance, and aggregate stability in sandy and sandy loam soil. Controversial effects were observed (Franzluebbers *et al.*, 1995; Ray and Gupta, 2001; and Chenu *el al.*, 2000). Coates and Thacker (2001) reported that tillage operations can contribute significantly to soil compaction, dust emission and degradation of the environment and may lead to wind and water erosions (Melvin, 2005). Improper selection and unjustified use of tillage implements may destroy the root zone in the soil and waste fuel and energy inputs for crop production (Hussein and Munir, 1986 and Melvin, 2005). Therefore, the selection of suitable tillage implements that match soil type and conditions is of great importance to have proper soil tilth and reduced energy cost of operations (Bowrs, 1989; Balel and Dahab, 1997; and Dahab and Hebiel, 2007).

The objectives of present study were to investigate the effects of three tillage implements (chisel plough, offset disc harrow and ridger) on the physical properties of two types of soils (sandy clay and clay).

Methods and Materials

The experiment was conducted in two different types of soil. The first soil was at the demonstration Farm of the Faculty of Agriculture, Shambat, University of Khartoum and classified as sandy clay. The second soil at Abu—Halima agricultural project was classified as clay. Both soils were in Khartoum state at a latitude of 15º 40' N and longitude of 32º 32' E. Some physical and chemical characteristics of the two soils are shown in **Table 1a** and **1b**.

A Massey Ferguson (F165) tractor (54.8 kW PTO power) and three tillage implements were used in this study. Other materials and equipments included a stop watch, measuring tape (30 m), pieces of chalk and a one liter graduated cylinder for refilling the tractor tank with fuel.

An area of 1,600 m^2 (64 × 25 m) was selected in each location and was divided into plots (20 × 7 m). The plots were separated by a distance of two meters with head lands left at the ends of blocks for turning during tillage operations. The treatments included three tillage implements (chisel plough, disc harrow and ridger) replicated three times giving a total of nine plots. A completely randomized block design was used with treatments randomly distributed in each black.

Soil bulk density was determined using the clod method as described by Blake and Hartage (1986) as follows:

$$B.D.\ (gm/cm^3) = Pw\ Wov / Wa - Wcv + Wca - (WcaPw / Pp)$$

Where:
Pw = density of water
Wov = oven dry weight of clod
Wa = clod weight in air
Wcw = coated clod weight in water
Wca = coated clod weight in air
Pp = density of coating material

Porosity was calculated by first determining soil particle density based on Archimedes principle and according to the formula of ASTM (1958):

$$Particle\ density = Pw\ (Ws - Wa) / [(Ws - Wa) - (Wsw - Ww)]$$

Where:
Ws = weight of pycnometer plus oven dry soil sample
Wa = weight of pycnometer filled with air
Wsw weight of pycnometer filled with air and water
Ww = weight of pycnometer filled with water

Total soil porosity was calculated thereafter by the equation described

Table 1 Some physical and chemical characteristics of experimental soils

Soil (a): Sandy clay

Soil depth cm	Mech. Sand %	Analysis Silt %	clay %	MC %	pH	ECe ds/m	SAR	B.D gcm^{-3}
0-15	48.4	11.5	40.1	5.1	7.8	0.57	2.09	1.26
15-30	45.8	10.3	43.9	5.3	7.8	1.57	4.49	1.28
30-45	46.4	10.0	43.6	2.4	7.9	0.78	3.18	1.30
45-60	45.2	10.0	44.8	4.7	7.9	0.95	4.66	1.34

Soil (b): Clay

Soil depth cm	Mech. Sand %	Analysis Silt %	clay %	MC %	pH	ECe ds/m	SAR	B.D gcm^{-3}
0-15	40.2	13.5	46.3	7.1	7.7	3.05	5.58	1.31
15-30	40.3	11.2	48.5	7.3	7.8	3.20	5.69	1.33
30-45	38.8	12.2	49.0	7.7	7.7	3.55	6.20	1.35
45-60	38.7	10.1	51.2	7.8	7.8	3.60	7.12	1.39

Table 2 Duncan multiple range tests of mean parameters measured as affected by different types of treatments

Parameters	Sandy clay soil				Clay soil			
	No till	Chi.	Harr.	Ridg.	No till	Chi.	Harr.	Ridg.
B.D.	1.30a	1.29a	1.29a	1.30a	1.35a	1.32a	1.33a	1.34a`
Porosity	51.9a	53.2b	53.1b	52.1a	48.1a	48.5a	49.6b	48.4a
Agg.Stab.	38.1a	35.6b	33.9c	36.0b	47.6a	44.2b	41.8c	41.6c
Pent. Resist.	7.95a	7.60b	7.50b	7.70b	10.0a	9.13b	8.55c	8.80c

Means followed by similar letters in a row are not significantly different according to Duncan Multiple Range Test by Danielson and Suthertand (1986) as follows:

Soil porosity = [1 − (Bulk density / Particle density)] × 100

Soil Aggregate stability was identified and determined using the method described by Rohoskova and Valla (2004) with the following formula:

Soil Agg. Stab. (%) = (Ws × 100) / (Ws + Ww)

Where:

Ws = weight of soil aggregates sample in dispersing solution

Ww = weight of soil aggregates sample in water

s = dispersing solution (calgon)

Soil penetration resistance was measured before and after tillage operations using a hand operated soil test penetrometer (**Fig. 2**). Penetration resistance in each plot was measured by placing the cone on the soil surface with the top rod oriented vertically (Herrick and Jones 2002).

Results and Discussion

Effect of Tillage Implement and Soil Type on

Soil Bulk Density

At the upper soil depth (0-15) cm the tillage implements reduced the bulk density more than the no-till treatment for both soils. Chisel and ridger implements reduced the bulk density of sandy clay in this layer by 2 % compared to no-till treatment while the offset disc harrow reduced the density by 3 %. In clay soil the bulk density at the same upper depth was reduced by 2 %, and 1 % when using chisel, offset disc harrow and ridger implements, respectively. In general the bulk density increased with depth in both soils (**Fig. 1**) and were higher in clay soil compared to sandy clay soil. The statistical analysis showed no significant difference between the treatments for both soils.

Soil Porosity (%)

Soil porosity, in percentage, of both soil types decreased with depth and the highest porosity values were at the surface soil depth (0-15 cm) (**Fig. 2**). The highest porosity percentage was 56.5 % for the offset disc harrow with sandy clay soil and 53.5 % and for the clay soil. Tillage implements increased porosity (%) compared to no-till treatment for both soils especially in the upper two soil depths (0-15 cm) and (15-30 cm). This may be due to the effect of tillage implements in breaking soil particles and increasing soil pores. The lowest porosity (%) valves were at the soil depth of 45-60 cm for all treatments and both soil types.

Generally, soil porosity and bulk density were interrelated and porosity tended to increase with decreased bulk density. Simple regression analysis showed high correlation between porosity and bulk density for different treatments and in both soils (R^2 = 88.6-99.1 %). This was in line with what was described by Chen *et al.* (1998) and Bukhari *et al.* (1992). There were highly significant differences between the effect of different tillage implements on average porosity percentage values

Fig. 1 Effect of tillage implement type on soil bulk density (a) Sandy clay (b) Clay

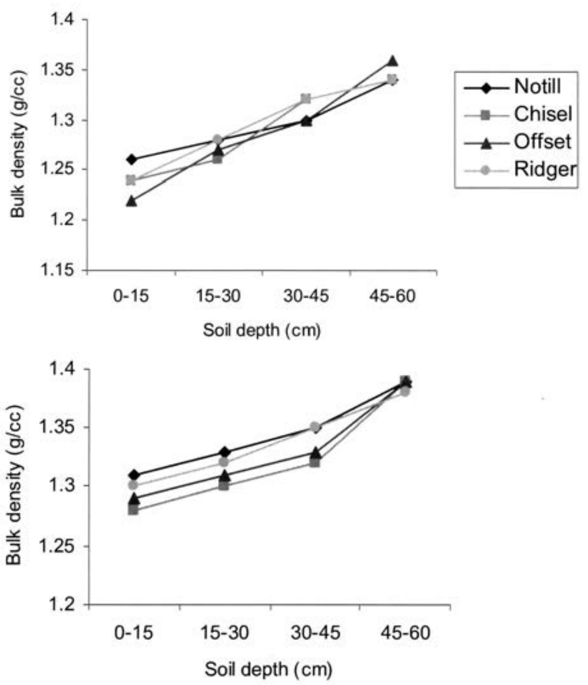
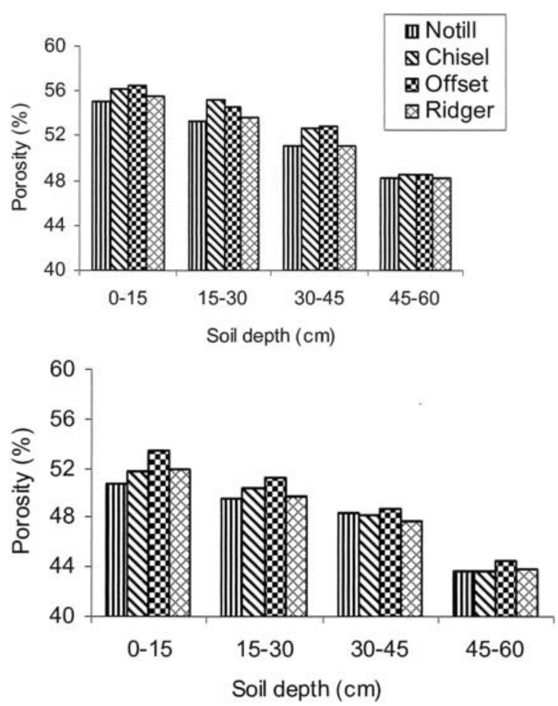


Fig. 2 Effect of tillage implement type on soil porosity (a) Sandy clay (b) Clay



at the 1 % level of for both soils (**Table 2**).

Soil Aggregate Stability (%)

Soil aggregate stability percentages were higher in the clay soil than in the sandy clay for all depths. The highest aggregate stability percentage was with the no-till treatment in the clay soil (54.7 %). At the upper soil layer of 0-15 cm, tillage treatments decreased the aggregate stability compared to the no-till treatment. Generally, the offset disc harrow had the lowest aggregate stability percentages for all depths in both soils (**Fig. 3**). This could be due to cutting and soil converting actions of the disc blades (Smith and Willces, 1986). At the upper soil depth of 0-15 cm for the sandy clay soil, the aggregate stability percentage was reduced by 15.9 %, 22.1 % and 16.4 % for the chisel, offset disc harrow and ridger, respectively, compared to the no-till treatment. In the clay soil it was reduced by 11.2 %, 17.5 % and 14.2 % for the above tillage implements in sequence.

Statistical analysis indicated highly significant differences at the 1 % level between the effects of treatments on average aggregate stability percentages for both soils. Multiple regression analysis of average aggregate stability percentage with average soil moisture content and bulk density of all treatments revealed high correlation (R^2) for both soils. It was 84.2 % for sandy clay and 92.4 % for clay soil.

Soil Penetration Resistance

Soil penetration resistance (cone index) of the upper soil depth (0-15 cm) was significantly decreased by the three tillage implements compared to the no-till treatment for both soils. The highest reduction in penetration resistance at this depth was with the offset disc harrow; 31 % in the sandy clay soil and 52 % in the clay soil. At deeper soil depths penetration resistance was increased and the highest cone index values were recorded at 15-30 cm and 30 -45 cm soil depths for both soils (**Fig. 4**). This was in line with the findings of Saber and Mrabet (2002) and Rahmati and Salokhe (2001). Statistical analysis showed non-significant differences between the mean values of treatments of sandy clay soil but highly significant differences at the 1 % level for clay soil (**Table 2**). A comparison of average cone index values showed that the chisel, offset disc harrow and ridger reduced the cone index by 10 %, 15 % and 13 %, respectively, in the clay soil. In general, the penetration resistance values were higher in clay than sandy clay soil, which could be due to characteristic properties of the clay soil. Multiple regression analysis showed that porosity and moisture content, jointly for the different treatments, accounted for 96.3 % and 93.2 % of the penetration resistance variability in sandy clay and clay soils, respectively. High correlation between the average penetration resistance and average aggregate stability percentage and bulk density in both soils was R^2 = 97 %.

Conclusions

The following conclusions were drawn from the present study:

Fig. 3 Effect of tillage implement type on soil aggregation stability (a) Sandy clay (b) Clay

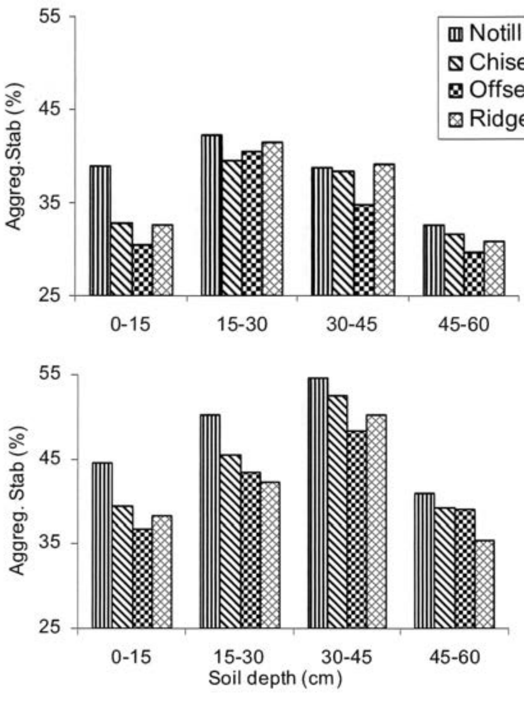
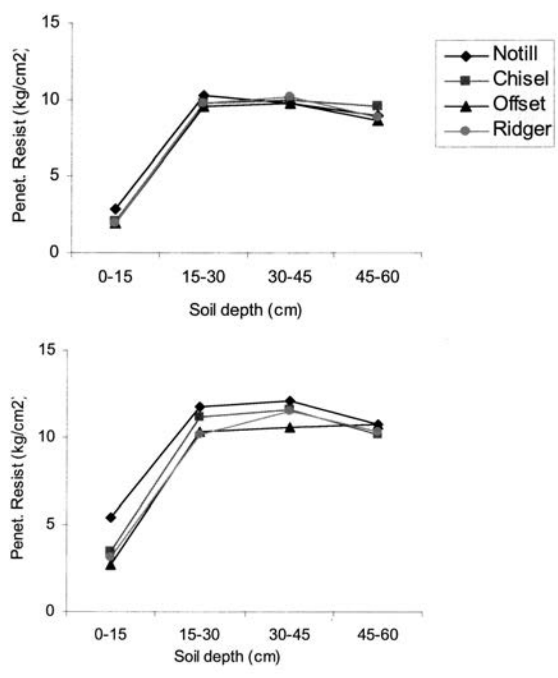


Fig. 4 Effect of tillage implement type on soil penetration resistance (a) Sandy clay (b) Clay



1. Tillage implements decreased the bulk density and penetration resistance and increased the porosity and aggregate stability of the upper soil depth (0-15 cm) for both soil types.
2. Higher aggregate stability (%) and penetration resistance (g/cm^2) values were recorded at the soil depth of 30-45 cm
3. The offset disc harrow was the most effective implement in altering the upper soil depth (0-15 cm) condition compared to the chisel and ridger implements for both soil types.

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Comparative Field Evaluation of Self-Propelled Paddy Transplanter with Hand Transplanting in Valley Lands of Kashmir Region

by

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Abstract

Paddy is most important and widely grown food grain crop of the region. The paddy is transplanted manually using root washed seedlings, which is labour oriented and involves high cost of transplanting. Suitable mechanization technology is needed to reduce drudgery, cost of transplantation and enhance timeliness of operation. A self propelled riding type paddy transplanter was tested for its feasibility in the small size fields of the region during kharief season of 2007-08. Mechanical paddy transplanting by self-propelled transplanter has ensured proper crop stand and has given 6.0 % increased yield compared to hand transplanting. The operation of the transplanter revealed that it needs fairly plain land and a well puddled field to maintain uniform depth of standing water and proper fixation of seedling in the ground. The cost of transplanting was a little higher than hand transplanting but it could be reduced by increasing the working hours of the machine. Hence, the mechanical paddy transplanter may be very useful in the region during labour scarcity and, also, to enhance the timeliness of operation for large land holdings.

Introduction

Paddy is the main cereal crop of the Kashmir. It accounts for 25.2 % of the area and 35.2 % of production on a total food grain basis. During 2006-07, it was cultivated on an area of 252.52 thousand hectare with a production of 554.6 thousand tonnes and productivity of 2,196 kg/ha (Anon, 2007). Valley lands are mild sloppy areas and allow the water to flow from upper to lower side in such a way that there is continuity of flow from each field. However, terraces are located on higher slopes. The rice is grown in these types of fields and the most widely adopted cultural practice of taking rice is by transplanting the root washed seedlings in the fields by manual labour only. It consumes maximum time, energy and cost as well as increase drudgery to the farmers and farm labourers. Also, a majority of the farming community has less than an average of 0.5 ha land holding and cannot economically afford tractors and big machinery because the main crops are grown in small terrace fields where easy access is not possible for tractors. During the peak season of transplanting, there is an acute shortage of labour, which normally results in delaying of transplanting of paddy. Hill distance and depth of planting are equal and accurate in mechanized transplanting and allow proper sunlight,?air and application of fertilizer that results increased in yields. This has set the need for mechanization of transplanting of paddy in hilly and valley land also. In order to assess the possibility of mechanization of the transplanting operation, an eight row self propelled paddy transplanter was tested for its performance in valley lands and these were compared with the hand transplanting method existing in the area.

Materials and Methods

The performance of an eight row self propelled paddy transplanter was tested in the valley lands (**Fig.**

1). The transplanter consisted of a front mounted engine, power transmission system, main frame, float and rice planting system. The rice planting system included a curved tray with eight partitions, planting arm with fingers, platform, tray guide and transmission box. The weight of the machine was supported by wheels and weight of the planting portion by a float that slides on the surface of the puddled field. The row to row spacing was 238 mm while three different hill spacing; viz. 120-140, 140-170 and 170-200 mm was available in the machine. The depth of transplanting was adjustable according to the field conditions and it varied from 0-60 mm. The transplanter was fitted with a 2.94 kW engine and the power derived from the engine was transmitted to the clutch section. The speed was reduced in the transmission and the power was further transmitted to the front and rear axles. An eight sectional platform provided in the rear of the operator oscillated sidewise on a reinforced rail by feed screw and guide. The feed screw had a spiral groove in both the directions on its surface. When the shaft rotated, the guide on the spiral grooves moved left or right. When the guide reached the flat section at the end of the feed screw it automatically turned to change the direction of movement. Rotary planting arms were provided in the transplanter. By increasing or decreasing the speed of the planting arms, hill to hill distance could be changed. The number of seedlings per hill was dependent on density and thickness of seedlings. The transplanter was also provided with two pneumatic wheels for transportation out of field. The detailed specifications of the machine are given in **Table 1**.

Raising Mat Type Seedlings

For mechanical transplanting of rice, mat type seedlings and optimum puddle with levelled bed conditions are two pre-requisites for efficient operation of a transplanter in terms of placement of seedlings. For raising mat type nursery (**Fig. 2**), the frames of MS flats with 8 compartments were fabricated in such a way that each compartment was 500 × 220 × 20 mm. Nursery was raised on 19-5-2008 at the University field. Soil, sand and FYM were mixed thoroughly in the ratio of 4 : 2 : 1. The MS frame was kept over a polythene sheet on a levelled field and filled with mixture of soil, sand and FYM up to height of 15 mm in the frame. About 650-700 gm (at 0.50 kg/sq. ft) of pre-germinated seeds of rice variety Shalimar rice-1 (SR-1) was spread evenly in each frame. Then seeds were covered with a thin layer of soil mixture and water sprinkled with a watering can for proper setting of the soil. The frames were removed after 20 minutes and put up to the next place. The seedbed was mulched with a plastic sheet until germination of seeds for protection

Fig. 1 Schematic diagram of Self Propelled Rice Transplanter

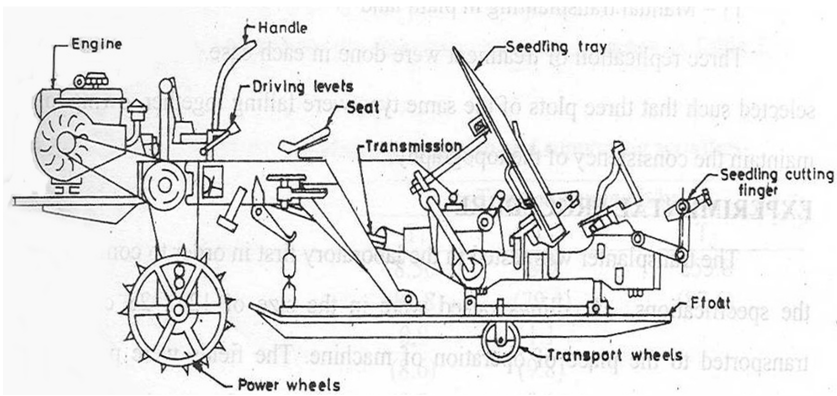
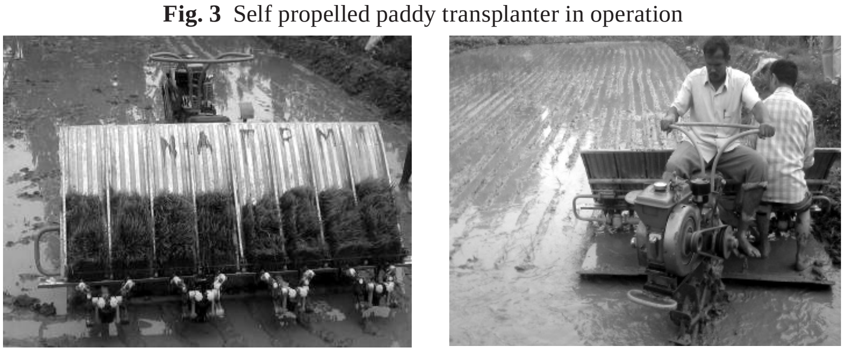


Fig. 2 Mat type nursery raising for self propelled transplanter

Fig. 3 Self propelled paddy transplanter in operation



from the birds. Water was applied daily with a watering can for one week and thereafter water was applied by the flooding the bed. The recommended dose of fertilizer was also given to the nursery. After 28-30 days from the date of sowing, the seedlings came to 3-4 leaf stage and were ready for transplanting. The seedlings mats were uprooted by cutting along the boundaries of the mat. The height of the seedlings at the time of transplanting was 15-20 cm. The brief data of seedling at the time of transplanting are given in **Table 2**.

Experimental Procedure

Before transplanting, puddling was done with a tractor drawn cultivator. After puddling, the field was planked in order to level it and left for settlement for one day. On the next day, excess water was drained from the field and 25-35 mm water was maintained to enhance the easy operation of machine and proper fixation of seedlings in the field. At the lower side of the land, the depth of standing water column was more than 50 mm. The transplanter was tested in the laboratory first in order to check the specifications and rectifie minor defects. For carrying out the field trials, seedlings mats were uprooted and transported to the place of operation of the machine. The field and nursery conditions at the time of transplanting are given in **Table 2**. The machine was operated on 16-6-2008 and 17-6-2008 at Sher-e-Kashmir University of Agricultural Sciences and Technology of Kashmir, Srinagar, and at the farmer's field in the village of Nadihal (Baramulla), respectively. The self propelled transplanter was tested in 0.05 and 0.25 ha of land at University and farmer's field, respectively. During field evaluation, no major breakdown was observed except for minor adjustment. The field performance of the transplanter in terms of field capacity, field efficiency, number of hills/m^2, missing hills, buried hills, floating hills, planting depth, standing angle, time consumed in operation and economic analysis were evaluated by using the standard method and compared with hand transplanting. Cost of operation of the transplanter was calculated by assuming a life of 5 years and an annual use of 200 hours. The total transplanting cost was determined by using the standard method. The cost of hand transplanting was determined by using cost of seed bed preparation, nursery raising and management and nursery uprooting and washing. Time consumed in the transplanting operation with respect to such events as planting, turning, adjustment and mat feeding was recorded and shown in **Table 3**.

Table 1 Technical Specifications of Self propelled rice transplanter

Particulars	Specifications
Make & model	Yanji Shakti, 8 rows, 2ZT-238-8, Riding type
Overall dimensions (L × W × H), mm	2,410 × 2,297 × 1,200
Weight, kg	305
Engine model	170F single cylinder, air cooled engine
Engine rated power, kW	2.94 at 2,600 rpm
Adaptable Seedlings	Mat Type
Planting speed, m/sec	0.44 and 0.54
Road travel speed, km/h	8.2
Row to row spacing, mm	238
Distance between hills, mm	120-140, 140-170, 170-200
Planting depth, mm	0-60 (adjustable)
Working width, mm	1,850
Float size (l × w × h), mm	1,850 × 920 × 10

Table 2 Field and nursery conditions at the time of transplanting

Particulars	Value
Date of nursery sowing	19-5-2008
Type of nursery	Mat type
Variety of rice	Shalimar Rice^{-1} (SR^{-1})
Seed rate, kg/sq. ft.	0.50
Age of seedlings, days	28
Plant density, no/cm^2	4- 5
Height of seedling, mm	150-200
Leaf stage	3-4
Root length, mm	25-30
Standing water level, mm	25-35

Results and Discussions

The transplanting operation was satisfactory at both sites by the mechanical transplanter. The average field capacity of the transplanter was 0.0094 ha/h with a field efficiency of 78-80 percent (**Table 3**). The average turning time loss was estimated as 9.0 % due to small field size. Nursery feeding consumed about 3.7-4.3 % of total time of operation. About 4.5 and 3.5 % missing hills were observed at the time of transplanting at SKUAST-K and the farmer's field, respectively, due to various factors like non release of seedlings from fingers due to loose soil and non picking of seedlings by finger. However there was negligible burying of hills while transplanting with transplanter. Some floating hills were observed at the place

Table 3 Field evaluation of self propelled paddy transplanter

Attribute	Self propelled rice transplanter		Hand trans-planting
	SKUAST -K, Srinagar	Farmer's field	
Row spacing, mm	230	230	Random
Planting distance, mm	170	140	Random
Planting depth, mm	46	50	55
No of seedlings/ hill	6.0	5.8	8.8
No. of hills/ m^2	43	40	55
Standing angle of transplanted seedlings (degree)	60-70	60-75	50-75
Missed hills/m^2, %	4.5	3.5	-
Floating hills/m^2, %	2.7	2.1	-
Buried hills/m^2, %	0.5	Nil	-
Area uncovered, %	7.8	8.1	Nil
Operational speed, m/sec	0.19	0.20	-
Fuel consumption, l/h	1.5	1.5	-
Field capacity, ha/h	0.093	0.095	0.0042
Field efficiency, %	78.0	80.0	92.5
Percent distribution of operating time			
a) Transplanting time	81.2	83.9	
b) Total time loss	18.8	16.1	-
i Turning loss	9.1	8.9	
ii Mat feeding	3.7	4.3	
iii Adjustment/ repair	6.0	2.9	
Labour requirement for transplanting, man^{-h}/ha	32.28	31.59	238.1
Effective tillers/hill	-	14.1	13.0
Yield of grain, kg/ha	-	3,580	3,376

Table 4 Cost of transplanting by the transplanter and manual transplanting

Attribute	Labour requirement Man^{-h}/ha	Cost, Rs/ha
Self propelled transplanter	-	
i. Type of seedling required		Mat Type
ii. Cost of operation of transplanter		3,706.0
iii. Cost of nursery raising	32	
a) Seed bed preparation	-	400.0
b) cost of polythene	24	200.0
c) nursery raising & management	24	300.0
d) Mat cutting & feeding	-	300.0
Total cost of transplanting by transplanter		4,906.0
Hand Transplanting		
i. Type of seedling reqd.	-	Root washed
ii. Cost of manual transplanting	238.1	2,381.0
iii. Cost of nursery raising		
a) Seed bed preparation	32	400.0
b) Nursery raising & management	8	100.0
c) nursery uprooting and washing	40	500.0
Total cost of manual transplanting	364	3,381.0

Assumed: labour cost: Rs. 100/day Operator cost: Rs. 200/ day

where standing water level was more than 45 mm. The transplanter could successfully transplant paddy seedlings at 238 mm row to row spacing and 140-170 mm plant to plant with 5-6 plants/ hill. The average number of hills per m^2 was 40 and 43 in the University field and farmer's field, respectively. Whereas, it was 55 for the manually transplanted field. The hills fixed by transplanter stood at an angle of 60 to 75 degrees, which became straight after 10 days from transplanting. The seedlings were well fixed in the ground and eight rows of paddy were transplanted altogether in one pass through the field so that the average area covered per unit time was quite high as compared to hand transplanting.

The economic analysis showed that the cost of operation of the transplanter was Rs. 348.37 per hour by considering 200 hours of annual use. Taking into account the average field capacity of the transplanter, the cost of transplanting of one hectare area was calculated as Rs. 4,906 (**Table 4**). The above estimated costs involved such as nursery raising, nursery management, uprooting, transportation, feeding, and operation cost of the transplanter. The cost of hand transplanting was estimated as Rs 3,381 per hectare with labour requirement of 364 man^{-h}/ha. The cost of transplantation by the transplanter was slightly higher as compared to hand transplanting due to high initial cost and less working hours of the machine. The effective tillers were observed more in mechanical transplanting (14.1 tillers/hill) as compared to hand transplanting (13 tillers/hill). This might have been due to maintenance of uniform and optimum spacing between row to row and plant to plant. Also, during hand transplanting, the root washed seedlings were shocked before establishment in the puddled soil while in mat seedling, the roots were carrying some soil within the root-net and seedlings being mechanically detached in clusters from the mat had more uniform placement in the puddle soil. The average yield of the crop was to be 3,580 kg/ha for mechanical transplanting compared to that of 3,376 kg/ha for hand transplanting. This might have been due to better plant stand and tillering. The yield could be increased further with the self propelled transplanter if untransplanted area left at head land (8 %) was reduced.

Conclusion

The use of the 8-row, self propelled paddy transplanter in Kashmir valley revealed that:

1. The operation of the 8- row manual paddy transplanter was satisfactory in the field and its field capacity was 0.093 and 0.095 ha/h with a field efficiency of 78 and 80 percent for the University's field and farmer's field, respectively.
2. To check floating hills, fairly plain land was required to maintain uniform depth of standing water.
3. To reduce missing hills, development of hard pan in the field and application of lubricants on the tray was necessary.
4. Performance of the transplanter in terms of seedlings placement depended solely on the quality of the nursery. The root mat seedlings should be a specific density and height.
5. The cost of transplanting by the transplanter was a little higher but the machine could be useful during shortage of labour and to maintain timeliness of operation for large land holdings.
6. There is a good scope of mechanization of paddy transplanting in valley lands through the self propelled paddy transplanter for augmenting and sustaining paddy production.

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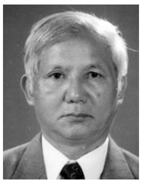
Evaluation of Agricultural Mechanization Level in Agricultural Production Areas

by



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Summary

Evaluation of Agricultural Mechanization Level in Production Areas

Fuzzy methodology with multi-hierarchy evaluation being applied in this paper is a new methodology that helps to analyze the rate of investment of equipment, efficiency in application of agricultural machinery and engineering for developing agricultural mechanization in every region.

The provinces of Mekong River Delta, South East and Red River Delta are those that level of agricultural mechanization, operation and management standard, and investment rate are favorably noticeable. Results from the study based on a system of relevantly complete criteria covering pre- and post-harvest operations could provide more information for a comprehensive overview that helps relevant authorities to be able to introduce solutions appropriate to the practical condition.

Materials and Methods

Agricultural mechanization (AM) plays a very important role in the area of industrialization and modernization. The implementation of agricultural-rural mechanization and electrification is also contributing to the transition of economic structure in agriculture. The 10th Congress of the Communist Party instructed "much attention should be paid to agricultural and rural industrialization and modernization towards large production, linking processing industry to the market; to implement mechanization-electrification and irrigation in order to increase production and quality suitably to the characteristics of individual region and locality".

At the 5th Central Conference (9th term, April 2002), the Central Committee of the Party emphasized: "Develop industry of machinery, equipment and implement manufacture for agriculture and rural development. Put in high priority the modernization of production areas, and invest on research, manufacture, and improve different types of machines and equipment conformably to practical conditions of Vietnam. Incentive policies need to be introduced for application of technical innovations, automation technology and new material technology, with the purpose to increase quality and reduce cost price of industrial products. There should also be policies to support farmers and production units who can afford to buy machines and equipment".

World scientists rank agricultural mechanization 7th in a total of 20 technical advances contributing to strengthening society improvement. In the past 100 years, agricultural mechanization, in each stage, was widely applied to agricultural production, leading to a fundamental change in agricultural production modality, increase of labor productivity on a large scale and the possibility of ensuring the development of agriculture and world food security. Facts have demonstrated that agricultural mechanization has been playing a very important and essential role in development of human society and occupying a very important part in progress of agricultural development and modernization. Therefore, at the international level, the scale and effectiveness of agricultural mechanization are taken as criteria to evaluate the level of agricultural industrialization and modernization.

Lessons identified in the process of survey and summing up experiences in agricultural mechanization (AM) development through pre-in-post harvest stages show that it is necessary to analyse and evaluate the mechanization level in every region. Based upon the analysis results, relevant mechanisms and policies will be recommended to support related stakeholders of the economy to invest and use agricultural machinery. This article applies "Fuzzy Set" methodology—a new method with multi-hierarchical evaluation—that helps to analyze the rate of machinery and engineering investment and application in the country for the recent years.

Define a System of Evaluation Criteria on Level of Agricultueal Mechanization (Am) in Vietnam

Establish the System:

For evaluating level of AM in Vietnam, the following aspects have been initially considered by the experts of the sector:

- Application rate of mechanized operations
- Management standard in AM
- Investment rate of equipment for implementing mechanized operations

The following table summarises the system of AM criteria broken-down from the above aspects:

Table 1 Classify systems of AM criteria

Content	Notation	Unit of measure
Application rate of mechanized operations		
Rate of agricultural land prepared by machine	x_1	%
Rate of annual crop land prepared by machine	x_2	
Rate of paddy land prepared by machine	x_3	
Rate of paddy land where rice is sowed and transplanted by machine	x_4	
Rate of agricultural land where crops care are done by machine	x_5	
Areas where operations of cutting, harvesting, and threshing is mechanized	x_6	
Areas irrigated by machine	x_7	
Management standard in AM area		
Investment rate in agricultural engineering from non-state owned enterprises	x_8	%
Technical management standard of the labor	x_9	mark from $0 \div 1$
Evaluation of staff management standards of AM sector	x_{10}	
Average land area per farm household	x_{11}	ha/household
Investment rate of equipment for AM		
Investment rate of agricultural motive power in 100 ha of agricultural land	x_{12}	kW/ha
Stationary motive power per 100 households	x_{13}	piece/100 households
Investment rate of agricultural motive power in the farms	x_{14}	
Processing machinery for livestock feed	x_{15}	
Processing machinery for agro-forestry-aquatic products	x_{16}	
Machinery used in aquatic catching operations	x_{17}	
Machinery used for transportation in agriculture	x_{18}	
Water pump for agriculture-forestry-aquaculture	x_{19}	

Methodology to Define Criteria:

Explicit quantitative number was summarized from general surveys nation-wide on rural areas, agriculture and fishery, especially in two recent times (2001 and 2006). Consultation method from experts involved in the field of AM and sector management in localities has been applied for qualitative criteria marking and ranking: the best: 1 point; the worst: 0 point, then getting the average.

In order to give points comprehensively, on level of AM in different areas, it was necessary to take into account their local economic and social characteristics and, at the same time, the feasibility of criteria. An optimal set of criteria was considered as standard for comparison. In this case, the biggest value among the criteria has been selected as the optimal value (**Table 2**).

Research Targets:

Areas of agricultural production in Vietnam include 9 ecological regions and cities as follows:

- Two big cities: Hanoi, Ho Chi Minh City
- Ecological regions (as division from the Government): Red River Delta (RRD), North-East; North-West; North Central Coast; South Central Coast; Central Highlands; South East; Mekong River Delta (MRD). **Table 2** summarizes value of the criteria system in production areas.

Evaluate Agricultural Mechannization Level

Fuzzy Correlation of Systematical Elements:

Define criteria set:

X = - set of production areas

• The initial value of unit criteria of production area number i:

X= (x_{i1}, x_{i2}, x_{i3} ... x_{in})

Xij - expresses value of criterium number i of area number j.

Matrix Structure for B:

$$B = \begin{vmatrix} x_{11} & x_{12} & \dots & x_{1n} \\ x_{21} & x_{22} & \dots & x_{2m} \\ \dots & & & \\ x_{m1} & x_{m2} & \dots & x_{mn} \end{vmatrix}$$

The calculated values of criteria reflecting AM development in 9 production areas are presented in **Table 2** as mentioned-above.

Criteria Set Up:

Evaluation criteria usually are of different dimensions, thus, it is impossible to evaluate the criteria based upon their values. Appropriate processing method needs to be applied to convert the criteria data to be dimensionless.

Sets of data are made "fuzzy" by the method of the biggest value and defining parameters are set by dependent function of fuzzy set.

With n as the number of areas, and m as the number of criteria, the matrix C with data from fuzzy set is presented as following:

$$C = \begin{vmatrix} c_{11} & c_{2} & \dots & c_{1n} \\ c_{21} & c_{22} & \dots & c_{2n} \\ \dots & & & \\ c_{n1} & c_{n2} & \dots & c_{nm} \end{vmatrix}$$

Evaluation from Experts' Consultation on Importance of Every Criterium about Agricultural Mechanization in Production Areas

According to results from expert consultation, vector $Di = (i + 1 \div 3)$ is acquired by method of marking.

Application Rate of Mechanized Operations:

$D_1 = (0.238;\ 0.119;\ 0.190;\ 0.095;\ 0.166;\ 0.119;\ 0.071)$

Evaluation from Experts on Management Standard:

$D_2 = (0.20;\ 0.32;\ 0.12;\ 0.36)$

Evaluation On Rate Of Investment:

$D_3 = (0.30;\ 0.081;\ 0.110;\ 0.061;\ 0.150;\ 0.09;\ 0.115;\ 0.093)$

The Level of Dependence is Acquired as Follows: **Table 3**.

Calculation Results

To correlate the importance level as identified by experts and transition results from matrix C, the conspectus of every area can be identified by the following formula:

$E_i = D_i.C_i$

$(i = 1 \div 3)$

And the derived results are presented below:

$E_1 = (1.155553;\ 1.155351;\ 0.556105;\ 0.310273;\ 0.749198;\ 0.802711;\ 0.778725;\ 1.283749;\ 2.190336)$

$E_2 = (1.116633;\ 0.881996;\ 0.912541;\ 0.689205;\ 1.025354;\ 1.025354;\ 1.024001;\ 1.141016;\ 1.18390)$

Table 2 Summaries of criteria system value in nine production areas in Vietnam.

Notation-on	Hanoi, HCM Cities	Production area								
		RRD	North East	North West	North Central Coast	South Central Coast	Central High-lands	South East	MRD	Biggest value
Application rate of mechanized operations										
x_1	0.65	0.70	0.25	0.1	0.4	0.55	0.3	0.65	0.8	1
x_2	0.65	0.80	0.30	0.15	0.5	0.4	0.48	0.72	0.9	1
x_3	0.8	0.9	0.24	0.18	0.6	0.62	0.52	0.8	0.98	1
x_4	0	0	0	0	0	0	0	0	0.3	1
x_5	0.28	0.15	0.2	0.1	0.1	0.1	0.3	0.5	0.25	1
x_6	0.8	0.75	0.4	0.3	0.55	0.6	0.6	0.7	0.8	1
x_7	0.65	0.8	0.5	0.3	0.6	0.5	0.2	0.4	0.8	1
Management standard										
x_8	0.8	0.8	0.4	0.3	0.6	0.6	0.7	0.7	0.8	1
x_9	0.8	0.7	0.6	0.3	0.7	0.7	0.5	0.7	0.8	1
x_{10}	0.8	0.8	0.7	0.3	0.7	0.7	0.6	0.7	0.8	1
x_{11}	0.58	0.35	0.65	0.7	0.65	0.65	0.8	0.8	0.7	1
Rate of investment of equipment										
x_{12}	45.53	78.6	23.6	24	43.8	34.04	97.1	34.2	37.3	97.1
x_{13}	4.7	2.7	2.6	2	2.82	3.65	4.28	3.98	12.35	12.35
x_{14}	1	5.3	6	3	4	3.7	13	11	13	13
x_{15}	0.13	0.2	0.76	0.13	0.38	0.17	0.23	0.1	0.14	0.76
x_{16}	2.12	3.5	6.9	5.74	3	1.9	1.6	0.8	0.75	6.9
x_{17}	0	0.12	0.16	0	0.45	0.3	0	0.37	3.6	3.6
x_{18}	6.5	0.5	0.45	0.2	0.4	0.45	0.35	0.6	0.9	6.5
x_{19}	43.34	13.54	18.76	0.62	4.86	24	17.44	17	33.36	43.34

Source: General Department of Statistics, 2007. Results of surveys and summaries of the authors, 2008.

E_3 = (0.89244; 1.065868; 1.0272; 0.741887; 0.7825505; 0.745908; 1.212543; 0.83222; 1.699427)

V. Remarks:

1. On the application rate of mechanized operations from soil preparation to harvest; the highest levels are found in the provinces of Mekong River Delta, South East, Hanoi, Ho Chi Minh City, and Red River Delta, respectively, by the way farmers themselves invest (or hire machines) to work for their own households and then do service for others.
2. On the management and organization ability in AM sector (related to the process of AM investment from different economic stakeholders), the highest levels are seen at MRD and the provinces in the South, then the two biggest cities.
3. On the rate of agricultural engineering-machinery investment in the operations of production, post-harvest processing and irrigation: MRD, Central Highlands, and RRD are those of highest rates of investment.
4. MRD has the highest level of equipment investment and the standard of management is good, leading to high effectiveness and highest level of AM application. The two cities of Hanoi and Ho Chi Minh City and the South East region have low rates of equipment investment, but high management standards. Therefore, the level of AM is still higher than those from other areas.

Conclusion

1. Agricultural mechanization in production areas in Vietnam has been gradually contributing to change of productive forces, effectively using labour, exploiting land potential, increasing quality of agro-products and, thus, contributing to the development of rural economy and the increase of income for farmers.
2. The selected criteria used for evaluating AM level as mentioned-above in Vietnam (taking into account of characteristics of every economic area) are quite plentiful and also are fuzzy information; thus, it is right to use the fuzzy set methodology in the analysis and comparison.
3. The above 19 criteria are elements that affect the processes of investment and application of agricultural engineering for developing AM. They are general and cover enough of all processes of pre and post harvest operations, investment and managment. They are also compared and evaluated objectively in the analysis.
4. Through model application and summation of expert ideas, it can be seen that the level of AM, management standard, and investment rate are noticeable in the provinces of MRD, South East and RRD, respectively. The rate of investment and management standard are factors being decisive to the result of AM development in production areas.

Although this study is based on acquired data for analysis using fuzzy methodology, some important elements have not been covered in the criteria system like those of field plot gathering and accumulation. The existing small field plot scale

Table 3 The Level of Dependence is Acquired as Follows:

C_1 =	1.329545	1.431818	0.511364	0.204545	0.818182	1.125	0.613636	1.329545	1.636364
	1.193878	1.469388	0.55102	0.27551	0.918367	0.734694	0.881633	1.322449	1.653061
	1.276596	1.43617	0.382979	0.287234	0.957447	0.989362	0.829787	1.276596	1.56383
	0	0	0	0	0	0	0	0	9
	1.272727	0.681818	0.909091	0.454545	0.454545	0.454545	1.363636	2.272727	1.136364
	1.309091	1.227273	0.654545	0.490909	0.9	0.981818	0.981818	1.145455	1.309091
	1.231579	1.515789	0.947368	0.568421	1.136842	0.947368	0.378947	0.757895	1.515789
C_2 =	1.263158	1.263158	0.631579	0.473684	0.947368	0.947368	1.105263	1.105263	1.263158
	1.241379	1.086207	0.931034	0.465517	1.086207	1.086207	0.775862	1.086207	1.241379
	1.180328	1.180328	1.032787	0.442623	1.032787	1.032787	0.885246	1.032787	1.180328
	0.903114	0.389273	1.012111	1.089965	1.012111	1.012111	1.245675	1.245675	1.089965
C_3 =	0.975499	1.692587	0.508207	0.516821	0.943198	0.733024	2.09097	0.736469	0.803225
	0.793919	0.456081	1.621622	1.554054	0.476351	0.616554	0.722973	0.672297	2.086149
	0.139686	0.73056	1.410833	0.419059	0.614621	0.51684	1.815924	1.536551	1.815924
	0.422383	0.649819	2.469314	2.469314	0.584838	0.552347	0.747292	0.649819	0.454874
	0.731876	1.208285	2.382048	1.981588	1.008055	0.638665	0.548907	0.248562	0.252014
	0	0.233533	0.287425	0	0.808383	0.538922	0	0.664671	6.467066
	1.3	1.0	0.9	0.4	0.8	0.9	0.7	1.2	1.8
	2.2557	0.704	0.476	0.032	0.2529	1.249	0.907	0.884	1.736

remains an issue that limits the government to apply strong measures to promote the development of AM.

General results from this study can provide more information to local authorities in establishing relevant policies and mechanisms to speed up the process of agricultural mechanization in the localities.

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NEWS

DIRECTORY/REPERTORIO 2011-2012

A Directory of Comacomp 2011/2012

COMACOMP (Manufacturers of Components for Agricultural and Earth-Moving Machinery and Garden Equipment) groups the Italian makers of components for agricultural, earth-moving and garden machinery and equipment as part of UNACOMA (Italian Farm Machinery Manufacturers Association). It covers more than 6,000 people from 80 companies.

They are arranged into 6 main sectors; Mechanical Components, Hydraulic Components, Miscellaneous Components, Components for Irrigation, Components for Spraying, and Electrical- Electric Components. Charts are tacked up on the pages of each sector, and you can see which product or a part is made in which company in one glance. The directory is written in English and Italian.

The numbers of companies and the variation of parts listed on the directory are as mentioned below.

Mechanical Components: 17 types of parts and 11 companies
Hydraulic Components: 21 types of parts and 23 companies
Miscellaneous Components: 88 types of parts and 51 companies
Components for Irrigation: 17 types of parts and 11 companies
Components for Spraying: 21 types of parts and 23 companies
Electrical- Electric Components: 23 types of parts and 21 companies

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The Cost of Fluidized-Bed Drying Using a Buffered Salt Layer

by
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Abstract

A dryer was fabricated and investigated that was based on the fluidized-bed method without using the conventional air distribution shaking screen but with a fixed screen that had a buffered dry salt layer as an alternative to promote the heat and humid exchange process effectively. The finished prototype had a capacity of 200 kg/hour and could be used to dry grain material for food processing and chemical industries. All the experiments were randomly conducted and some critical parameters of the machine were determined by a multi-factor experimental plan. The experimental results helped to establish an equation that represented the dependence of humidity versus the active area of the screen, the height of barrier plate, and the temperature of the drying agent for the drying of pure salt.

$W_b = 5.05612 - 0.213657\varphi - 8.673737 \times 10^{-3}\ h_c - 0.0292034_t + 8.62965 \times 10^{-3}\ \varphi^2 + 1.48033 \times 10^{-5}\ h_c^2 + 9.694 \times 10^{-5}\ t^2$ .... (1)

The experimental results also defined an equation for the dependence of the cost of heating energy versus the active area of the screen, the height of barrier plate, and the temperature of the drying agent for the drying of pure salt.

$B_r = 3905.5 - 82.5117\varphi - 2.45799h_c - 34.5311t + 0.177083 \times \varphi \times h_c + 2.51969\varphi^2 + 7.02027 \times 10^{-3}\ h_c^2 + 0.1181123t^2$ .... (2)

Through the regression equation, some optimum parameters were determined as follows[o]:

- For the requirement with a humidity of material of 0.3 %, the heat cost was 882 kJ/kg, equivalent to the active area of the screen of 12.1 %, a height of the barrier plate of 249 mm, and a drying temperature of 142 ºC.
- For the requirement with a humidity of material of 0.4 %, the heat cost was 846 kJ/kg, equivalent to the active area of the screen of 11.79 %, a height of the barrier plate of 203 mm, and a drying temperature of 139 ºC.

Introduction

Salt is ultimately necessary for our life, and an indispensable substance for human daily ration. It is a part of food preserving and processing. Besides that, it is also used as input material for some chemical industries. The ratio of salt for chemical synthesize in the world is 60 %, 30 % for food processing and 10 % for other usage. Salt processing is one of the oldest industries and is wide spread throughout the world. With the participation of 120 countries, the world productivity reaches 240 millions tons per year, equivalent to 35 kg/person/year. Based on the demand of the market and the requirements of some industries, pure salt has to be dried with high quality, which satisfies the quality standard of food processing companies as well as the criteria of consumers. As a result, it is necessary to study and fabricate an appropriate dryer model that satisfies all the above requirements. The requirements for the qualities of the dried salt for this experiment are a white color and bright, size and shape that satisfy the required standard and a temperature of the product after removal from the dryer that is low to prevent curdling.

On the economic aspects, the designed machine should have a stable quality, the harvested finished product high, the heating cost low. The schematic of the design is illustrated in **Fig. 1**.

Content and Research Methodology

The research was designed to study the fundamental theories for the calculation and design of a fixed screen fluidized-bed dryer with a buffered salt layer and a capacity 200 kg/h in the following steps:

- Fabricate the machine for the calculated and designed results.
- Test the fabricated machine to affirm the working ability.
- Investigate and choose appropriate parameters for the experiment.
- Study the effect of some parameters of the fluidized-bed dryer with buffered salt layer on the

drying heat cost and the humidity of salt after drying. The expected humidity of the salt is Wb = 0.3-0.4 %.

Statistical methods in experimental data processing were used with StatGraphic 7.0 software to investigate the effect of some factors on the technical criteria of the dryer, to find the optimum parameters for the designed machine.

Results and Discussion

Theoretical Results

The key points in calculating the fluidized-bed drying system were to specify the velocity of the drying agent, the velocity for the fluidizing mode, ω_{th1}, the velocity for the optimum drying, ω_k, the thickness of the salt, the dimensions of the drying chamber and the total resistance of the system. The minimum velocity was the velocity of the drying agent which emerged from the fluidizing in the grain layer, ω_{th1} = 0.3 m/s. The optimum working velocity was ω_k, = 1.1 − 2.3 m/s. The working velocity of the drying agent through the fluidizing layer was ω_k, = 1.5 m/s. This value was equivalent to the sponginess of the material, ε = 0.65. The height of the fluidizing bed was calculated as:

$$h = h_o (1 - \varepsilon_o / 1 - \varepsilon) = 0.171\ m$$

The flow rate of the drying agent was V_k = 0.4 m^3/s and the total resistance of the system was ΔP = 400 mm of water.

Experimental Investigation Results

The input variables that can affect the output variables and can be controlled were mentioned in the fundamental theory and experimental investigation. They are: (X_1) the active area of the screen, φ = 9-15 %; (X_2) the height of the barrier plate between the drying chamber and cooling chamber related to the height of the fluidizing grain layer, hc = 200-300 mm; and (X_3) the temperature of the drying agent, t = 120-160 ºC. These factors have a large affect on the pure salt drying process, the quality of the salt grain and the drying heat cost. Using the experimental plan, several factors were investigated simultaneously on the dried salt quality and drying heat cost.

Output Variables:

- The humidity of salt: in encoding form, y_1, and in real form, W_b %. This was a characteristic parameter for the research. It received the effect from most of input factors and disturbance, and was measured by the screen analyzing method.
- Drying heat cost: in encoding form, y_2, and in real form, B_r kJ/kg. This was, also, a characteristic parameter for the research that reflected the economic and technical index for the drying process. It was counted by measuring the consumed volume of gas over the dried salt quantity.

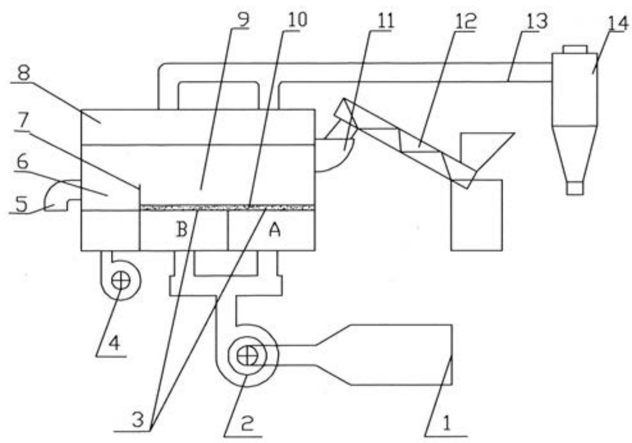
The processed experimental data showed that the linear function was not appropriate for describing the parametric relation between dried salt quality and heat cost with active area of screen, height of barrier plate and drying temperature. This meant that the linear model cannot be used to reflect the relationship. Therefore, it was necessary to increase the experimental method to a nonlinear, second order function and the experimental field was extended to ± α.

Table 1 illustrates parameters for the experimental nonlinear, second order polynomial function.

Experimental Results and Processed Experimental Data

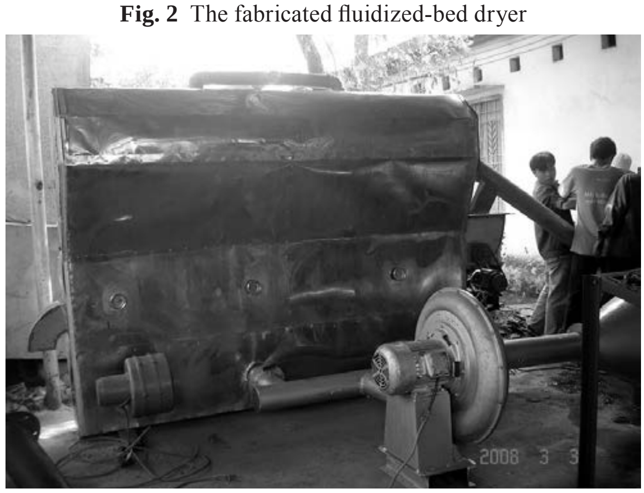
The experiment proceeded according to the established experimental matrix and the collected data, after the processing was en-

Fig. 1 The schematic of the fluidized-bed dryer.



1- Furnace; 2- Fan; 3- Air distributing Screen.; 4- Cooling fan; 5- Outlet of Material; 6- Cooling chamber; 7- Barrier; 8- Separated Chamber; 9- Drying chamber; 10- Buffered salt layer; 11- Supply gate; 12- Vis conveyer; 13- Exhausted air pipe; 14- Dust collection cyclone.

Fig. 2 The fabricated fluidized-bed dryer



tered into the matrix, to calculate the required parameters of the model. The resulting regression equation for the humidity of salt, $y_1(W_b)$, is shown below:

In the real form, the humidity function, W_b, relates to the active area screen, φ, the height of barrier plate, h_c, and drying temperature, t.

$$W_b = 5.05612 - 0.213657\varphi - 8.67373 \times 10^{-3}h_c - 0.0292034t + 8.62965 \times 10^{-3}\varphi^2 + 1.48033 \times 10^{-5}h_c^2 + 9.694 \times 10^{-5}t^2$$

When the above equation was placed in the parameteric form, y1, three dimentional figures were plotted to show the effect of each couple of factors on the humidity of material. The plots are illustrated in **Fig. 3**: $y_1 - x_1 - x_2$; $y_1 - x_1 - x_3$; $y_1 - x_2 - x_3$. These figures were plotted when the other factor was kept in its basic level.

The regression equation for the drying heat cost y_2 (B_r) is shown below.

Based on the real form of y_2 to plot figures showing the effect of each couple of factors on the heat cost, **Fig. 4** includes 3 plots: $y_2 - x_1 - x_2$; $y_2 - x_1 - x_3$ and $y_2 - x_2 - x_3$. These figures are plotted when the other factor is kept in its basic level.

Results from the Optimum Processing:

- For the requirement that the humidity of material is 0.3 %, the heat cost is 882 kJ/kg, equivalent to the active area of the screen of 12.1 %, the height of the barrier plate of 249 mm, and the drying temperature of 142 ºC.
- For the requirement that the humidity of material is 0.4 %, the heat cost is 846 kJ/kg, equivalent to the active area of the screen of 11.79 %, the height of the barrier plate of 203 mm, and the drying temperature of 139 ºC.

Discussion

Thanks to the buffered dried salt layer being heated in advance, the supplied material was not sticky with each other when exposed to high temperature, the heat and humidity exchange process not only conducted between drying agent and dried material but also between each layer of dried material with those results in low humidity of dried material in the end of the process.

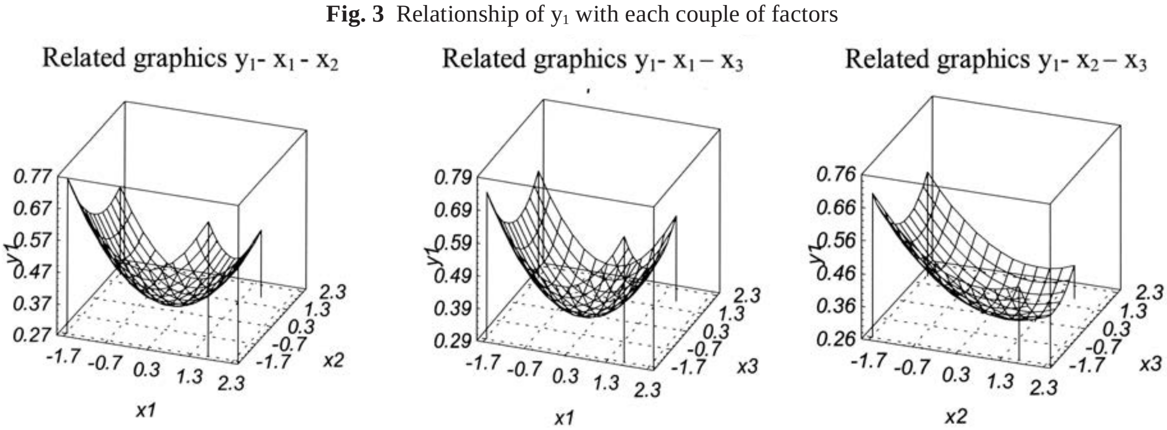
Through the investigation in theory and experiment, the temperature of the drying agent after the drying chamber was still rather high was normal for the common fluidized-bed dryers or pneumatic dryers. However, in this drying agent, besides the exchanged humidity from material, there was still salt dust. It was required that the heated, recirculated part of the exhaust air should be considered for the fabricated material as well as the humidity and heat exchange of the drying agent.

The humidity of dried material is one of the basic criteria to evaluate the quality of dried products due to the consuming and preserving procedure after drying, this quota depends on several factors: drying temperature, the height of the barrier plate, active area of the screen. The drying temperature has the largest effect on the humidity of dried material. However, if the temperature of the drying agent is too high, the temperature of the exhausted air is high also. This shifts the drying heat cost and the temperature of the dried material that is also high at the output. It cost more energy to make it cool and sometimes makes the dried material easy to be curdled.

Table 1 Variant level and range of the input factors

Level \ Factor	X_1 (φ) Active area of screen (%)	X_2 (h_c) Height of barrier plate (mm)	X_3 (t) Temperature (ºC)
−1.68	6,954	166	106
−1	9	200	120
0	12	250	140
+1	15	300	160
+1.68	17,046	334	174
Variant range	3	50	20

Fig. 3 Relationship of y_1 with each couple of factors



The active cross-section of screen for the air distribution affects the fluidizing process of the material. If this cross-section is small, the free area of screen shall be small leading to the increased velocity of the drying agent through the screen. In the other aspect, a larger active cross-section of the screen leads to a slower velocity for the drying agent. Therefore, the active cross-section of the screen has a greater effect on the fluidizing process as well as the heat and humidity exchange process between the dried material and the drying agent.

On the aspects of the height of the barrier plate, its dimensions also affect the humidity of the dried material. If it is low the material will be easy to flow across. In this case, the humidity of material is high and the heat cost is low. Inversely, if the barrier plate is higher, flow across the material is more difficult and humidity is low and the heat cost is high.

Conclusions

The drying of pure salt using the fluidized-bed dryer with buffered dry salt layer prevents the sticky of the high humidity drying material encountered with a high temperature drying agent, promoting the heat and humidity exchange among the grains of drying material.

The machine has a simple design, is easy to fabricate and its components do not require a high precision. This is one of the advantages for the application in small manufacturing firms.

Using the experimental plan method, the mathematical equation was defined that illustrated the dependence of the humidity of the drying material and heat cost on the active cross-section of screen , height of barrier plate and temperature of drying agent.

The optimum parameters for the designed machine in the multi objective function were:

- For the requirement with a humidity of material of 0.3 %, the heat cost was 882 kJ/kg, equivalent to the active area of the screen of 12.1 %, the height of the barrier plate of 249 mm, and the drying temperature of 142 ºC.
- For the requirement with a humidity of material of 0.4 %, the heat cost was 846 kJ/kg, equivalent to the active area of the screen of 11.79 %, the height of the barrier plate of 203 mm, and the drying temperature of 139 ºC.

A finished prototype for the fluidized-bed dryer with a capacity 200 kg/hour was designed and fabricated. It could be used to dry grained materials for food processing and chemical industries.

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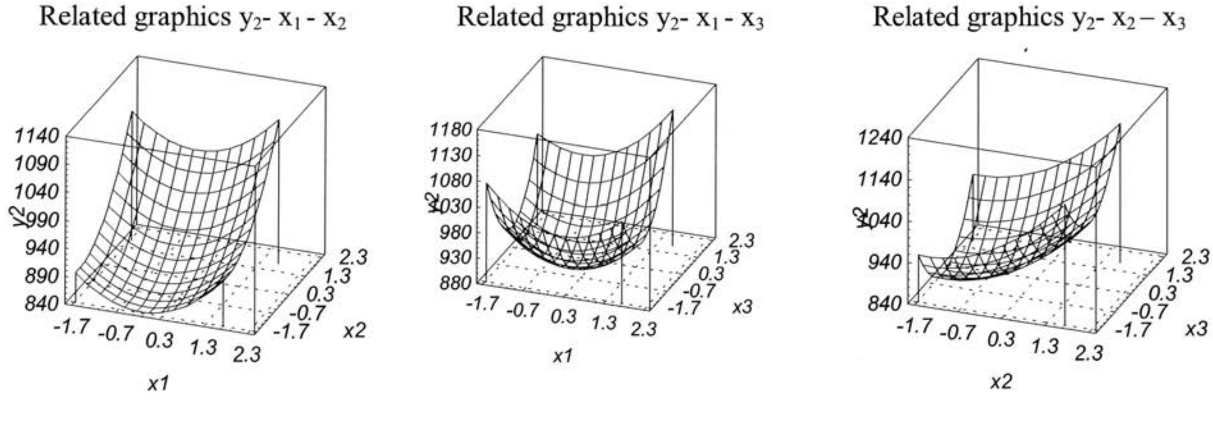
Fig. 4 Relationship of y_2 with each couple of factors



Development and Evaluation of Tractor Operated Rotary Spading Machine

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Abstract

The manual spading operation in rice fallow cotton is a highly labour intensive operation. The success of the crop depends on this crucial operation. The labour availability for such work is very limited and manual operation is costly. A tractor operated rotary spading machine was developed that provided an exciting alternative to tillage under conditions where other tillage techniques cannot be adopted. The unit consisted of a main frame, gearbox, crank shaft assembly, digger arm and shovel assembly and depth control wheel. The shovel with digging arms was thrown into the soil and, with the help of a 4-bar linkage mechanism, broke and loosened topsoil with crop residue and mixed the surface applications well so that air and moisture sped the decomposition of vegetation. The performance of the rotary spading machine was evaluated and compared with the existing practice of manual digging with a spade. The furrow bottom without any smearing of compaction was highly beneficial for root growth. It was also observed that as depth was increased the area increased. This clearly showed that as the tillage tool worked deeper, the size of the side failure crescent increased and led to increased area of furrow. The quality of work done by the spading machine in terms of uniformity in depth and degree of soil break up was better than the manual spading. The operation of the spading machine resulted in 38.5 and 96.5 percent saving in cost and time, respectively, when compared to manual spading. Also, it resulted in 15 and 90 percent savings in cost and time, respectively, when compared to manual digging of a manure pit around a coconut tree.

Introduction

Cotton is raised as a rice fallow crop. The requirements of the crop are typically different. The sown crop usually utilizes the residual moisture available at the time of the paddy crop harvest. During the 30th to the 45th day, a second dose of fertilizer is applied. The practice is to dig up the soil manually using a spade and form ridges and furrows. This is done along with the second dose of fertilizer. In the heavy clay soils of the Cauvery delta, proper irrigation is possible only if the field is formed into ridges and furrows. The labour availability for such work is very limited and manual operation is costly.

Based on the need expressed by the agronomists for mechanizing this operation, the development of a rotary spading machine for rice fallow cotton was conducted. The spading machine provided an exciting alternative to tillage under conditions where other tillage techniques cannot be adopted. The machine could also be used extensively in horticulture, green houses and other applications where deep tillage without damage to soil structure was desired. The spading machine alleviated the labour scarcity problem as the implement was powered by a tractor. This research was aimed at development of a rotary spading machine for inter-row tillage in rice fallow cotton.

Review of Literature

Eugene and Canales (1997) developed a spading machine, which used 6 spades driven by a single crankshaft powered by a central gearbox that received its power from the PTO of a 30 hp tractor. The gearbox had several stroking speeds and each crank arm was 60º out of phase so that each spade could penetrate alone and save power. This first machine was 57 inches wide and used 6 spades digging 12 inches deep. Pezzi (2005) evaluated the performance of traditional and alternative implements for deep tillage. Three implements were compared: a plow and two PTO-driven machines (spading machine and rotary chisel) along with two soil depths, 0.30 and 0.40 m, and two forward speeds. Tests were made on a level plot, containing silty-clay soil. Forward speed, wheel slip, fuel consumption and energy at the PTO and draw bar were measured. Cloddiness of the tilled soil was evaluated. The two PTO-powered implements did not show advantages in terms of capacity, but gave better results in fuel consumption and soil pulverization. Economic evaluation showed lower unit costs (17-28 %) for the spading machine.

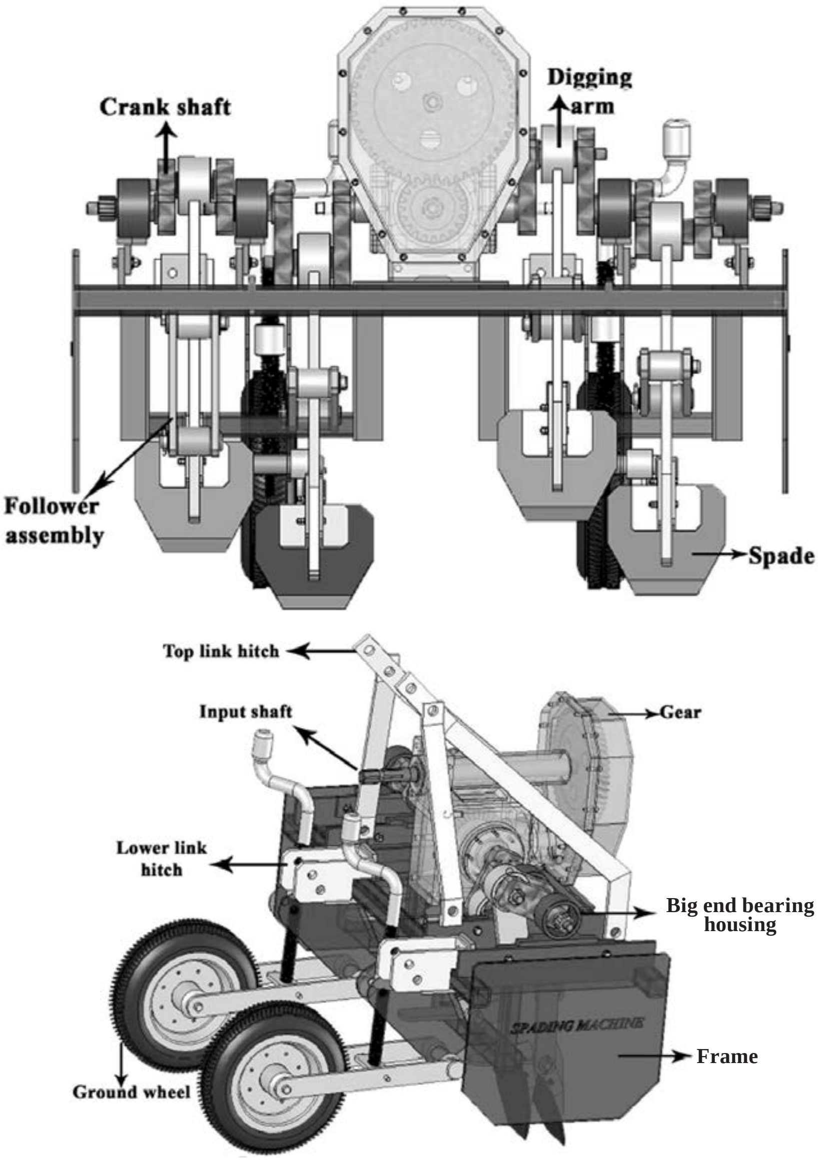
Kepner *et al.* (2000) explained the working principles of different rotary spading machines. A spading machine has a transverse, powered rotor with spades attached to arms mounted on it. The spades dig in from the front, cut soil sections loose, and lift them at the rear. Donafo and Tortella (2006) started to produce a spader for hard and rocky works. Their spading machines were lightweight and reduced dimensions made the 001 Series spading machine very useful for spading work in greenhouses, vineyards and gardening of all kinds (vegetables, flowers, etc.). There was a special linkage for each type of motorized cultivator, which together with this lightweight implement created a perfect balance between machines. The penetration of the small blades into the soil was adjustable and, therefore, allowed the equipment to be suitable to every kind of situation. Coleman (1995) reported that there were two types of spaders: rotary and reciprocating. Rotary spaders had spade shaped blades on arms much longer than that of a rototiller. These moved more slowly through the soil and were less harmful than the high speed tines of a conventional tiller. Reciprocating soil spaders used a mechanical drive to simulate the action of spading with a shovel. The reciprocating soil spaders were gentler to the soils and should be considered the tillage tool of choice on most projects.

Methods and Materials

Tractor Operated Rotary Spading Machine

The tractor operated rotary spader consisted of a main frame, gear box, crank shaft assembly, digger arm and shovel assembly and depth control wheel (**Fig. 1**). The main frame was a rectangular frame 1,028 × 480 mm. It had two square bars made

Fig. 1 Isometric view of rotary spading machine



of 50 × 50 mm mild steel hollow square sections connected by plates 10 mm thick on either side. The frame had a standard three point hitch system. All the other functional components were fitted with the mainframe. The gear box was a bevel gear box to change the direction by 90 deg and also provided for mounting of the crankshaft on both sides. The common PTO speed of tractors is 540 rpm. Unlike the standard rotavator that had three blades arranged in each line of cut, the spader had only one spade and, hence, the speed of the spader was higher to ensure proper bite length. The major difficulty of designing a spader for Indian makes of tractors was that the minimum forward speed of these tractors was above 2.5 km/hr and the PTO speed was 540 rpm. This required the crank speed to be maintained at higher levels to ensure proper bite length. These parameters could only be optimized under field conditions. Hence, the gearbox was designed with a 10:31 reduction ratio. The secondary gearbox was a spur gear type for changing the input speed to the primary gearbox. Its speed increase ratio was 1 : 2.5.

The crankshaft assembly was mounted on either side of the primary gearbox and had two throw and two main bearings. The digging arms were mounted to the frame through a 4-bar linkage. The spader had four digging arms each carrying spades of adjustable width (100/150/200 mm). Each arm was mounted on a crank. To enable mounting of ball bearings, the crankshaft was designed as a built-up shaft with crank links consisting of a single crank arm and crank pin forming the basic unit. The adjacent crank links were connected by splines.

The digger arm and shovel assembly had four digging arms with shovel, which were mounted on the crank shaft with bearings (**Fig. 2**). The shovels were mounted at the ends of the arms. The shovels were rectangular and chamfered at the bottom. The height of the shovel was 250 mm. The shovel was made of 6 mm mild steel plate. The width of the shovel was adjustable (100/150/200 mm). Two depth control wheels were provided in the front portion of the main frame for controlling depth of penetration of shovels. A screw rod was provided for adjusting the depth of operation. The power from the PTO shaft was transmitted through cordon shaft to a secondary gear box. The direction of rotation was changed through 90° and fed to the input shaft of the primary gear box. From the primary gearbox output shaft, the power was transmitted to the crank assembly.

Table 1 Specifications of the spading machine

Parameter	Value
Over all dimensions (L × B × H), mm	1,068 × 500 × 960
Weight , kg	360
Main frame type	Box type
Type of primary gear box	Bevel gear type
Tractor coupling	Thee point connection Category II and III
Transmission of drive from PTO	Cardon shaft
Main Transmission	Gear box with helicoidal gearing in constant drive in oil sump
Speed ratio	3 : 1
Secondary gear box	Helical spur Gear type
Speed ratio	1:2.5
Radius of drive crank, mm	100
Number of digging arms	4
Number of shovels	4
Width of shovel, mm	120, 150 and 200
Bite length, m	0.141
Diameter of depth control wheel, mm	300
PTO speed, rpm	540

Table 2 Results of field evaluation of the spading machine

Details	Value		
	Trial I	Trial II	Trial III
Size of the field, m	6 × 88	9 × 88	10 × 110
Actual operation time, min	20	32	38
Time lost in turns, min	4	7	7
Time lost in adjustments, min	1	2	2
Average PTO speed, rpm	150	150	160
Rotational speed of spades, rpm	100	120	130
Theoretical working width, m	1.50	1.50	1.5
Forward speed, kmph	1.2	1.2	1.2
Operating working width, m	1.50	1.5	1.5
Depth of operation, cm	6	6	8
Width of crest, cm	20	18	15
Theoretical field capacity, ha hr^{-1}	0.18	0.18	0.19
Actual field capacity, ha hr^{-1}	0.126	0.116	0.13
Field efficiency	70.0	64.40	70
Number of plants in sample pass	586	575	582
Number of plants damaged during turning at head land	4	5	5
Damage to crops, %	0.68	0.86	0.85
Soil throwing distance, m	0.3	0.45	0.3

With the help of a 4-bar linkage mechanism the digging arms were activated in the desired digging path. The specifications of the spader are furnished in **Table 1.**

The performance of the spading machine was evaluated in rice fallow cotton. **Fig. 3** shows the working of the spading machine in typical cotton crop raised in rice fallow. The unit was also evaluated for spading around coconut trees to dig pits for manuring.

Furrow Geometry

Furrow geometry was measured (**Fig. 4**) using a furrow profile meter (Jesudas 1994).The furrow profile was recorded on the graph sheet using the furrow profile meter and the area of furrow measured. The width, depth and cross sectional areas of the furrow were recorded.

Economics of Spading Machine

The cost of conventional spading practices adopted for rice fallow cotton was evaluated. Conventional practices involve, spading between rows, weeding in the inter row area and earthing up. The spading machine does only the first operation. The cost of inter-row cultivation was compared with the manual method.

The spading machine was also used for digging a pit (40 cm width and 15 cm depth) around the coconut tree for applying manure, fertilizer and farming basins. Out of the four shovels mounted on digging arms, two shovels were removed from one side to work with two shovels only. The side of the unit without a shovel was lifted using the lower link.

Results and Discussion

The results of the field trials in the cotton crop are furnished in **Table 2**.

The damage to the plants due to the spading operation varied from 0.68 to 0.86 and 2 to 4 % of plants were damaged while taking turns at headlands. The actual working width of right and left spade was 150 mm after reducing the width. The width of the spaded furrow was 350 mm and the space left after spading was 60 to 100 mm based on the passes made. The theoretical field capacity was 0.19 ha/hr. When the turnings times at the headlands were considered, the actual field capacity was 0.133 ha/hr.

Economics of Intercultural Operation

The performance of the rotary spading machine was evaluated and compared with the existing practice of manual digging with a spade.

Number of man hours required for spading one ha	240
Wages of men workers, Rs/h	15
Cost of spading operation with men worker, Rs/ha	3,360.00
Cost of operation of tractor, Rs/hr	250
Field capacity, ha/hr	0.121
Cost of operation with spading machine	**2,066.00**
Saving in cost of spading with spading machine when compared to spading with male worker, (%)	38.50
Saving in time of spading with spading machine when compared to spading with male worker, (%)	96.5

The operation of the spading machine resulted in 38.50 to and 96.50 percent saving in cost and time, respectively, when compared to spading with a male worker.

Economics of Digging Circular Trench

The cost comparison for digging a semi circular trench manually by a team of two persons is furnished below.

Time required for digging	15 minutes per pit
Cost of digging operation with men workers	Rs. 12 per pi

Fig. 2 Rotary spading machine

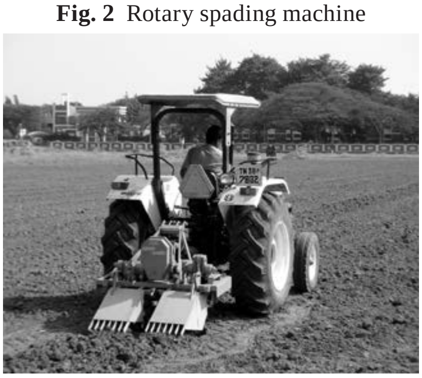
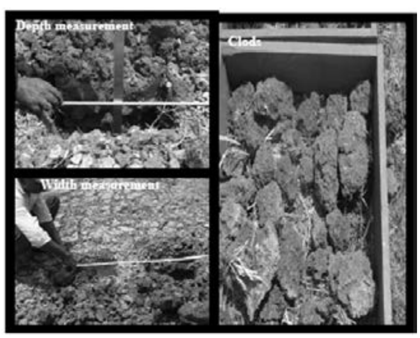


Fig. 3 Spading in rice fallow cotton

Fig. 4 Spading around a coconut tree

Fig. 5 Measurement of furrow profile using furrow profile meter



Digging by Spading Machine:

No. of pits per hour	30
Cost of digging operation with machine	10
Cost of manual digging manuring trench/ha of coconut palm	Rs. 6000
Cost of machine digging manuring trench/ha of coconut palm	Rs. 5000
Percentage saving in time	90 %
Percentage saving in cost	15 %

Furrow Geometry Generated by Spading Machine

The furrow geometry was recorded using a furrow profile meter shown in **Fig. 5.** Recording of furrow profile showed that the furrow bottom was very rough, which indicated tearing of furrow slices from the soil mass. Such furrow bottom without any smearing of compaction was highly beneficial for root growth. Also, as depth was increased the area was increased, which clearly showed that, as the tillage tool works deeper, the size of the side failure crescent increased and this led to increased area of furrow. The quality of work done by the spading machine in terms of uniformity in depth and degree of soil break up was better than the manual spading.

Conclusions

A tractor operated rotary spading machine for rice fallow cotton was developed. The unit consisted of a main frame, gearbox, crank shaft assembly, digger arm, shovel assembly and depth control wheel. The shovel with digging arms were thrown into the soil with the help of a 4-bar linkage mechanism and loosened broken topsoil. Crop residue and surface applications were well mixed so that air and moisture could speed the decomposition of vegetation. The performance of the rotary spading machine was evaluated and compared with the existing practice of manual digging with a spade. The operation of the spading machine resulted in 38.50 and 96.50 percent saving in cost and time, respectively, when compared to manual spading with a male worker for intercultural operations in rice fallow cotton. The cost of the spading machine was Rs.60,000 and could cover one ha per day. The operation of the spading machine resulted in 15 and 90 percent saving in cost and time, respectively, when compared to manual digging of manure pit around a coconut tree. . The quality of work done by the spading machine in terms of uniformity of depth and degree of soil break up was better than the manual spading.

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Transplanting Machine Operation Analysis

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Abstract

Transplanting has been used for most vegetables and ornamental plants to achieve better establishment and early maturing of the crops. Manual transplanting is slow and labour intensive. Semi-automatic transplanters are a low capital alternative that can increase productivity, speed of work and lower costs. In the present paper a comparison between the manual and semi-automatic transplanting was carried out during field work. Leaks, cabbage and peppers were transplanted in parallel rows manually and mechanically. The results showed that the cost for automatic transplanting was 73-86 % lower than manual transplanting, while the operating performance was 32.7-38.5 % higher with the semi-automatic transplanter. It was also shown that the performance of the workers on the transplanter increased with experience gained while working continually for six working days.

Introduction

The increased land area dedicated to vegetable production and the increasing number of vegetable species grown in open fields demand higher mechanization for better efficiency and working rates to achieve a reduction of production costs. Transplanting is recommended for plants with small seeds, as most horticultural and ornamental plants are, as well as for some arable crops. Transplanting saves land as the plants are produced in small area nurseries while another crop is producing in the field. But, mainly it offers an early in the season production. Early production permits a premium price for the products that covers the increased cost of transplanting. Plants that are usually transplanted are tomatoes, peppers, cabbages, leeks, lettuce, as well as arable crops such as tobacco, rice and, to a lesser extend, sugar beets (Nambu *et al.*, 1987). Recently, transplanting of processing tomatoes has been widely used by Greek farmers.

The savings of high cost hybrid seeds as well as better weed control has led to the expansion of transplanting despite higher operational cost. The advantages of transplanting are better performance because of selection of healthy plants to be planted; better utilization of soil nutrients and moisture as the plants are transplanted in exact distance intervals; less competition with weeds; faster growth as they are planted well grown; and, finally, better crop rotation as the growing season is shorter in intensive agriculture (Tsatsarelis, 2000). Earlier production can achieve easier placement and higher prices in the market. The disadvantages of transplanting are related to the demand of increased labour in the short term and increased fatigue by the workers, leading to mistakes in placing the plants at the right distances.

For a successful alleviation of the aforementioned disadvantages, there is a steady development of transplanter machines that either help the worker or they carry out the transplanting operation semi- or fully automatically. For transplanting, there is, currently, equipment that could be classified as:

a) placing the plants in the soil directly by the workers. The machine opens the furrow and carries the workers with the plants.
b) semi-automatic systems, which are systems that only require the worker to feed the plants. The machine then places the plants into the soil at the required distances.
c) fully automatic systems that transplant with soil around the plants without human labour. The later are late developments and very expensive. Their use is limited to large mechanized farms. Automatic machines can transplant 100-140 seedlings/min per row of the machine (Srivastava, *et al.*, 2006). In most smaller farms, where most vegetables are produced, the type used is the semi automatic.

The efficiency of the transplanting machine is related to the tractor speed, which is related to the feeding rate by the worker. The number of plants each worker can feed ranges from 20 to 40 plants/minute (Tsatsarelis, 2000; Srivastava, *et al.*, 2006) and, depending on the plant density, corresponds to a travel velocity of 0.5-1.3 km/h (Srivastava, *et al.*, 2006). The efficiency of the transplanting machine is also related to the experience of the workers that feed the plants into the transplanting mechanism. Culpin (1975) stated that observations in USA and UK indicated that a person working on a transplanter can accurately feed 50 plants per minute. But, due to stoppages and turnings the final feeding rate was 28 pl/min. He referred to a study in the UK with cauliflower transplanting at 27,000 pl/ha with a three row machine and a gang of five workers (three feeding the machine, one feeding the three and one driver) that required 27 man-h/ha. Manual transplanting required 54 man^{-h}/ha. Using trays that can eliminate the fifth worker the overall work rate increased and only 22 man^{-h}/ha were sufficient.

Semi-automated transplanters have been developed over the last decade. They fulfill three main tasks: 1) feeding the seedlings to the planting mechanism, one at a time; 2) opening a furrow or hole for insertion of the seedlings and 3) firming the soil around the roots of the seedlings. In many machines an adjustable amount of water can be placed at the seedling placement position to assist in the success of transplanting.. The feeding rate is related to the forward speed, number of rows and seedling spacing along the row as given below (Srivastava, *et al.*, 2006):

$$R_{st} = \frac{60V\lambda_r}{x_x} \quad \text{.............................} (1)$$

where

R_{st} = required feed rate of seedlings, seedlings/min

v = forward speed of transplanter, m/s

λ_r = number of rows planted simultaneously by transplanter

x_x = seedling spacing along the row, m

Even though automatic transplanters have been developed, the semi-automatic transplanters are the mainstream machines, especially in open fields and in countries with small farm holdings (Srivastava, *et al.*, 2006). In Southern and Eastern Europe, as well as in Balkan countries and in other countries in Asia and Africa, manual transplanting is still the case in most open fields due to small farm holdings and because the farmers are not convinced that investing in a transplanter machine is best. Therefore, the efficiency of transplanting machines and the factors influencing it is needed.

This study presents the results of experiments carried out to identify the performance of semi-automatic transplanters and the influence on human workers as well as the comparison to manual transplanting.

Materials and Methods

A semi-automatic OTMA 4-row transplanter that could be adjusted for row spacing was pulled by a New Holland T-30 45 HP tractor for transplanting bare root plants. Both the tractor and the transplanter were 5 years old.

The study was focused on transplanting pepper, leeks and cabbage in the Vourkou region in the Perfecture of Delvine, Albania. The experiments compared transplanting with manual workers to transplanting with the 4-row semi-automatic machine. The planting mechanism was a vertical disc with four plant places of pinch type. The workers placed the plants in the plant places, which, with a slight push, closed and held the seedling until it was placed in the furrow opened by the furrow opener. The plant places were covered by rubber to avoid damages to the plants. The transplanting disc placed the seedlings in the furrow at exact distances. The disc rotating speed was transferred from the ground wheel through replaceable gears and chains. The firming of soil around the roots of the seedlings was achieved with two wheels for each row. Line markers secured the proper movement in consecutive rows of the transplanter.

The distance between the seedlings in the furrow depended on the number of plant places on the disc and the relationship between the rotation on the ground wheel (driving wheel) and the transplanting disc (adjusted by the use of appropriate gears). The gear on the ground wheel had 10, 12, 16, 18, and 23 teeth, while the gear on the transplanting disc had 16, 18, 23 and 12 teeth. The combination of the gears, the diameter of the ground wheel and the number of plant places on the transplanting disk determined the transplanting distance in the furrow. The machine also had a water tank with four water outlets, one for each row, that opened each time a seedling was placed on the ground. Watering the seedling was very important as it allowed seedlings to have adequate water for the first period in the soil until irrigation water was applied. The diameter of the transplanting unit (disk plus plant places) was 48 cm. The machine in the first trials had no line markers, while during the experiment it was added for keeping straight lines in the adjacent rows. The machine was operated on a clay soil, with a wilting point in dry basis of 18.4 % and field capacity of 32 % at a 30 cm depth (Natsis, 2001).

Soil tillage before transplanting was carried out as follows: first, ploughing at 26-30 cm depth followed by a disc harrow and a light leveller. Manure was then applied with a manure spreader. Second, ploughing at 21-25 cm depth followed by seedbed preparation with a rotary harrow. The soil tillage was the same in all crops under investigation.

Workers were uprooting the seedlings from the cold frame. The selection of the healthy seedlings for transplanting was very important for both operations (machine and manual). The seedlings were placed in disks and transported to the headlands for use. The sizes of the seedlings are shown on **Table 1**.

Transplanting by hand was carried out as follows: first, the marking of the rows by placing a thread on the soil. Then a worker with a planter placed the seedling at the exact distance within the row. A ruler with marks of the distances between the plants was used to assist the worker.

The machine worked in parallel rows. Four workers that feed the seedlings and the tractor driver. Timing of all operations was made using a stop watch. The work was followed for six days in order to assess the effect of the increased experience of the workers.

The estimation of the cost for transplanting by hand included the labour cost and the cost of marking the field. The estimation of the cost for transplanting by the semi-automatic transplanter machine included the fixed and running costs of the tractor and the implement and the labour cost. Five workers were required as noted above. The tractor was operated for 500 h/year, the cost for the driver was 6 €/h, and the cost of diesel was 0.9 €/l. The transplanter was operated 50 hours per year while transplanting 100 ha. The recovery rate was 10 % and the economical life of the tractor 15 years. The cost for the soil cultivation and the seedbed preparation was not included as it was the same either by hand or by the transplanter.

The estimation of the fixed and running costs of the tractor and the transplanter was made using the following equation (Tsatsarelis, 2000):

$$AC = \frac{(FC\,\%)\,P}{100} + \frac{cA}{Swe_f} \times [(R\,\&\,M)\,P + L + O + F + T] \quad \text{.......}(2)$$

where

AC is the annual cost of operating machine, €/yr

FV % is the annual fixed cost percentage

P is the purchase price of machine

Cis a constant = 10

A is the annual area use in ha

S is the forward speed, km/hr

W is the effective with of action of machine, m

ef is the field efficiency, decimal

R&M is the repair and maintenance cost, decimal of purchase price per hour,

L is the labor rate, €/hr

O is the oil cost, €/hr

F is the fuel cost, €/hr

(T = 0 if self-propelled)

The transplanter performance was calculated using the following equation:

$$C = 0.1b \times n \times v \times Ef \quad \text{................}(3)$$

Where

C is the performance of the machine, ha/h

n the number of rows r, n = 4

b is the transplanting width, m

v is the forward speed (km/h)

Ef is field efficiency

Ef = 1 for transplanting by hand and

Ef = 0.75 for transplanting using the transplanter.

Table 1 Size of seedlings for transplanting

Seedling sizes	Crop		
	Pepper	Cabbage	Leeks
Seedling length	125-200 mm	145-200 mm	100-200 mm
Seedling diameter	2- 5 mm	4-6 mm	4-6 mm

Results and Discussion

The performance of transplanting of the three crops used in this experiment are presented in **Table 2**.

The total costs of the tractor and the transplanter using **Eqn. 2** was 29.5 €/h, where 7.5 €/h was the cost for the tractor and 12 €/h the cost for the transplanter. The cost of transplanting both with the workers and the transplanter was as follows:

For Leeks:

1. *By hand: 39.68 (h/ha) × 4 × 6 (€/h) = 952.38 (€/ha)*
2. With transplanter: *Total Cost = a + b*
   a) Workers: *(4) × 13.88 (h/ha) × 6 (€/h) = 333.12 (€/ha)*
   b) *Tractor + transplanted machine = 13.88 (h/ha) × 29.5 (€/h) = 409.46 (€/ha)*

Total: *333.12 + 409.46 =742.58 (€/ha)*

For Peppers:

1. By hand: *13.74 × 4 (h/ha) × 6 (€/h) = 329.76 (€/ha)*
2. With transplanter machine:
   a) Workers: *4 × 5.29 (h/ha) × 6 (€/h) = 126.96 (€/ha)*
   b) *Tractor + transplanter machine = 5.29(h/ha) × 29.5 (€/h) = 156.05 (€/ha).*

Total: *126.96 + 156.05= 283.02 (€/ha)*

For Cabbage:

1. By hand: *11.16 (h/ha) × 4 × 6 (€/h) = 267.84 (€/ha)*
2. With transplanter machine:
   a) Workers: *4 × 3.66 (h/ha) × 6 (€/h) = 87.84 (€/ha)*
   b) Tractor + transplanter machine: *3.66 (h/ha) × 29.5 (€/h) = 107.97 (€/ha)*

Total: *107.97 + 87.84 = 195.81 (€/ha)*

The above analysis of the cost were summarized in **Table 3.**

As it can be seen in **Table 3**, the cost for transplanting was significantly lower with the use of transplanter than by hand. However, it is very important to take into consideration the timeliness as it can be seen in **Table 3** (ha/h). The timeliness is very important especially for vegetable production to achieve early maturity for better market prices. Moreover, the work is less tiring with the use of the transplanter.

The results of the measurements for the continuous days of experiment are presented in **Table 4**. As the workers became more experienced with the transplanter, the efficiency increased and by 81% on the sixth day.

Table 2 Transplanting performance for 3 crops

Parameters	Leeks		Pepper		Cabbage	
	By hand*	Trans-planter	By hand*	Trans-planter	By hand*	Trans-planter
Seedlings/ha	240,000	240,000	80,000	80,000	40,000	40,000
Spacing in row, cm (b)	14	14	18	18	36	36
Rows distances in cm (b)	30	30	70	70	70	70
Sowing depth in cm	8-10	8-10	8-10	8-10	7-12	7-12
Working speed km/h (avg) v	0.21	0.8	0.26	0.9	0.32	1.3
Performance (C) C = 0, 1b*n*v*Ef ha/h	0.0252	0.072	0.0728	0.189	0.0896	0.273
Working hours per ha (h/ha)	39.68	13.88	13.74	5.29	11.16	3.66
Working hours in % $\frac{transp.}{hand}$	100	34.9	100	38.5	100	32.7

* There are four workers

Table 3 Transplanting costs per ha

Transplanting	Leeks	Peppers	Cabbage
Transplanting machine (€/ ha)	742.58	283.02	195.81
By hand (€/ha)	952.38	329.76	267.84
Machine transplanting Cost/ transplanting by hand) (%)	77.97	85.82	73.10

Table 4 Performance of the transplanting operation of the machine

Transplanting Machine Efficiency	Working days					
	1	2	3	4	5	6
Speed in km/h (average)	0.65	0.84	0.93	1.08	1.15	1.3
Number of seedlings planted by each worker per minute	17	18	21	25	28	30
Transplanter work rate for each working hour per ha (average)	0.16	0.19	0.23	0.25	0.26	0.29
Comparative efficiency %	100	118.8	143.7	156.3	162.5	181.2

Conclusions

From the results of this work it can be concluded that:

- Transplanting with the semi-automatic transplanter machine compared with transplanting by hand, achieved higher performance in ha per hour and less fatigue to workers.
- Manual transplanting was more expensive than by the semi-automatic transplanter.
- The efficiency increased with the days of operating, as the workers in the machine obtained more experienced. This should show the importance of using experienced workers.

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The Effect of some Components on Performance of Flat Plate Solar Water Collectors

by

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Abstract

Copper and aluminum type flat plate solar water collectors are widely used for supplying hot water for domestic applications in the world. It is possible to see many studies about thermal efficiency of any type of flat plate solar collector in the literatures. However, not enough test results are reported in the literature on the effect of solar water collector components. Therefore, in this experimental study, the effects of some components such as construction materials (copper and aluminum), absorber surfaces (black painted copper sheet, black cooper selective surface and black painted aluminum sheet), covers (normal flat glass, prismatic glass and normal tempered glass) and radiation reflector material located between insulation and absorber surfaces on performance of solar water collectors were investigated. All tests were conducted according to EN (European Norms) 12975-2 (2003) (Anon., 2003). The solar water collectors were tested in summer conditions (in August, in Samsun, Turkey). Each collector was tested four days for each component and 5 hours (10:00 to 15:00) for each day. Four different fluid inlet temperatures were used for each test. Fluid inlet temperatures were changed between 35 ºC…60 ºC while the ambient air temperature changed between 28 ºC…34 ºC, during experiments. The results showed that thermal efficiencies of 1) copper type collector with black painted copper absorber surface and normal flat glass cover, 2) copper type collector with black copper selective absorber surface and normal flat glass cover, 3) copper type collector with black painted copper absorber surface and prismatic glass cover, 4) copper type collector with black copper selective absorber surface, 5) normal flat glass and radiation reflector, aluminum type collector with black painted aluminum absorber surface and normal flat glass cover, and 6) aluminum type collector with black painted aluminum absorber surface and normal tempered glass cover were 56.26 %, 63.39 %, 65.63 %, 64.51 %, 58.19 % and 67.82 %, respectively.

Introduction

Water is used in households predominantly for drinking, cooking and hygiene purposes such as bathing and the washing of dishes. Each person in a household requires at least 20 L of potable water per day, half of which is for personal hygiene. Water is also needed to be heated to meet hot water necessity for cooking and hygiene. The use of hot water during the day is approximately the same in the morning as it is in the evening, but less during the afternoon in households (Nieuwoudt and Mathews, 2005).

Different heat sources are employed for heating water. However, in most developing countries, supplies of non-renewable sources of energy are either unavailable, unreliable or too expensive. In renewable energy sources, solar energy is the most appropriate for heating water. This energy allows independent systems to be constructed and possesses a thermal conversion mode which necessitates a simple technology (Koyuncu, 2006).

Solar energy received on the ground is abundant and inexhaustible. In addition to its inexhaustible nature, solar energy has the advantage of being a source of nonpolluting energy. This energy could be harnessed in several ways. The most promising energy forms are solar collectors with thermal conversion, which can be used to heat water for domestic and industrial applications. These applications are developing most rapidly and are the basis of a small but growing industry (Hazami *et al*., 2005).

Collectors are the heart of the solar processes. In a solar collector, energy is transferred from a distant source of radiant energy to a fluid. The flux of incident radiation is, at best, approximately 1,100 W/m^2 (without optical concentration), and it is variable. The wavelength range is from 0.3 to 3 μm, which is considerably shorter than that of the emitted radiation from most energy absorbing surfaces. Thus, the analysis of solar collectors presents unique problems of low and variable energy fluxes and the relatively large importance of radiation. Flat plate collectors can be designed for applications requiring energy delivery at moderate temperatures, up to perhaps 100 ºC above ambient temperature. They use both beam and diffuse solar radiation, do not require tracking of the sun, and require little maintenance. The major applications of these units are in solar water heating, building heating, air conditioning, and industrial process heat. The importance of flat plate collectors in thermal processes is such that their thermal performance is treated in considerable de-

Fig. 1 Technical parameters of copper type collector with black painted copper absorber surface and normal flat glass cover (all measurements are in mm)

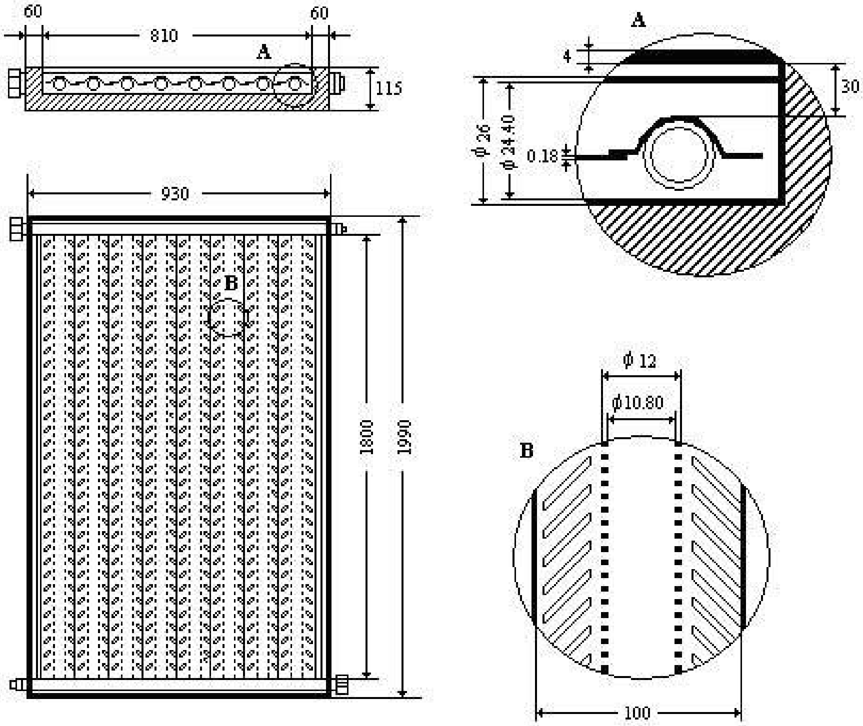
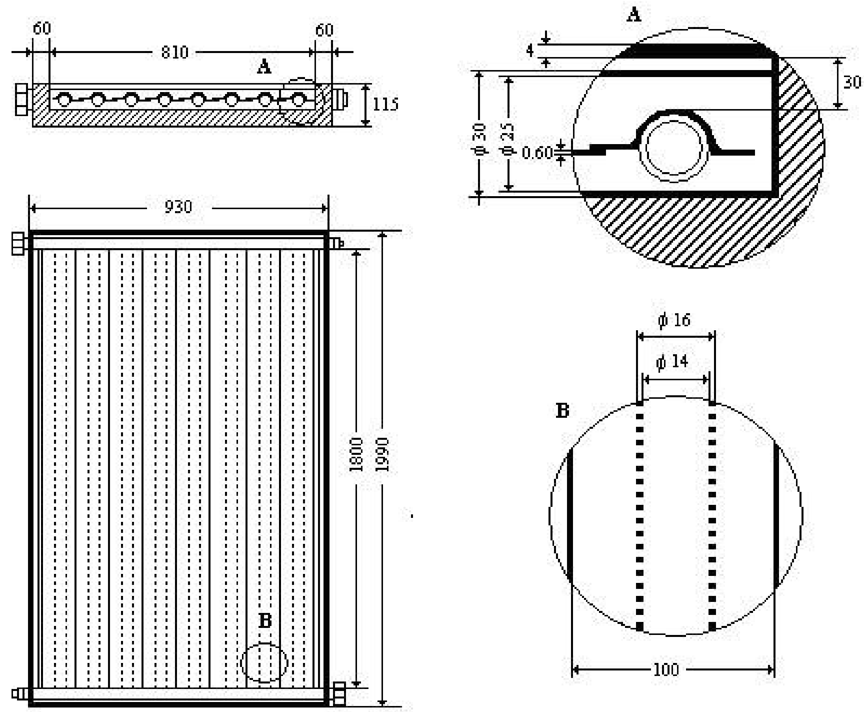


Fig. 2 Technical parameters of aluminium type collector with black painted aluminium absorber surface and normal flat glass cover (all measurements are in mm)



tail. This is done to develop an understanding of how the component functions (Duffie and Beckman, 1991).

Many types of conventional solar collectors with metal absorber plates and glass covers are widely used to transform solar energy into heat for heating water for domestic applications and industrial processes in Turkey and the world (Koyuncu and Ultanir, 1997 and Nahar, 2003). These collectors have high efficiency and they are approximately, 2.0, 3.5, 4.0, 6.0, 7.0 and 12.0 times more profitable when compared with other energy sources such as wood, coal, natural gas, oil, LPG and electricity, respectively, for heating water for domestic applications in Turkey (Koyuncu and Ultanir, 1997). It is possible to see many studies about thermal efficiency of any type of solar flat plate collector in the literature (Lu *et al*., 2003 and Karim and Hawlader, 2006). However, not enough test results are reported in the literature on the effect of solar water collector components. Therefore, in this paper, we present the effects of components such as construction materials, absorber surfaces, covers and reflectors on performance of solar water collectors.

Solar Collectors and Experimental Setup

Collectors were made of a number of components as listed below (**Figs. 1** and **2**).

Construction Materials:

1. ***Copper type collector:*** All pipes and absorber surface completely made of copper material (**Fig. 1**).
   Aluminum type collector: All pipes and absorber surface completely made up of aluminum material (**Fig. 2**).
2. ***Absorber surfaces:*** Black painted copper sheet, black cooper selective surface and black painted aluminum sheet,
3. ***Covers:*** A transparent cover of one layer of glass material that transmits visible and near infrared radiation (< 3 µm) and absorbs far infrared radiation (> 3 µm) (Riffat *et al*., 2000). The covers are in normal flat glass, prismatic glass and normal tempered glass with a thickness of 4.00 mm.
4. ***Radiation reflector material:*** Bright aluminum sheet located between the insulation and the absorber surface.
5. ***Gap:*** 30 mm gap maintained between the cover and absorber plate.
6. ***Frame and insulator:*** The frame constructed from aluminum profiles and sheets for casing these components, with glass wool insulating material on the back and on the sides to minimize heat losses. The insulator is in glass wool of 60 mm thickness. The thermal conductivity coefficient of the insulator is equal to 0.04 W/(m K).

The effect of components on efficiency of flat plate solar water collectors were separately investigated. In the first step, copper type collector with black painted copper absorber surface and normal flat glass cover (**Fig. 1**); in the second step, copper type collector with black copper selective absorber surface and normal flat glass cover; in the third step, copper type collector with black painted copper absorber surface and prismatic glass cover; in the fourth step, copper type collector with black copper selective absorber surface, normal flat glass and radiation reflector; in the fifth step, aluminum type collector with black painted aluminum absorber surface

Fig. 2 Technical parameters of aluminum type collector with black painted aluminum absorber surface and normal flat glass cover (all measurements are in mm) and normal flat glass cover (**Fig. 2**) and in the sixth step, aluminum type collector with black painted aluminum absorber surface and normal tempered glass cover were tested. All tests were conducted according to EN (European Norms) 12975-2. Solar water collectors were mounted on a stand facing south at an inclination angle of 45º. The experimented collector, mainly equipped with an electric fluid heater, fluid inlet temperature adjuster, thermostat, centrifugal fluid circulation pump, pyranometer, datalogger, anemometer, axial fan (blower for producing wind), air speed adjuster (regulator of variable transformer), temperature indicators, mass flow measurement container, RS 232 connection, a PC and specially designed 'Smart Control for OPUS' data processing software (**Fig. 3**). The experimental setup was instrumented for the measurement of solar radiation, temperature of the atmosphere air, outlet and inlet fluid temperature, absorber surface temperature, fluid mass low and wind velocity. These various thermal efficiency parameters were measured and recorded at an interval of 30 min. The experiments took place at Samsun (North of Turkey): Latitude = 41,21º, Longitude = 36,15º and Altitude = 4 m. The collector system was tested in summer conditions (in August). It was tested four days for each component and 5 hours (10:00 to 15:00) for each day. Four different fluid inlet temperatures were also used for each test. Fluid inlet temperatures were changed between 35 ºC…60 ºC while the ambient air temperature changed between 28 ºC…34 ºC, during experiments. The fluid mass flow rate through the collector was kept constant at 0.02 kg/(s m^2) by manipulating the flow path (by adjustable circular valve) of the fluid throughout the experimentation.

Theoretical Analysis

The collectors operate under quasi steady-state conditions (**Fig. 4**). In these conditions, the performance of a solar collector is described by an energy balance that indicates the distribution of incident solar energy into useful energy gain, thermal losses, and optical losses (**Figs. 5** and **6**), (Anon., 2003; Duffie and

Beckman, 1991; Koyuncu and Ultanir, 1997 and Riffat *et al*., 2000).

The useful heat gain by a collector can be expressed as

$$q_s = I_R A_c = q_{loss} + q_u \quad (1)$$

$$q_u = q_s - q_{loss} = I_R A_c - q_{loss} \quad (2)$$

$$q_{loss} = q_{loss,\,opt} + q_{loss,\,t} + q_{loss,\,be} \quad (3)$$

$$q_u = I_R A_c - q_{loss,\,opt} - q_{loss,\,t} - q_{loss,\,be} \quad (4)$$

$$SA_c = I_R A_c - q_{loss,\,opt} = I_R A_c\,(\tau_c \alpha_p) \quad (5)$$

$$q_u = SA_c - q_{loss,\,t} - q_{loss,\,be} \quad (6)$$

$$q_u = I_R A_c\,(\tau_c \alpha_p) - q_{loss,\,t} - q_{loss,\,be} \quad (7)$$

$$q_u = \dot{m} c_{p,f}\,(T_{f,o} - T_{f,i}) \quad (8)$$

A measure of collector performance is the collector efficiency, defined as the ratio of useful heat gain over any time period to the incident

Fig. 3 Schematic presentation of solar collectors experimentation

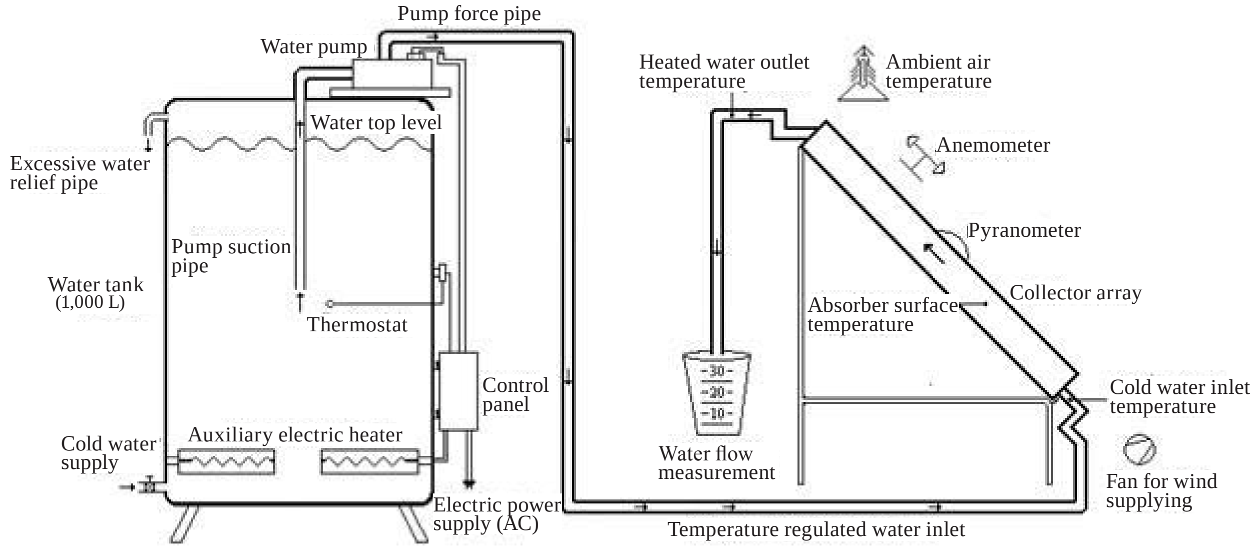


Fig. 4 Description of the flat plate solar water collector (main parts and general heat transfer exchanges)

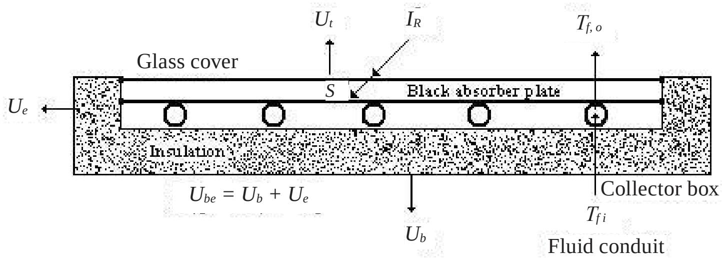


Fig. 5 Equivalent thermal network for flat plate solar water collector

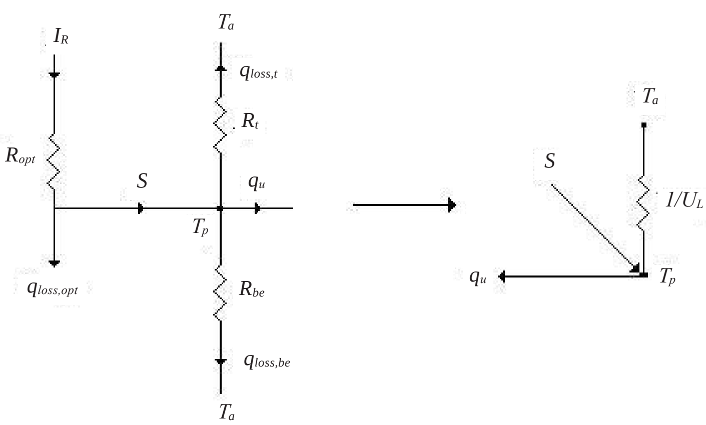
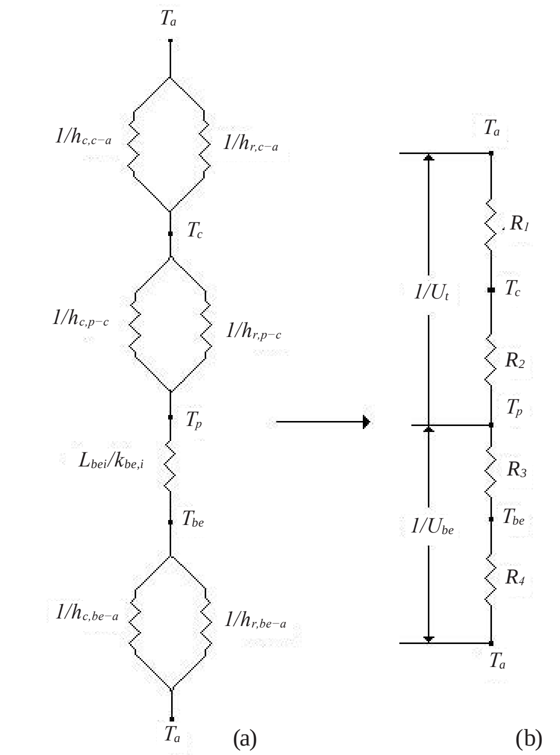


Fig. 6 Thermal network for the flat plate solar water collector in terms of conduction, convection and radiation (a), and in terms of resistances between plates (b)



solar radiation over the same period we can, thus, define efficiency as,

$$\eta = \frac{q_u}{q_s} \quad \text{(9)}$$

$$\eta = \frac{\sum q_u}{\sum q_s} \quad \text{(10)}$$

From **Eqns. 1, 8, 9** and **10**,

$$\eta = \frac{\dot{m} c_{p,f} \left(T_{f,o} - T_{f,i}\right)}{I_R A_c} \quad \text{(11)}$$

In addition, it is convenient to define a quantity that relates the actual useful energy gain of a collector to the useful gain if the whole collector surface were at the fluid inlet temperature. This is quantity is colled the collector heat removal factor F_R. The actual useful energy gain and the collector heat removal factor can be expressed as

$$q_u = A_c F_R \left[F_{D,S} S - U_L \left(T_{f,o} - T_{f,i}\right)\right] \quad \text{(12)}$$

$$F_R = \frac{\dot{m} c_{p,f} \left(T_{f,o} - T_{f,i}\right)}{A_c \left[S - U_L \left(T_{f,i} - T_a\right)\right]} \quad \text{(13)}$$

$$F_R = \frac{\dot{m} c_{p,f}}{A_c U_L} \left[\frac{T_{f,o} - T_{f,i}}{\frac{S}{U_L} - \left(T_{f,i} - T_a\right)} \right] \quad \text{(14)}$$

$$F_R = \frac{\dot{m} c_{p,f}}{A_c U_L} \left[\frac{\left(T_{f,o} - T_a - \frac{S}{U_L}\right) - \left(T_{f,i} - T_a - \frac{S}{U_L}\right)}{\frac{S}{U_L} - \left(T_{f,i} - T_a\right)} \right] \quad \text{(15)}$$

$$F_R = \frac{\dot{m} c_{p,f}}{A_c U_L} \left[1 - \frac{\frac{S}{U_L} - \left(T_{f,o} - T_a\right)}{\frac{S}{U_L} - \left(T_{f,i} - T_a\right)} \right] \quad \text{(16)}$$

U_L is the collector overall heat loss coefficient. The thermal energy is lost from the collector to the surroundings by conduction, convection and infrared radiation. U_L is equal to the sum of energy loss through the top U_t, bottom U_b and edge U_e of the collectors given below (**Figs. 5** and **6**) (Koyuncu and Ultanir, 1997):

$$U_L = U_t + U_b + U_e = U_t + U_{be} \quad \text{(17)}$$

The energy loss through the top is the result of convection and radiation between parallel plates. The top loss coefficient from the collector plate to the ambient is

$$\frac{1}{R_1} = \frac{1}{\frac{1}{h_{c,c-a}}} + \frac{1}{\frac{1}{h_{r,c-a}}} = \frac{h_{c,c-a} + h_{r,c-a}}{1} \quad \text{(18)}$$

$$R_1 = \frac{1}{h_{c,c-a} + h_{r,c-a}} \quad \text{(19)}$$

$$\frac{1}{R_2} = \frac{1}{\frac{1}{h_{c,p-c}}} + \frac{1}{\frac{1}{h_{r,p-c}}} = \frac{h_{c,p-c} + h_{r,p-c}}{1} \quad \text{(20)}$$

$$R_2 = \frac{1}{h_{c,p-c} + h_{r,p-c}} \quad \text{(21)}$$

$$\frac{1}{U_t} = R_1 + R_2 = \frac{1}{h_{c,c-a} + h_{r,c-a}} + \frac{1}{h_{c,p-c} + h_{r,p-c}} \quad \text{(22)}$$

$$U_t = \left[\frac{1}{h_{c,c-a} + h_{r,c-a}} + \frac{1}{h_{c,p-c} + h_{r,p-c}} \right]^{-1} \quad \text{(23)}$$

Besides, h_W, U_b and U_e must be obtained from the equations as follows (Bagach *et al.*, 2000):

$$h_{c,c-a} = h_w = 2.8 + 3\upsilon \quad \text{(24)}$$

$$U_t = \left[\frac{1}{h_w + h_{r,c-a}} + \frac{1}{h_{c,p-c} + h_{r,p-c}} \right]^{-1} \quad \text{(25)}$$

$$R_3 = \frac{L_{be,i}}{k_{be,i}} + \frac{A_c}{\left(\frac{k_{be,i}}{L_{be,i}}\right) A_{c,e}} \quad \text{(26)}$$

$$\frac{1}{R_4} = \frac{1}{\frac{1}{h_{c,be-a}}} + \frac{1}{\frac{1}{h_{r,be-a}}} = \frac{h_{c,be-a} + h_{r,be-a}}{1} \quad \text{(27)}$$

$$R_4 = \frac{1}{h_{c,be-a} + h_{r,be-a}} \quad \text{(28)}$$

$$\frac{1}{U_{be}} = R_3 + R_4 \quad \text{(29)}$$

$R_4 \cong 0$ is very small and negligible, so

$$\frac{1}{U_{be}} = R_3 = \frac{L_{be,i}}{k_{be,i}} + \frac{A_c}{\left(\frac{k_{be,i}}{L_{be,i}}\right) A_{c,e}} \quad \text{(30)}$$

$$U_{be} = \frac{k_{be,i}}{L_{be,i}} + \frac{\left(\frac{k_{be,i}}{L_{be,i}}\right) A_{c,e}}{A_r} \quad \text{(31)}$$

$$U_t = \left(\frac{1}{h_{c,p-c} + h_{r,p-c}} + \frac{1}{h_w + h_{r,c-a}} \right)^{-1} \quad \text{(32)}$$

In order to find $h_{c,\ p-c}$, $h_{r,\ p-c}$ and $h_{r,\ c-a}$ for these solar water collectors, a series of equations seen below must be used (Koyuncu and Ultanır, 1997).

$$R_a = \frac{g \beta_{p-c} \Delta T_{p-c} L_{p-c}^{\ 3}}{\nu_{a,p-c} \lambda_{a,p-c}} \quad \text{(33)}$$

$$\beta = \frac{1}{T_m} \quad \text{(34)}$$

$$T_m = \frac{T_c + T_p}{2} \quad \text{(35)}$$

$$P_r = \frac{\nu_{a,p-c}}{\lambda_{a,p-c}} \quad \text{(36)}$$

$$R_a = \frac{g \Delta T_{p-c} L_{p-c}^{\ 3} P_r}{T_m \nu_{a,p-c}^{\ 2}} \quad \text{(37)}$$

$$R_a = \frac{g \beta_{p-c} \Delta T_{p-c} L_{p-c}^{\ 3}}{\nu_{a,p-c}^{\ 2}} \quad \text{(38)}$$

$$N_u = 1 + 1{,}44 \left[1 - \frac{1708 \left(\sin 1{,}8 \beta_{p-c}\right)^{1,6}}{R_a \cos \beta_{p-c}} \right] \left[1 - \frac{1708}{R_a \cos \beta_{p-c}} \right] + \left[\left(\frac{R_a \cos \beta_{p-c}}{5830} \right)^{1/3} - 1 \right] \quad \text{(39)}$$

$$h_{c,p-c} = \frac{N_u k_{a,p-c}}{L_{a,p-c}} \quad \text{(40)}$$

$$h_{r,p-c} = \frac{\sigma \left(T_p^{\ 2} + T_c^{\ 2}\right)\left(T_p + T_c\right)}{\frac{1}{\varepsilon_p} + \frac{1}{\varepsilon_c} - 1} \quad \text{(41)}$$

$$h_{r,c-a} = \varepsilon_c \sigma \left(T_c^{\ 2} + T_a^{\ 2}\right)\left(T_c + T_a\right) \quad \text{(42)}$$

$$T_c = T_p - \frac{U_t \left(T_p - T_a\right)}{h_{c,p-c} + h_{r,p-c}} \quad \text{(43)}$$

Another equation to find U_t was developed by Klein. U_t was also calculated by using this equation (44). It gave the same results with very small and negligible differences (Duffie and Beckman, 1991).

$$U_t = \left[\frac{n}{\frac{C}{T_p} \left[\frac{\left(T_p - T_a\right)}{\left(n + f\right)} \right]^e} + \frac{1}{h_w} \right]^{-1} + \left[\frac{\sigma \left(T_p + T_a\right)\left(T_p^{\ 2} + T_a^{\ 2}\right)}{\left(\varepsilon_p + 0{,}00591 n h_w\right)^{-1} + \frac{2n + f - 1 + 0{,}133 \varepsilon_p}{\varepsilon_c} - n} \right] \quad \text{(44)}$$

In **Eqn. 44**,

$$f = \left(1 + 0{,}089 h_w - 0{,}1166 h_w \varepsilon_p\right)\left(1 + 0{,}07866 n\right) \quad \text{(45)}$$

$$C = 520\left(1 - 0{,}000051 \beta_{a,p-c}^{\ 2}\right) \quad \text{(46)}$$

$$e = 0{,}430 \left(1 - \frac{100}{T_p}\right) \quad \text{(47)}$$

Results and Observations

Copper type collector with black painted copper absorber surface and normal flat glass cover, copper type collector with black copper selective absorber surface and normal flat glass cover, copper type collector with black painted copper absorber surface and prismatic glass cover, copper type collector with black copper selective absorber surface, normal flat glass and radiation reflector, aluminum type collector with black painted aluminum absorber surface and normal flat glass cover, and aluminum type collector with black painted aluminum absorber surface and normal tempered glass cover described in the previous sections were tested under the same outdoor conditions in the summer of 2006. Data such as fluid inlet temperature (Tf, i), fluid outlet temperature (Tf, o), fluid inlet and outlet temperature differences (ΔT), mean temperature of the fluid (Tm) ambient temperature (Ta), collector thermal efficiency (η) and incident solar radiation (IR) regarding these collectors are given in **Figs. 7, 8, 9, 10, 11** and **12**. These figures show that the efficiencies of these collectors were 56.26 %, 63.39 %, 65.63 %, 64.51 %, 58.19 % and 67.82 %, respectively. The black copper selective absorber surface increase thermal efficiency by 7.13 % (**Figs. 7** and **8**).

The prismatic glass increased the efficiency as 9.37 % (**Figs. 7** and **9**). The radiation reflector increased the efficiency by 1.12 % (**Figs. 8** and **10**). The normal tempered glass cover increased the efficiency by 9.63 (**Figs. 11** and **12**).

Conclusions

The main conclusion of this experimental investigation was that black copper selective absorber surface, prismatic glass, radiation reflector made up of bright aluminum sheet and located between insulation and absorber surface, and normal tempered glass increased thermal efficiencies of solar water collectors by 7.13 %, 9.37 %, 1.12 % and 9.63 %, respectively. Therefore, it can be recommended that, in order to develop thermal efficiency of flat plate solar water collectors, black copper selective absorber surface, prismatic or normal tempered glass and a radiation reflector should be used instead of black painted copper absorber surface, black painted aluminum absorber surface, normal flat glass and unreflecting materials.

Fig. 7 The results of copper type collector with black painted copper absorber surface and normal flat glass cover

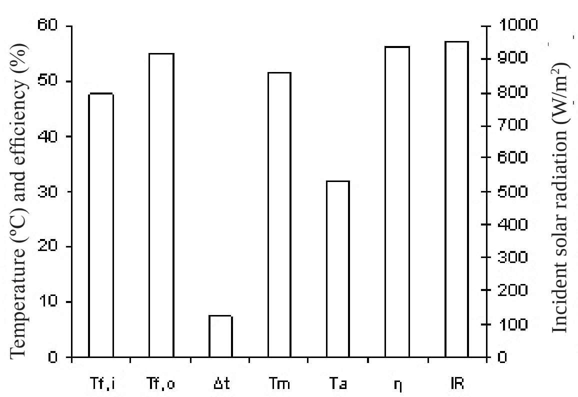


Fig. 8 The results of copper type collector with black copper selective absorber surface and normal flat glass cover

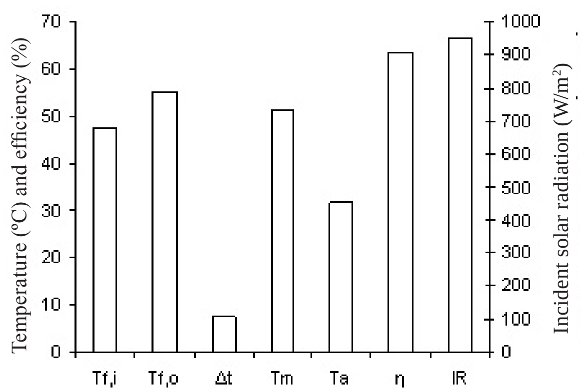


Fig. 9 The results of copper type collector with black painted copper absorber surface and prismatic glass cover

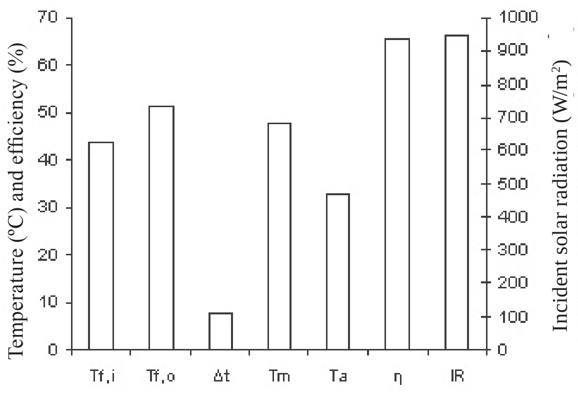


Fig. 10 The results of copper type collector with black copper selective absorber surface, normal flat glass and radiation reflector

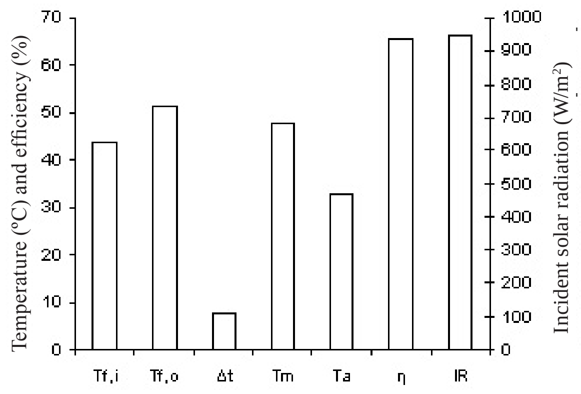


Fig. 11 The results of aluminum type collector with black painted aluminum absorber surface and normal flat glass cover

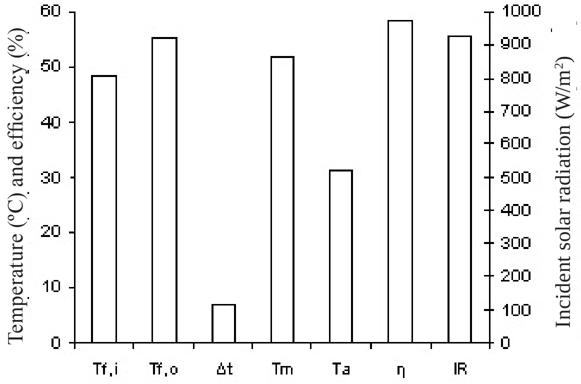
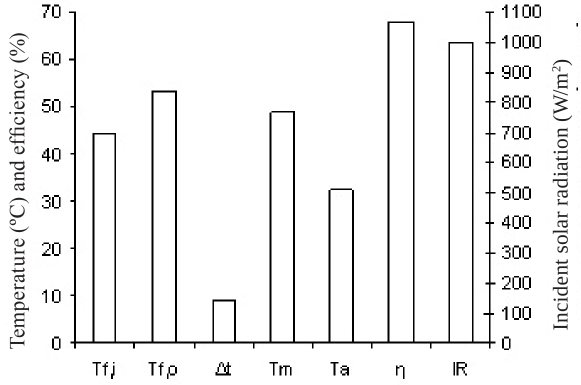


Fig. 12 The results of aluminum type collector with black painted aluminum absorber surface and normal tempered glass cover



Nomenclature	
A_c	Collector area, m²
$A_{c,e}$	Collector edge surface area, m²
$c_{p,f}$	Fluid specific heat at constant pressure, J/(kg. K)
$F_{D,S}$	Collector dust and shading factor
F_R	Collector heat removal factor
g	Gravitational acceleration, m/s²
$h_{c,c\text{-}a}$	Convection heat transfer coefficient between cover and ambient air, W/(m². K)
$h_{c,be\text{-}a}$	Convection heat transfer coefficient between bottom-edge and ambient air, W/(m². K)
$h_{c,p\text{-}c}$	Convection heat transfer coefficient between plate and cover, W/(m². K)
$h_{r,be\text{-}a}$	Radiation heat transfer coefficient between bottom-edge and ambient air, W/(m². K)
$h_{r,c\text{-}a}$	Radiation heat transfer coefficient between cover and ambient air, W/(m². K)
$h_{r,p\text{-}c}$	Radiation heat transfer coefficient between plate and cover, W/(m². K)
h_w	Wind heat transfer coefficient, W/(m². K)
I_R	Incident solar radiation, W/m²
$k_{a,p\text{-}c}$	Air thermal conductivity between plate and cover, W/(m . K)
$k_{be,i}$	Bottom-edge insulation thermal conductivity, W/(m . K)
$L_{be,i}$	Bottom-edge insulation thickness, m
$L_{a,p\text{-}c}$	Air length between plate and cover, m
$\dot{m}$	Fluid mass flow rate, kg/s
n	Number of cover
N_u	Nusselt Number
P_r	Prandtl number
q_{loss}	Thermal energy losses, W
$q_{loss,be}$	Thermal energy losses through the bottom-edge, W
$q_{loss,opt}$	Optical energy losses, W
$q_{loss,t}$	Thermal energy losses through the top, W
q_s	Incident solar power, W

Nomenclature	
q_u	Useful thermal power gain, W
R_a	Rayleigh number
R_{be}	Bottom-edge resistance, (m². K)/W
R_{opt}	Optical resistance, (m². K)/W
R_t	Top resistance, (m². K)/W
$R_{1,2,3,4}$	Resistances, (m². K)/W
S	Absorbed solar radiation, W/m²
T_a	Ambient air temperature, K
T_{be}	Bottom-edge surface temperature, K
T_c	Cover temperature, K
$T_{f,i}$	Fluid inlet temperature, K
$T_{f,o}$	Fluid outlet temperature, K
T_p	Plate surface temperature, K
T_m	Mean temperature, K
U_{be}	Bottom-edge heat loss coefficient, W/(m². K)
U_L	Collector overall heat loss coefficient, W/(m². K)
U_t	Top heat loss coefficient, W/(m².K)
υ	Wind speed, m/s

Greek	
$\lambda_{a,p\text{-}c}$	Thermal diffusivity of air between plate and cover
α_p	Plate absorptance coefficient
$\beta_{a,p\text{-}c}$	Volumetric expansion coefficient of air between plate and cover (for an ideal gas β = 1/Tm)
$\Delta T_{p\text{-}c}$	Temperature differences between plate and cover, K
ε_c	Cover emittance
ε_p	Plate emittance
η	Collector thermal efficiency
$\nu_{a,p\text{-}c}$	Kinematic viscosity of air between plate and cover, m²/s
σ	Stefan-Boltzmann constant (5.67×10^{-8}), W/(m². K⁴)
τ_c	Cover transmittance
$\tau_c\alpha_p$	Transmittance-absorptance product

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NEWS

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We would like to call all of you who combat the global crises of food and natural resources using PA technologies, and who contribute local community and industry development with agriculture, and who crate agro-information technology. We are looking forward to seeing all participants in Obihiro.

Sakae SHIBUSAWA

Chair of Management Committee of Asian Conference on Precision Agriculture

Effect of Various Air Drying Temperatures on Quality of Dehydrated Garlic Slices (Allium Sativum. L.)

by

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Abstract

Prepared garlic slices of 2.3 mm thickness divided into six lots for experimental study were dried at six different temperature regimes viz., 40 ºC 45 ºC, 50 ºC, 55 ºC, 60 ºC and 65 ºC, respectively, in a hot air cabinet drier to study the effect of various air drying temperatures on quality parameters of garlic after dehydration and after a storage period of 4 months. Dehydration at 65 ºC (T_6) resulted in the shortest drying run duration of 337.67 minutes and least product moisture content of 4.38 % after dehydration when compared to other treatments and even after 4 months of storage (8.21 %). Treatment T_6 (65 ºC), in terms of biochemical attributes, was also found to be quite acceptable with significantly higher carbohydrate content (69.06 g/ 100 g), proteins (14.85 g/ 100 g), ascorbic acid (5.50 mg/ 100 g), volatile oil percentage (0.323) and low browning index scores (0.045) after dehydration and even after a storage period of 4 months (carbohydrate content; 67.12 g/ 100 g, proteins; 11.21 g/ 100 g, ascorbic acid; 5.50 mg/ 100 g, volatile oil percentage; 0.300 and browning index; 0.065). In terms of sensory quality treatment, T_6 (65 ºC) was superior with an overall acceptability score of 8 after dehydration and 7 after a storage period of 4 months.

Introduction

Garlic (Allium Satvium L.) a native of middle east Asia is a semi perishable spice. The moisture content of the bulb and surrounding relative humidity play an important role in maintaining its keeping quality (Ambrose and Sreenarayanan,1998). It has a tendency to loose its moisture on storage, which causes weight loss and shriveling of cloves to an extent of 40-45 % (Prakash *et al*., 1994). Due to lack of suitable storage and transport facilities, about 20 % of the fresh crop is wasted by respiration, transpiration and microbial spoilage (Pruthi *et al*., 1959). Dehydration is the simple, low cost and economical method of preservation, where quality parameters can be maintained considerably. However, dehydration of garlic tried by earlier workers is time consuming and there is a very little information regarding the dehydration of garlic for extending its shelf life. In view of the above, an investigation was made with the objective to identify the most suitable air drying temperature for dehydration of garlic to extend shelf life and maintain the quality parameters.

Material and Methods

The present investigation was carried out in the Department of Post Harvest Technology, Bidhan Chandra Krishi Viswavidyalaya. Freshly harvested local variety of garlic (Lashan) was procured from the local market of District Nadia for the study, and brought to the laboratory. After loosening the cloves from the bulb by manual scrubbing and peeling, the cloves were cut into 2-3 mm thickness using sharp stainless steel knives. Experimental treatments comprised of six different temperature regimes viz., 40 ºC, 45 ºC, 50 ºC, 60 ºC and 65 ºC. The prepared garlic slices were spread on aluminum trays with tray dimensions of 59 cm^2 and tray loading capacity of 3 Kg/m^2 in a cabinet air drier. The cabinet dryer was operated at an air velocity of 1.45 m/sec and was constant for all the treatments during the drying operation, Each drying run of a particular treatment was continued until the product attained constant weight. For assessing the termination time of drying, the tray containing the material was weighed regularly at an interval of 1 hour for loss in weight up to the point when it attained a constant weight. Each drying run of a particular treatment was replicated thrice. The dehydrated garlic slices, dried at six different temperature regimes as mentioned earlier, were analyzed for various physical and biochemical constituents; viz., moisture content (% wb), carbohydrates (g/100 g), ascorbic acid (mg/100 g), proteins (g/100 g), volatile oil (%), browning index and organoleptic quality. The moisture content (% wb), both initial and after dehydration was determined, using standard methods of AOAC (1980). Total carbohydrates (g/100 g) were estimated by Anthrone method and proteins (g/100 g) by lowry method (Sadasivam and Manickam, 1992). Ascorbic acid (mg/100 g) was estimated by 2, 6-dichlorophenol indophenol dye method as determined by the method of Ranganna (1991). Percent volatile oil was determined by chloramine-T method (Shankaranarayana *et al*., 1982) and browning index by the method of Ranganna (1991). For sensory quality evaluation, dehydrated material was subjected to organoleptic tests after dehydration, i.e. color, flavour, texture and overall acceptability by a semitrained panel of eight judges and acceptability expressed in terms of orgonaleptic score on a 9 point hedonic scale with a rating of 1 for extremely dislike and 9 for excellent (Dasgupta *et al*., 1999). Shelf life of dehydrated garlic slices were evaluated by packing them in 200 gauge polyethylene pouches sealed air tight and stored on racks at room temperature. The temperature during the period of study ranged from 25-32 ºC. The relative humidity varied between 85-90 %. Samples were drawn at the end of storage period (4 months) and analyzed for various physical and biochemical constituents as mentioned earlier. The data regarding various quality parameters were statistically analysed by completely randomised design (Panse and Sukhatme, 1985.)

Results and Discussion

Moisture Content and Drying Time

Effect of various air drying temperatures on moisture content of dehydrated garlic slices after dehydration and after a storage period of 4 months is presented in the **Table 2**. Pursual of the data revealed significant differences among the treatments for the moisture content values after dehydration. Dehydration in all the treatments terminated at a moisture content range of 4.38 to 5.75 % with significantly higher moisture content of 5.75 % recorded in treatment T_2 (45 ºC) and lowest moisture content of 4.38 % recorded in treatment T_6 (65 ºC). However, moisture content of T_1 (40 ºC), T_2 (45 ºC), T_4 (55 ºC) and T_5 (60 ºC) were statistically at par with each other. Moisture content of treatment T_6 (65 ºC) was quite acceptable, as the water activity of the product was expected to be low, which determines its storage stability. It was also evident from the **Table 2** that moisture content of dehydrated garlic slices after dehydration had an inverse relationship with the drying temperature levels, which can be attributed to rapid moisture diffusivity and higher moisture migration at high temperature. Dehydration time varied significantly among treat-

Table 1 Proximate composition of freshly harvest garlic cloves

Constituent	Content
Moisture content (% wb)	68.29
Carbohydrates (g/100 g)	21.36
Ascorbic acid (mg/100g)	14.21
Proteins (g/ 100g)	4.96
Volatile oil (%)	0.10

Table 2 Final moisture content of dehydrated garlic slices after dehydration and its corresponding drying time and final moisture content after storage period of 4 months

Treatment	Final moisture content (% wb) after dehydration	Drying time (minutes)	Final moisture content after 4 month of storage
T_1 (40 ºC)	5.63	590.00	11.97
T_2 (45 ºC)	5.75	571.67	11.20
T_3 (50 ºC)	5.37	458.33	11.58
T_4 (55 ºC)	4.95	410.00	9.97
T_5 (60 ºC)	4.95	388,878.33	9 .69
T_6 (65 ºC)	4.38	337.67	8.21
SE (m ±)	0.2906	10.1325	0.3624
C.D (P = 0.05)	0.6332	22.0766	0.7896

wb = wetbasis

ments with a significantly maximum time of 540 minutes taken for dehydration in treatment T_1 (45 ºC) and minimum drying time of 337.67 minutes taken for dehydration at the higher temperature level of 65 ºC (T_6). This was because the product in contact with the drying surface at higher temperature dried quicker than at a comparatively lower drying temperature. The results were in agreement with Ambrose and Sreenarayanan (1998) who reported the shortest time duration of 4.5 hours during mechanical drying of garlic at an air drying temperature of 65 ºC. Moisture content recorded a significant rise after 4 months of storage in all the treatments with a significantly high moisture content (11.97 %) recorded in treatment T_1 (40 ºC) and the lowest moisture content of 8. 21 % recorded in treatment T_6 (60 ºC). However, moisture content was non significant among the treatments T_4 (55 ºC) and T_5 (60 ºC).

Biochemical Characteristics

Effect of various air drying temperatures on biochemical characteristics of garlic slices after dehydration and after a storage period of 4 months is presented in the **Table 3**. Pursual of the data revealed significant differences among the treatments for all the biochemical parameters under study. Initial carbohydrate content of 21.36 % (**Table 1**) was found to increase in all the treatments after dehydration with significantly higher carbohydrate content (69.06 g/ 100 g) recorded in Treatment T_6 (65 ºC) and lowest carbohydrate content (52.75 g/100 g) recorded in treatment T_1 (40 ºC). It was evident from the **Table 3** that lower carbohydrate losses of 2.81 % were exhibited by T_6 treatment in comparison with other treatments while losses of carbohydrates were found to be high in treatment T_1 (40 ºC). Protein content was significantly higher in treatment T_6 (65 ºC) (14.85 g/ 100 g). The significantly lowest protein content of 13.51g/100 g was exhibited by treatment T_1 (40 ºC). Protein content decreased appreciably after a storage period of 4 months with significantly higher protein content (11.21 g/ 100 g) and less loss of protein (24.52 %) recorded in treatment T_6 (65 ºC) in comparison to other treatments and significantly lowest protein content (9.51 g/ 100 g) and higher losses of protein (29.61 %) recorded in treatment T_1 (40 ºC) in comparison to other treatments. Increase in the concentration of carbohydrates and proteins after dehydration could be attributed to concentration of solids because of very low moisture content (Ambrose and Sreenarayanan, 1998). Ascorbic acid decreased from its initial value of 14.21 mg/100 g (**Table 1**) in different treatments after dehydration. However, significantly higher ascorbic acid content of 5.5 mg/100 g was recorded in treatment T_6 (65 ºC) followed by 4.96 mg/100 g in T_5 (60 ºC) and the lowest ascorbic acid content was in treatment T_1 (2.97 mg/100 g). There was a significant decrease in the ascorbic acid content after 4 months of storage with significantly higher ascorbic acid content (3.74 mg/ 100 g) and less losses of ascorbic acid (32 %) in treatment T_6 (65 ºC). Ascorbic acid content was lowest in T_1 (1.89 mg/ 100 g) followed by T_2 (1.96 mg/ 100 g) in that order for all treatments with respective percentage losses of 36.37, 37.18, 38.36, 37.83 and 38.51 %. Volatile oil percentage was found to decrease from its initial value of 0.10% (**Table 1**) in different treatments after dehydration with significantly higher volatile oil percentage of 0.323 % in treatment T_6 (65 ºC) and lowest volatile oil percentage of 0.130 % recorded in treatment T_1 (40 ºC). However, treatments T_2 (45 ºC) and T_3 (50 ºC) and T_4 (55 ºC) and T_5 (60 ºC) were statistically at par with respect to volatile oil percentage values. Volatile oil percentage decreased after 4 months of storage with significantly higher volatile oil percentage with less losses of 7.13 % recorded in treatment T_6 (65 ºC) in comparison to other treatments. The lowest volatile oil percentage of 0.080 was recorded in treatment T_1 (40 ºC) with corresponding loss of 38.47 % when compared to other

Table 3 Biochemical constituents of garlic slices after dehydration and after storage period of four months

Treatment	Carbohydrate content, g/100g		Loss (%)	Proteins, g/100g		Loss (%)	Ascorbic acid, mg/100g		Loss (%)	Volatile oil, %		Loss (%)	Browning index	
(Storage period) months	0	4		0	4		0	4		0	4		0	4
T_1 (40 ºC)	57.79	52.75	8.73	13.51	9.51	29.61	2.97	1.89	36.37	0.130	0.080	38.47	0.089	0.210
T_2 (45 ºC)	61.63	56.66	8.12	13.65	9.66	29.24	3.12	1.96	37.18	0.176	0.136	27.73	0.079	0.153
T_3 (50 ºC)	62.39	58.40	6.4	14.39	10.39	27.80	3.65	2.25	38.36	0.180	0.140	22.23	0.072	0.120
T_4 (55 ºC)	64.54	61.51	4.7	14.55	10.52	27.70	4.23	2.63	37.83	0.213	0.180	15.50	0.064	0.086
T_5 (60 ºC)	65.10	62.43	4.11	14.47	10.47	27.65	4.96	3.05	38.51	0.233	0.210	9.88	0.056	0.087
T_6 (65 ºC)	69.06	67.12	2.81	14.85	11.21	24.52	5.50	3.74	32.00	0.323	0.300	7.13	0.045	0.065
SE (m ±)	1.0360	1.1733	-	0.3222	0.3891	-	0.1711	0.2436	-	0.0285	0.0276	-	0.0039	0.0145
CD (p = 0.05)	2.2573	2.5564	-	0.7020	0.8478	-	0.3727	0.5307	-	0.0622	0.0600	-	0.0085	0.0317

treatments. It appeared from **Table 3** that ascorbic acid content and volatile oil percentage decreased with decrease in the dehydration temperature from 65 ºC (T_6) to 40 ºC (T_1). This could be attributed to the fact that higher temperature facilitated rapid dehydration, and reduced the period of exposure to elevated temperatures resulting in less deterioration of ascorbic acid and consequently higher retention (Desrosier and Desrosier, 1977). Losses in volatile oil percentage could be attributed to enzymatic action initiated by cell wall collapse during slicing and also due to losses during drying (Ambrose and Sreenarayanan, 1998). Browning index values from **Table 3** revealed significantly less browning in treatment T_6 (65 ºC) with a browning index value of 0.045 followed by 0.056 in treatment T_5 (60 ºC). Quality in terms of browning index was poor in treatment T_1 with a browning index value of 0.089 followed by T_2 (0.079), T_3 (0.072), T_4 (0.064) and T_5 (0.056) in that order. Dehydrated garlic slices were comparatively quite acceptable in terms of browning index values after 4 months of storage when dried at a temperature level of 65 ºC (T_6) in comparison to other treatments where browning index scores were comparatively higher as shown in **Table 3**. This could be attributed to browning because of moisture mediated deteriorative reactions due to increase in the moisture content after a storage period of 4 months.

Sensory Quality

Sensory quality evaluation scores, as evident from **Table 4**, revealed that desirable colour, flavour, texture and over all acceptability (score rating 8) was obtained in T_6 treatment (65 ºC) followed by T_5 (score rating 7) and treatment T_4 (score rating 7). After 4 months of storage period organoleptic quality deteriorated in all the treatments except treatment T_6 (65 ºC) where over all acceptability score was 7. Sensory score for colour immediately after dehydration revealed excellent quality in T_6 (score 9) and even after a storage period of 4 months (score 8). Texture was good in T_1 (40 ºC), and T_3 (50 ºC), perhaps because of drying at comparatively lower temperature (40, 45 and 50 ºC, respectively). Thus, it can be concluded from the present investigation that treatment T_6 (65 ºC) was quite acceptable in terms of maintaining the overall physical and biochemical attributes of garlic slices. Next preference could go to drying at a temperature level of 60 ºC (T_5). Drying at lower temperature should be discarded in order to prevent the quality deterioration from setting in and to maintain the storage stability of dehydrated garlic slices.

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Table 4 Sensory evaluation of dehydrated garlic after dehydration and after a storage period of four months

Treatment	Color		Flavour		Texture		Over all acceptability	
(storage period) Months	0	4	0	4	0	4	0	4
T_1 (40 ºC)	5	3	7	5	8	7	6	5
T_2 (45 ºC)	6	4	7	5	8	7	6	5
T_3 (50 ºC)	6	4	7	5	8	6	6	5
T_4 (55 ºC)	7	6	7	6	7	6	7	6
T_5 (60 ºC)	8	7	7	6	7	6	7	6
T_6 (65 ºC)	9	8	7	7	7	6	8	7

Score rating = 9 point hedonic Scale, 9 = Excellent, 1 = Extremely Dislike

Design, Development and Performance Evaluation of Radial Arm Type Cashew Nut Sheller

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Abstract

A radial arm type cashew nut sheller (RCS) was developed to ease shelling operation in the cashew nut processing industries. The basic principle of cutting and shearing was employed to extract the edible kernel from a steam boiled cashew nut in a single operation. The developed sheller could be operated in a sitting posture reducing the operator drudgery experienced in the existing type sheller in Indian cashew nut processing industries. The operational capacity and the quantitative efficiency of the developed sheller was 9.30 kg hr^{-1} and 87.52 %, respectively. The qualitative efficiency was determined in terms of whole kernel recovery as it fetched premium price. The ratio between whole to broken kernels was 9.84, which was 2.04 times more than the hand cum pedal operated sheller (HPS) for steam treated cashew nuts. The cost of the sheller was Rs. 2,050, which was15 % more than HPS.

Introduction

The engineering phase of processing cashew nuts deals with the process involved in changing or altering the raw cashew nut by the application of various unit operations and using the machinery to bring about the desired changes. The multi-operations can be briefly classified as conditioning, shelling, drying, peeling and grading. In India, cashew has been a major source of foreign exchange. About 0.01 million tones of cashew kernels were exported during the year 2007-08, earning a foreign exchange of INR 22.89 billion. The raw material wealth is a positive factor for promoting cashew processing industry in this country. Over 3,650 processing factories, which included 1850 tiny scale processing units, constituted the cashew nut processing sector that employed more than 0.45 million workers in India (Anonymous, 2008).

Cashew nut processing is essentially an operation of materials separation, designed to recover from the raw cashew nut the edible kernel and cashew nutshell liquid (CNSL) contained in the tissues of the middle shell layer. In its natural state, the shell is pliable and unsuitable for any type of manual or mechanical opening. Heating, however, hardens and makes cashew nuts brittle and susceptible for cracking or splitting. Opening the cashew nuts has traditionally been achieved by an impact on the brittle shell, which ideally splits into two halves, setting the kernel free. Considerable skill is required for cracking the cashew nuts with a wooden mallet without damaging the kernel. The output of the whole kernel by an experienced person is about 70 to 85 percent (Ohler, 1966). The impact approach was mechanized during the fifties through the use of a centrifugal impeller, running within and hurling the cashew nuts against the inside wall of a metal cylinder. An alternative way to open the brittle cashew nuts is to clamp it between two converging blades along its seam and pry open by lifting one of the

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blades.

Various decorticating methods such as explosive decompression of the cashew nuts by which the kernel emerges through a natural point of weakness of the shells; freezing the cashew nuts and breaking by a standard centrifugal nutcracker and cutting the shell around its major axis are employed to allow the kernel to be exposed. All these methods have the problem of the cashew nutshell liquid affecting a portion of the kernel due to low whole kernel recovery (Hall and Banks, 1965).

Hand operated cashew shelling units are used in Thailand. A lever is used to lower the upper blade to cut the raw cashew nut placed at the bottom and also to pry open the cut shells. The kernels are then extracted using a pin manually. Due to strenuous hand cutting, it has changed to pedal operated system (Matthew, 1995).

A survey of the need of cashew nut farmers and manufacturers for appropriate shelling machinery indicated definite needs for both the manually operated and power driven sheller. Design considerations on the manually operated sheller utilized the new principle of press twist actions of the sheller blade resulting in two versions. A manually operated sheller and a semi-automatic sheller (Thivavarnvongs *et al*., 1995).

The single cashew nut cutter developed by Mechanical Engineering Research and Development Organisation (MERADO) to shell the roasted cashew nuts (Anonymous, 1997) separated the kernel by a hand cum foot-operated mechanism. In HPS (**Fig. 1**), the cashew nuts split open between two blades acting in opposite directions. Although this method of splitting is effective, a high degree of dexterity is required to manipulate the size of the cashew nut and the force applied on pedal to avoid kernel breakage and damage to the hands. Moreover, the method of operation develops fatigue in a short duration in addition to the health hazard.

Preparation of cashew kernels for trade and export is an intricate procedure compared to that of other edible nuts. Shelling has been the greatest problem in processing and the contributing factors are irregular shape of the cashew nut, the brittle nature of the kernel and the presence of the cashew nutshell liquid (CNSL). Considering the shortcomings such as operator's drudgery, lower qualitative efficiency and inferior production rate, an attempt has been made to develop a radial arm type cashew nut sheller.

Design of Radial Arm Type Sheller

The principal constraints in developing mechanical devices for shelling cashew nuts are the tough and elastic nature of the outer shell wall and the peculiar shape of the cashew nut whose curvature of the sides varies considerably with individual cashew nuts. Avoiding the contamination of the kernel by the corrosive CNSL and minimizing kernel breakage are the two major considerations while shelling cashew nuts.

The single operation that enables cashew nut holding and subsequent splitting was achieved by providing a 1) Radial type cashew nut holding arm; 2) Radial type cashew nut splitting arm; 3) Twin blades; 4) Split blade link; 5) Drive assembly and 6) cashew nut size adjuster. The design details of radial arms, link assembly and blades are depicted

Fig. 1 Hand cum pedal operated cashewnut sheller (HPS)

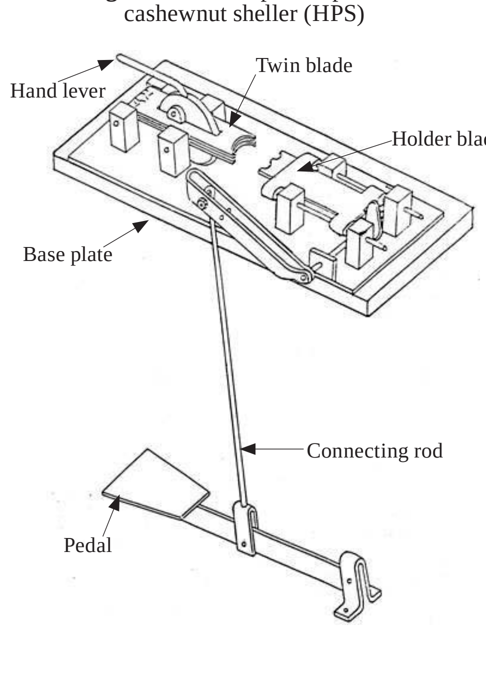
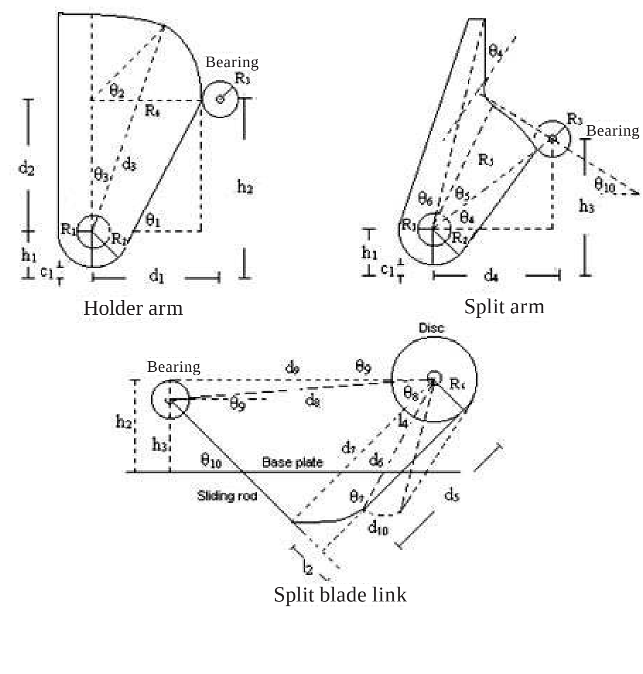


Fig. 2 Design of radial arms and split blade link of Raduak arm type cashwnut sheller



in **Fig 2**. Design of various components of the sheller is given below.

Design of Holder Arm

Angular position of holder arm

$\theta_1 = Tan^{-1} \times (h_2 - h_1) / (d_1 - R_2 - R_3) = 45^o$ ....(1)

Push angle in holder arm

$\theta_2 = (l_1 / R_5) \times (180 / \pi) = 46.5^o$ ....(2)

Slip angle of holder arm

$\theta_3 = (l_3 / R_4) \times (180 / \pi) = 23^o$ ....(3)

Design of Split-Arm

The slip surface of the radial arm becomes perpendicular to the line passing through the center of the bearing and sliding rod, while contacting the bearing for push. Both holder and split arm were fabricated with 5 mm thick mild steel plate.

Angular position of split- arm

$\theta_4 = Tan^{-1} \times (h_3 - h_1) / (d_4 - R_2 - R_3) = 30^o$ ....(4)

Slip angle of split-arm

$\theta_5 = 26^o$

Push angle of split-arm

θ_6 = graphically found as 31º

Design of Split Blade Link

The twin-split blades are coupled to a disc through a slot and pin, as shown in **Fig 2**, at an offset distance of 20 mm from the disc center. A radial movement of 12 mm in the disc enables the split blades to open the cashew nut effectively.

Angular position of link rod

$\theta_6 = Tan^{-1} \times (R_6) / (d_5^2) = 31.6^o$ ....(5)

Displacement in link rod

$l_3 = l_4 d_8 / R_6$

$l_3 = l_4 (d_7^2 + R_6^2)^{1/2} / R_6 = 23$ mm ....(6)

The connecting mechanism of slide and link rod was so adjusted that a linear displacement of 30 mm in sliding rod produced a radial movement of 23 mm in link rod.

Angular position of sliding rod

$= 90 - \theta_6 - \theta_8 = 27^o$

Design of Blades

High carbon steel sheet 1.5 mm thick was used to fabricate blades. The length of holder and split blades was decided considering the maximum value of major axis dimension of raw cashew nuts. The depth of curvature in blades was selected based on the average sphericity of a cashew nuts (Balasubramanian, 2001) and the bevel angle by trial and error method for better penetration of spongy cashew nuts and split open.

Length of holder blade =40 mm
Width of blade (assumed) = 20 mm
Depth of curvature in holder blade = 7 mm
Depth of curvature (split blade) = 7 mm
Bevel angle = 15º

Design of Drive Assembly

The angular position of the pedal was based on the height of base plate from the ground and the total angular movement required in holding and splitting arms. Height of the table was selected based on the average height of women labourers working with the shelling machine (i.e. 1,500 mm). Accordingly, the size of the pedal and its inclination were designed and connected to the drive shaft through connecting rod. The holder and split arms were coupled to the drive shaft at middle and extreme point. In order to feed cashew nuts on either side of split blades with ease, the lever arm length was restricted to the length of sliding plate, i.e. 160 mm.

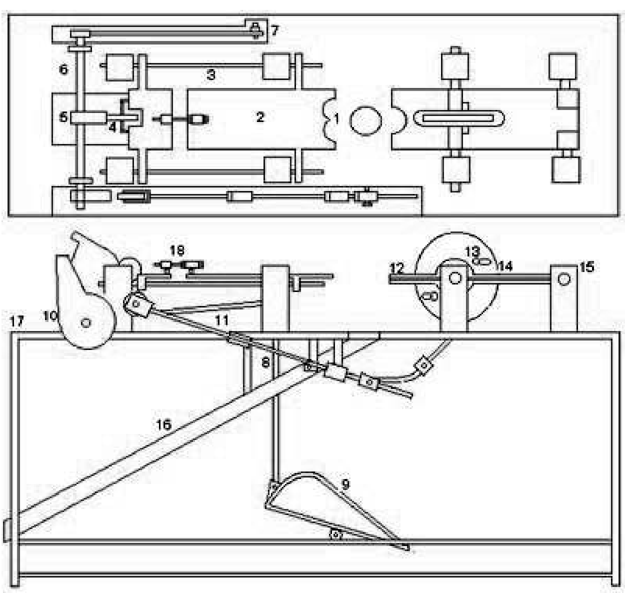
Design of Size Adjuster

Cashew nut width decides the distance between the holder and split blade. In order to adjust the movement according to the cashew nut size, i.e. width, bolt and lock cashew nut assembly was provided at the adjoining point of two sections of sliding plate. Loosening the cashew nut will increase the arm length to suit small size cashew nuts. The gap between the two sections of the sliding plate was kept at minimum for shelling larger size cashew nuts.

Development of Radial Arm Type Sheller

The radial arm type shelling unit for cashew nuts works on the basic

Fig. 3 Radial arm type casherwnut sheller



1. Holder blade
2. Sliding plate
3. Guide rod
4. Bearing
5. Holder arm
6. Driving shaft
7. Lever arm
8. Connecting rod
9. Pedal
10. Split arm
11. Sliding rod
12. Twin blade
13. Eccentric joint
14. Central disc
15. Support with hinge
16. Spout
17. Base plate
18. Cashewnut adjuster

Drawing not to scale

principle of cutting and shearing. In this method, a single operation enabled splitting of cashew nuts between two horizontally mounted blades especially shaped to suit the contour of the raw cashew nut. The major parts of sheller are classified in to i) Cashew nut holding assembly, ii) Cashew nut plitting mechanism and iii) Drive assembly.

The cashew nut holding assembly was designed to make a slit in to the concave or notched portion of the cashew nut. It was comprised of a sliding arm, guiding rod and compression springs. A blade was riveted at the end of the sliding plate and a bearing on the other end. This plate moved back and forth over a guide rod. Guide rods were held in position firmly with the help of supports. A compression spring, provided with the guide rods, ensured that the sliding plate would regain its original position after releasing the force applied on the holder arm.

The cashew nut splitting mechanism consisted of a twin-blade plate, vertical disc and disc actuating assembly. Twin-blade plates were hinged at one end with supports and a set of concave blades riveted at the opposite end. A central disc was positioned vertically inside the slot of twin-blade plates and rotated freely with the help of shaft provided at the centre. The central disc and twin-blade were eccentrically joined through cotter pins inserted at the top and bottom of the central disc. Radial movement in the central disc helped to move twin blades apart due to the hinge and eccentric joint to aid in splitting action. The sliding rod with bearing at one end transferred motion to central disc. This assembly converted the linear to radial movement in the central disc.

The swing type pedal provided at the bottom transferred motion to the drive shaft through a connecting rod and lever arm. Both holder and split arms were in contact with bearing while in action for smooth movement. The radial arm located at the centre of the shaft ensured a gentle push to the sliding plate until the riveted blade on the other end penetrated the placed cashew nut and slipped preventing the cashew nut holder assembly to return to original position. At this moment, the split arm contacted the sliding rod to actuate the disc to aid in opening the twin-blade and split the cashew nut.

Raw cashew nuts were placed between blades in such a way that the notched portion of the cashew nut was towards the holder blade and its convex side rested on the edges of the twin-blades. After positioning the cashew nuts properly, the sliding arm was actuated by pressing the pedal and the sliding plate was allowed to move forward to hold and penetrate the cashew nut. After pushing the sliding plate a set distance, the holder arm sliped and, exactly at this moment, the split arm coupled at the exterior end of drive shaft contacted the sliding rod and moved downwards to transfer circular movement to the central disc. Due to the eccentric joint in the central disc and split plates, outward movement of twin-blade was attained to split open the cashew nutshell.

Performance Evaluation

The radial arm type cashew nut sheller was evaluated for its performance and compared with the existing hand operated sheller. Raw cashew nuts were procured from local market and graded manually in to small (20-22 mm), medium (≤ 23-25 mm) and large (≥ 26 mm) based on the minor axis. The graded raw cashew nuts were conditioned in an industrial type steam boiler with a centralized steam supply system maintained at 4 kg m^{-2} for 16 min. The steam boiled cashew nuts were dried in a thin layer for 20 hrs to make the cashew nuts hard and brittle. The average temperature and relative humidity during this period was 26.3 ºC and 90.50 %, respectively.

About 5 kg of conditioned cashew nuts of various sizes (viz., small, medium, large) were shelled separately using two operators. The time taken for shelling the given quantity of conditioned cashew nuts was noted and the operational capacity of the sheller was calculated using **Eqn. 7**.

After shelling, the intact kernels in the shells were scooped out using a needle carefully. Then, whole and broken kernels were segregated to calculate the qualitative efficiency using **Eqn. 8**. The unshelled cashew nuts were sorted from shelled nuts and its quantitative efficiency calculated using **Eqn. 9**. The operator shelled 10 kg of steam conditioned cashew nuts continuously and the shelled cashew nuts were weighed for whole, broken kernels and unshelled cashew nuts after 10 min interval until the completion and the rate of production was calculated. The various processing parameters with respect to shelling were calculated using following equations.

Production capacity (kg hr^{-1}) = Quanity shelled (kg) / Time taken (hr)...............................(7)

WB ratio = Weight of whole kernels (kg) / Weight of broken kernels (kg) ....(8)

Quantitative efficiency (%) = 1 − Weight of unshelled cashewnuts (kg) / Weight of cashewnuts shelled (kg) × 100..................(9)

The data collected were analyzed statistically using a factorial design with split-plot approach.

Result and Discussion

The average operational capacity of RCS was determined as 9.30 kg hr^{-1}. Cashew nut size and the sheller operators were significant at the 5% level. The average qualitative efficiency of the developed shell, in terms of whole and broken kernels, was 9.84. The constant onward movement of the holder blade and split blade displacement enabled the penetration into the inner edge of the shell, protecting the kernel damage to a greater extent. As the economic efficiency of cashew nut processing depended on whole kernel recovery at the shelling level, increased whole kernel yield using RCS was meritorious. The statistical analysis indicated that the qualitative efficiency remained unchanged irrespective of the operators.

The overall quantitative efficiency, which is otherwise called shelling efficiency, was 87.52 % due to lack of operator experience with the newly developed shelling unit. An increasing trend was observed in all the processing parameters; viz., production capacity, WB ratio and quantitative efficiency during continuous operation. Sitting posture reduced the operator's drudgery, indicating that the operators can improve the performance after gaining experience.

NOTATIONS

CNSL	Cashewnut Shell Liquid
c_1	clearance between arm bottom and base plate, mm
DCCD	Directorate of Cashew and Cocoa Development
d_1	horizontal distance between holder arm and bearing centers, mm
d_2	vertical distance between the shaft centre to the perpendicular drawn to the intersection of holder arm and bearing, mm
d_3	radius of curvature for slip in holder arm, mm
d_4	horizontal distance between split arm and bearing centers, mm
d_5	length of link rod, mm
d_6	distance between the disc centre and link rod, mm
d_7	horizontal distance between the disc centre and the meeting point of sliding and link rods, mm
d_8	length between bearing and disc centers, mm
d_9	horizontal distance between the disc centre and the intersection of vertical line drawn from bearing centre, mm
d_{10}	displacement in link rod, mm
HPS	Hand cum pdal operated seller
h_1	height of drive shaft form base plate, mm ($R_2 + c_1$)
h_2	height of holder arm bearing from base plate, mm
h_3	height of sliding rod bearing from base plate, mm
l1	displacement in holder arm, mm
l_2	arc length of slip in holder arm, mm
l_2	displacement in sliding rod, mm
RCS	Radial arm type cashew nut seller
R_1	radius of drive shaft, mm
R_2	bottom radius of holder arm, mm
R_3	radius of holder arm bearing, mm
R_4	radius of arc (for push in holder arm), mm
R_5	radius of arc (for slip in split arm), mm
R_6	radius of offset disc, mm
θ_1	angular position of holder arm, degree
θ_2	push angle in holder arm, degree
θ_3	slip angle in holder arm, degree
θ_4	angle between line joining centers of arm and bearing, degree
θ_5	slip angle in split arm, degree
θ_6	push angle in split arm, degree
θ_7	angular position of link rod, degree
θ_8	angle between line connecting bearing, disc and link rod, degree
θ_9	angle between the line joining centers of bearing and disc with horizontal, degree
θ_{10}	angular position of sliding rod, degree

Production Capacity

The production capacity of RCS and HPS showed that both are non significant with each other with respect to operators. The overall capacity of both the shellers was 9.30 Kg hr^{-1} (**Fig. 4**). Although operators were not experienced with the newly developed sheller, achievement of same production capacity with RCS over the existing unit indicated a positive sign of improving individual performance. Both types of shellers were found to be significant with respect to size of the cashew nut at the 5% level. The small (20-22 mm), medium (≤ 23-25 mm) and large (≥ 26 mm) size cashew nuts ranged from 180-190, 150-160 and 120-130 cashew nuts per kg, respectively. Therefore, it was obvious that the capacity of the sheller was highly influenced by the size of the cashew nut. The simple lock nut and bolt assembly to adjust the distance between the holder and splitting blade required less skill.

Wholes to Broken Kernel Ratio (W/B)

The average qualitative efficiency for RCS and HPS was 9.83 and 4.69, respectively (**Fig 5**). In the existing sheller, the operator had to manipulate the size of the cashew nut and apply required pressure to extract the whole kernel. But in the radial arm type, the constant movement of holder and splitting blades ensured the higher whole kernel yield. Only proper positioning of the cashew nut against the blades was essential in this type of sheller. The holder and split blade design was based on the contour of the cashew nut and the kernel, respectively. Penetration of blades took place according to its design and the position of cashew nut. Although interaction effect with respect of operators was non significant, qualitative efficiency with respect to the size of the cashew nut showed significant results. Variation in shell thickness with respect to size of the cashew nut was the reason for the variation in qualitative efficiency.

Quantitative / Shelling Efficiency

It is evident from **Fig. 6** that the quantitative efficiency of both RCS and HPS units are non significant with respect to operator and size. The difference in overall efficiency between the two shellers is due to the lack of experience of the operators with the newly developed unit.

Rate of Production

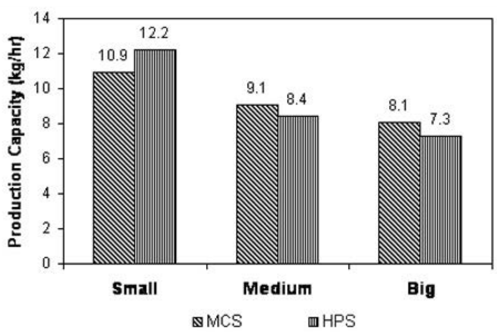
The comparative rate of production of both shellers is given in **Fig 7**. An increasing trend in the shelling rate was observed in the case of RCS. The cashew nut feeding and pedal operation to hold and split open needed to follow a uniform time interval for better production rate. Although rate of production of RCS was lower in the beginning, simple pedal operation and feeding the cashew nut made the operators to get adjusted with the shelling unit in a shorter period and resulted in higher production. But, in the case of PS, the operators showed better performance in the beginning and, as the time elapsed, a declining production rate was observed. The continuous operation of pedal according to the size of the cashew nut and lifting the lever to open the cashew nut led to fatigue in a shorter period. Based on the degree of fatigue, the individual performance declined with respect to time.

Conclusions

A radial arm type sheller for cashew nuts was developed by introducing low cost automation to obtain higher whole kernel recovery in the shelling operation for better economic results. Besides, drudgery experienced in HPS was totally eliminated while operating with newly designed radial arm type sheller. Moreover, the sheller design accommodated cashew nut feeding from both sides, indicated a positive sign of increasing production capacity.

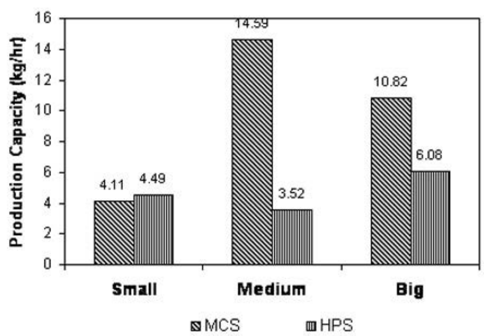
The gap between the two blades could be adjusted according to the size of the cashew nuts. In order to avoid constant readjustments, the raw cashew nuts were graded based on minor axis dimension and cashew nuts of similar size could be taken at a time for shelling.

Fig 4. Comparison of production capacity of RCS and HPS



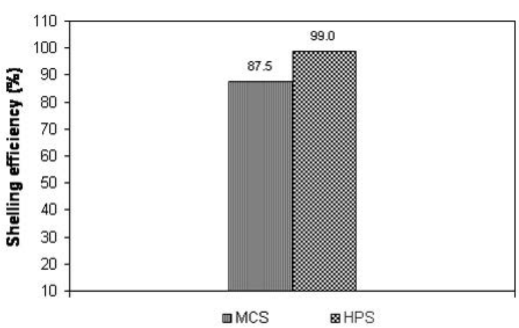
CD for operator = 0.74 CD for Sheller type = NS
CD for nut size = 1.12

Fig 5. Comparison of whole to broken kernel ratio of RCS and HPS



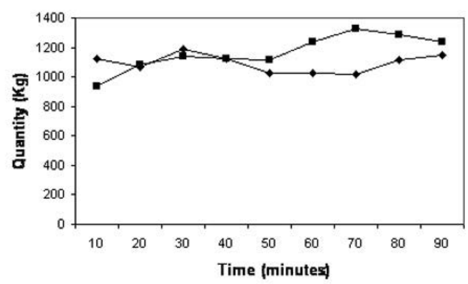
CD for operator = NS CD for Sheller type = 3.03
CD for nut size = 3.23

Fig. 6 Comparison of production efficiency of RCS and HPS



CD for operator = NS CD for Sheller type = NS
CD for nut size = NS

Fig. 7 Comparative performance of rate of production in RCS and HPS



—■— Modified cashew nut sheller —◆— Hand cum pedal operated

As cutting with a mechanical aid showed considerable improvement in yield of whole kernels, an attempt can be made to develop a high performance sheller by the introduction of mechanical feeding of cashew nuts and synchronized power operation of the sheller.

At this juncture, industry has to look for less elaborate processing equipment having a modest capacity, which need not necessarily be fully automatic. The serious obstacle to mechanization of cashew nut factories in India is the problem of the redundant labour force. Therefore, change over to mechanization of the industry in India, will have to be cautiously and gradually attempted on a long-term basis.

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NEWS

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Design and Development of a Weeder for both Lowland and Upland Conditions



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Abstract

An experiment was conducted to design and develop a weeder suitable for both lowland and upland conditions for a row crop field under the research plan of 2007-2008 of the Farm Machinery and Postharvest Technology (FMPHT) division, Bangladesh Rice Research Institure (BRRI). Engineering design of the weeder used the AutoCAD computer program with the BRRI recommended spacing of planting, ease of operation, repair and maintenance and locally available materials. A prototype was fabricated according to the design mentioned above in the FMPHT divisional research workshop and was performance tested in the research field of Bangladesh Rice Research Institute (BRRI) and Bangladesh Agricultural Research Institute (BARI), Gazipur. The size and weight of the weeder were 170 × 12 × 22 cm and 5.5 kg, respectively. Two driving wheels and skids were used in the weeder to facilitate the operation in upland and lowland conditions, respectively. There was an adjustment lever with five options to control the height of the handle according to the operators height (76-152 cm). The horizontal component of force that facilitated the forward motion of the weeder during operation decreased from 0.875 R to 0.777 R with the increase of handle height in adjacent option (Where R was the applied force of the operator). The lower option was more comfortable for operation of the weeder in the field. A total of 4 hoes were used as weeding tools in different degrees of inclination to form a horizontal line to avoid clogging. In lowland conditions, the weeder was suitable in the field with 1 cm or below standing water. In upland conditions, it was suitable with row planted vegetables, pulse, nut and aus rice field. During field operation, the average effective field capacity in lowland and upland was about 8 decimal/hr and 6 decimal/hr, respectively. The average degree of weeding was calculated to be about 90 % and 83 % in the lowland and upland, respectively. Farmers had the benefit Tk 141 per hectare of land using the single weeder developed for both conditions over two weeders used in separate conditions.

Background

Rice is the staple food for nearly half of the world's population, most of whom live in Asia (Sattar, 1994). However, rice is the main source of energy and cash income for most of the farmers in Bangladesh (BARC, 1983). Asia, as a whole, contributes to about 92 percent of the world's rice harvests. Bangladesh ranks fourth in the world both in area and production (FAO, 1994) and 39th in yield among the rice growing countries (IRRI, 1995). In Bangladesh, Aman rice and Boro rice cover about 52.76 % and 35.70 % of total rice area and they contribute to 44.13 % and 48.53 % of the total rice production respectively (BBS, 2004). But, the crop yield is much lower than that of transplanted rice in other rice growing countries of the world. Severe weed infestation constitutes one of the reasons for such a low yield (Mamun, 1981). The yield loss due to weed competition in Aman rice is 40 % in Bangladesh (BRRI, 1991).

In Bangladesh, weeds are being controlled manually by hand pulling or by using simple tools like niranee, Japanese rice weeder, BRRI and BARI developed weeder. Usually, two to three hand weedings are done for growing a transplanted rice crop, depending upon the nature of weeds and their intensity of infestation. But, these methods are laborious, less comfortable, time consuming, expensive and usable only for either lowland or for upland filed.

In Bangladesh manual weeders were introduced from Japan during the sixties. Comilla Co-operative Karkhana first introduced the Japanese type push weeder (Islam *et al*., 2009). These Japanese weeders are being used by the farmers at limited areas because they are laborious to operate in heavy soils. Female workers cannot operate the Japanese weeders. Therefore, FMPHT division of BRRI developed the BRRI weeder and BRRI Kishan weeder. The BRRI weeder and BRRI Kishan weeder are only suitable for lowland rice fields. However, the BARI weeder is suitable to operate in upland conditions. As a result, farmers have to buy both lowland and upland weeders for different operations. Effects to suppress the weed infestation with simultaneous increase in crop production through improved cultivation requires the introduction of an improved weeder that will be suitable to operate in both conditions of the field. Considering the present problem, this experiment was conducted with the following objectives:

Objectives

- To design and develop a weeder suitable for weeding both in lowland and upland fields
- To study the performance of the weeder both in upland and lowland conditions

Materials and Method

GI pipe, GI sheet, MS sheet, MS flat bar and MS shaft were used to fabricate the weeder. Engineering design was done with the help of an AutoCAD computer program and a prototype was fabricated according to design in the FMPHT divisional research workshop. The fabricated weeder was tested in the BRRI and BARI field in lowland and upland conditions and data were collected as per the following design:

Collection of data

The following data were collected to calculate the walking speed (Km/hr), weeding efficiency (%), plant damage (%) and field capacity (ha/hr):

- Time required to travel 100 m distance in the field during weeding.
- Number of weeds in 1 m^2 area of the land before weeding and numbers of weeds in a similar area after weeding.
- Number of plants in 1 m^2 area of the land before weeding and numbers of damage free plants in asimilar area after weeding.
- Weeding time including losses in time and area of weeding in decimal.

Design considerations of the weeder

- The weeder should be simple and easy in operation and maintenance
- It should be maneuverable both in lowland and upland conditions of the field with row plated crops.
- Locally available materials should be used to minimize the fabrication cost.
- Weeding capacity should be acceptable in comparison with the presently available weeder.
- During operation, it should be

Fig. 1 Top view of wheels and hoe arrangement of the weeder

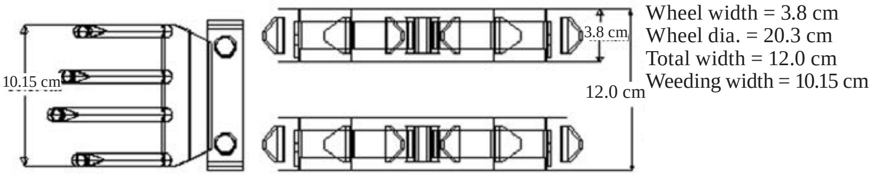
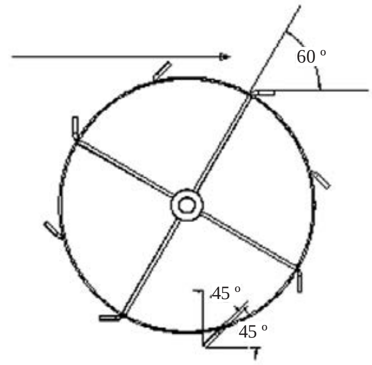


Fig. 2 Spike arrangement on the top of the weeder



comfortable and trouble free.

Cost Analysis

The cost of operation of the developed weeder was estimated following the RNAM (1995) recommendations to find out an over all financial benefit per unit land of the single weeder developed for both conditions over two weeders used in separate conditions so that the user could get a clear picture of the benefit of the developed weeder.

Details of the Design

Width of the Weeder

The width of the weeder was selected based on the row spacing (line to line distance) of transplanted and drum seeded rice and other crops. Overall width of the weeder was 12 cm (**Fig. 1**). The hoe was used as the weeding tool.

Weeding width depended on hoe arrangement and it was 10.15 cm to minimize the crop damage (**Fig. 1**). The width of the latest version of BRRI weeder was 13 cm that was suited to operate in the field sown by a drum seeder (Islam *et al*., 2009).

Spike Arrangement

The wheels of the weeder were also designed as weeding tools because of their spike arrangement on the top of the wheel rim.

A total of 8 spikes were used on the top of each wheel at 60 degree inclination from vertical toward the motion that grips the soil for weed-out during operation (**Fig. 2**). Hence, it made a 45 degree angle during contact in the soil and gradually decreased to 30 degrees with spike penetration in the soil. The spike of the weeder also made more traction in the upland soil.

Handle Size with Adjustment Lever

Length of the handle (132.08 cm) and height from ground level were directly related with the force required to operate the weeder and comfort of the operator.

There was an adjustment lever to control the height of the handle from ground level with five options. The height of the handle from ground level at lowest option of the adjustment lever was 76.2 cm and at the highest option was 152.4 cm (**Fig. 3**).

Hoe and Hoe Arrangement

The hoe was used as the weeding tool of the weeder. With a total of 4 hoes, length of outer two was 11.43 cm, The length of the inner one was12.7 cm and the length of the other one was 13.97 cm. They were used in the weeder at different elevations. The outer two hoes were arranged at 44 degree inclination from the horizontal and inner two hoes were at 42 degrees inclination(**Fig. 4**).

Different angle of inclination was used to avoid the clogging of the weeded-out weeds. Normally, 15-25 degree tilt angle was suitable for easy penetration since penetration was a problem in dry land weeding (Bainer *et al*., 1995). Tilt angle of the apex point of the outer and inner hoe were 25 and 23 degrees, respectively. Tilt angle and inclined angle of the hoe changed with the handle adjustment.

Fig.3 Handle of the weeder with adjustment lever

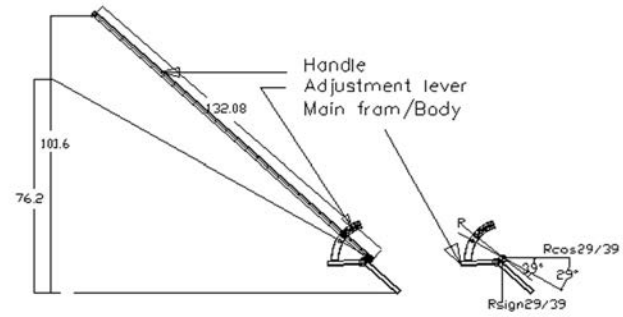


Fig.4 Top and side view of the hoe arrangement of the weeder

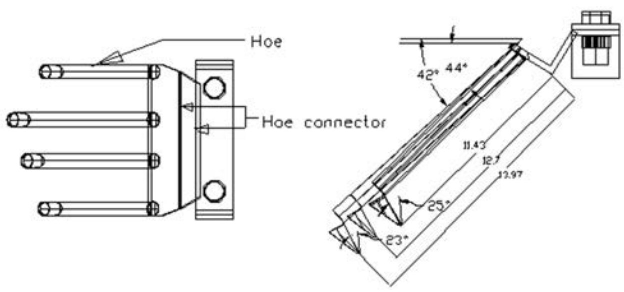


Fig.5 Top and side view of the skid

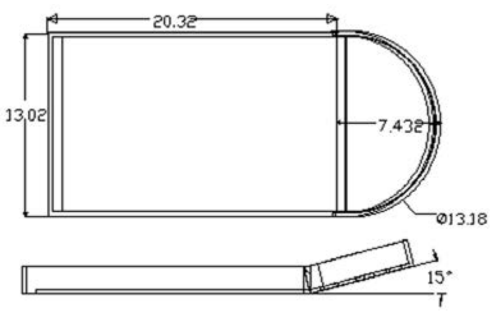
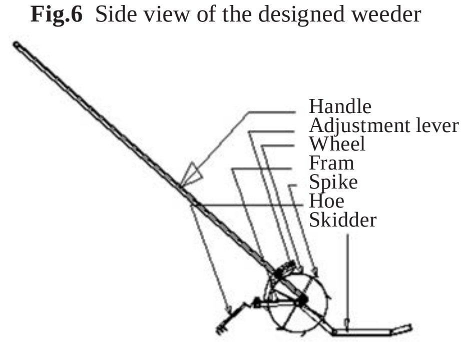


Fig.6 Side view of the designed weeder



Skid of the Weeder

A skid of the weeder was used to operate the weeder in the lowland conditions. It helped protect the weeder from penetration into the soil.

A 15 degree skidding angle was used in the apex of the skid(**Fig. 5**). The size of the skid was designed based on cone penetration resistance of the soil. Cone penetration resistance varied with depth, soil type and moisture content of the soil. The minimum and maximum cone penetration resistance was 168 kN/m^2 and 5,605 kN/m^2, respectively, of Gazipur soil (Afzal, 2008). The size of the skidder was 265 cm^2 (length 20.32 cm and width 13.02 cm). Considering minimum cone penetration resistance (168 kN/m^2) a 265 cm^2 size skidder can resist 4.452 kN (1 kN=9.81 Kg) force during operation.

Weeder Operation

Adjustment of the hoe was important for efficient weeding. It varied with operator height and field conditions and could be adjusted accordingly. On the other hand, adjustment of the handle height was also important considering the height of the operator, force requirement and comfort of operation. Height of the handle from ground level was increased by 25 cm with the change of one adjacent option of the adjustment lever (**Fig. 3**). Angle of the direction of resultant force from a horizontal line was also increased from 29 to 39 degrees with the change of one adjacent option that reduced the component of horizontal force from 0.875 R to 0.777 R and increased the downward vertical component of force from 0.485 R to 0.629 R (**Fig. 3**). As a result, the weeder showed penetration tendency rather than

Fig. 7a Photographic view of the fabricated weeder

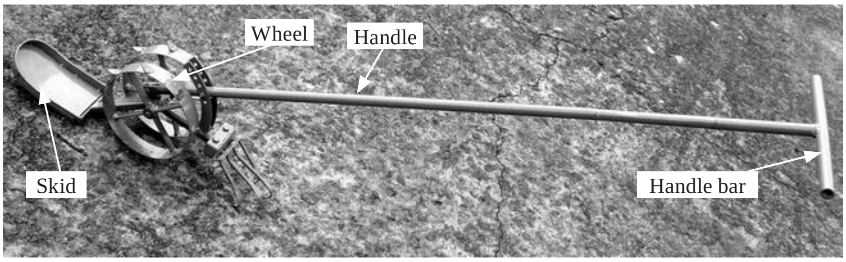


Fig. 7b Side view of the fabricated weeder

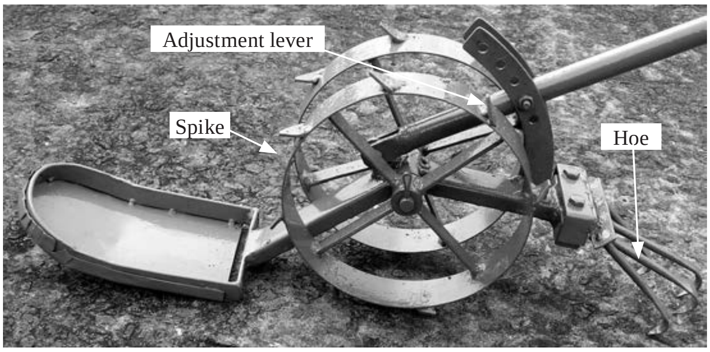


Table 1 Details specification of the weeder

Name of the components	Number	Size (cm)	Raw materials used
Handle	01	Length: 132.080 and Dia: 1,270	1,270 cm dia. G.I. pipe
Handle bar	01	Length: 35.560 and Dia: 1,270	1,270 cm dia. G.I. pipe
Wheel	02	Width: 3.810, Dia: 20.320 and Thickness: 0.159	20 gauge G.I. sheet
Wheel to wheel distance	-	Out side to out side: 11.430	-
Wheel spoke per wheel	04	Length: 8,890, Width: 1,905 & Thickness: 0.318	0.318 cm M.S. flat bar
Spike per wheel	08	Base width: 2,540, Top width: 0.635, Height: 1,905 & thickness: 0.318	0.318 cm M.S. flat bar
Spike to spike distance	-	Linear: 8.001	-
Spike angle	-	60 degree toward motion from vertical line	
Main axle	01	Length: 10.478, Dia of taped end: 1.270 & Dia of middle part: 2.540	M.S. Shaft
Bush	02	Length: 2.540, Inner dia: 1.270 and Outer dia:2.540	M.S. Shaft
Hoe connector	01	2.540 angle bar, 10.160 length with 1.270 reduce part	M.S. Shaft
Hoe arrangement for dry land and wet land	04	0.953 cm dia four rod of 11.430, 12.70, 13.970 & 11.430 cm length. Top 2.159 inch 100 degree curved	0.318 cm M.S. flat bar
Depth controller guard	01	Aligned length: 14.275, Width: 2.540, Thickness: 0.318 & Curved: 35 degree	0.318 cm M.S. flat bar
Skidder	01	Width: 12.70, Length: 20.320, Front 7.620 cm, 15 degree up from base line	0.318 cm M.S. flat bar

forward motion with the increase of handle height from ground level. Therefore, minimum operational force was required to operate the weeder at the lowest option of the adjustment lever (**Fig. 6**).

Results and Discussion

Fabrication of the Weeder

As per design, the weeder was fabricated in the FMPHT divisional research workshop. Different views of the fabricated weeder are given below on **Fig. 7a** and **7b**).

Detail Specifications of the Weeder

The weeder was designed considering the conditions of upland and lowland. For lowland conditions, the skidder was essential for operation of the weeder in the field with a simple and thin hoe. On the other hand, the wheel was essential to operate the weeder in upland conditions with a thick hoe. Hence, the wheel and skidder were incorporated for suitability in both types of soil. The components of the weeder are given in **Table 1** with specifications and materials that were used in fabrication.

Field Performance of the Weeder

The field performance of the weeder was tested in both lowland and upland conditions and data were collected. The performance of the weeder in wet land conditions and upland conditions is given in **Table 2**. In lowland conditions, trials were conducted after 25 days of transplanting of BRRI dhan 29 in Boro season in 2007. Trial 1 was conducted in clay type muddy soil with light weed infestation and trials 2, 3 and 4 were conducted in clay type muddy soil with medium weed infestation in the research plots of BRRI. But in upland conditions, trials 1 and 4 were conducted in the pulse research plot in clay-loam soil with light and heavy weed infestation and trial 2 and 3 were conducted in the nut research plot of BARI in the same type of soil with medium weed infestation.

Operation in Up Land Conditions

Operation in Low Land Conditions

Field capacity and degree of weeding varied with the variation of weed infestation, type of weed, type of soil condition and time of weeding. On the contrary, number of plants damaged varied with the variation of plant density and spacing of planting or transplanting. In lowland conditions, field capacity (9.2 decimal/hr) and degree of weeding (91.48 %) was more in trial 1 because of light weed infestation. The average effective field capacity, degree of weeding, plant damage and walking speed were around 8.3 decimal/hr, 90.57 %, 1.22 % and

Table 2 Field performance of the weeder in lowland and upland conditions

Items	Trial 1	Trial 2	Trial 3	Trial 4	Average
Lowland conditions					
Time of operation, min (t)	98	118	108	111	108.75
Area of weeding, decimal (a)	15	15	15	15	15
Field capacity, decimal/hr (a × 60/t)	9.2	7.6	8.3	8.1	8.3
No. of weed in 1 m^2 before weeding (w_1)	987	1,214	1,320	1,276	1,199
No. of weed in 1 m^2 after weeding (w_2)	84	124	127	118	113
Degree of weeding, % ($w_1 - w_2 / w_2 \times 100$)	91.48	89.76	90.38	90.75	90.57
No. of plant in 1 m^2 before weeding (p_1)	187	169	154	143	163
No. of damage free plant in 1 m^2 after weeding (p_2)	184	167	153	140	161
Plant damage, % ($p_1 - p_2 / p_2 \times 100$)	1.6	1.18	0.65	2.10	1.22
Time to travel 100 m distance in min.(t)	2.83	2.86	2.84	2.94	2.87
Waking speed (km/hr = 6/t)	2.12	2.10	2.11	2.04	2.10
Upland conditions					
Time of operation, min (t)	158	94	91	152	123.75
Area of weeding, decimal (a)	20	10	10	15	13.75
Field capacity, decimal/hr (a × 60/t)	7.6	6.4	6.6	5.9	6.6
No. of weed in 1 m^2 before weeding (w_1)	184	217	255	298	238
No. of weed in 1 m^2 after weeding (w_2)	28	37	51	40	39
Degree of deeding, % ($w_1 - w_2 / w_2 \times 100$)	84.78	82.95	80.00	86.57	83.61
No. of plant in 1 m^2 before weeding (p_1)	108	58	62	98	81.5
No. of damage free plant in 1 m^2 after weeding (p_2)	106	57	62	98	80.75
Plant damage, % ($p_1 - p_2 / p_2 \times 100$)	1.85	1.72	0.00	0.00	0.92
Time to travel 100 m distance in min (t*)	2.51	3.42	3.40	3.95	3.32
Waking speed, km/hr (6/t*)	2.39	1.75	1.76	1.52	1.81

2.10 km/hr, respectively, in wetland conditions. But in upland conditions, the average effective field capacity, degree of weeding, plant damage and walking speed were around 6.6 decimal/hr, 83.61 %, 0.92 % and 1.81 km/hr, respectively. Islam and Haque (1985) reported that the field capacity of the indigenous weeders were 9.98 decimal/hr at weeding density of 250 weeds/m^2.

Cost Analysis

The price of the weeder varies greatly with the quality of M.S pipe, M.S flat bar, M.S. sheet, plain sheet and nut-bolt. The present price of the weeder is about Tk. 600. Comparative cost analysis of the developed weeder, presently used lowland weeder and upland weeder is given below on **Table 3**.

It was concluded that the user gets the benefit of Tk 141 per hector of land using the single weeder developed for both conditions over two weeders used in separate conditions.

Conclusion

Farmers can use this weeder in both lowland and upland conditions. The average effective field capacity in lowland and upland is 8.3 decimal/hr and 6.6 decimal/hr, respectively. This type of weeder is suitable for wet land conditions with minimum standing water. However in dry conditions, it is suitable with row planted vegetables, pulse, nut and rice. Farmers get the benefit of Tk 141 per hectare of land using the single weeder developed for both

Fig. 8a Operation in up land condition

Fig. 8b Operation in low land condition

Table 3 Comparative cost analysis

Items	Both lowland and upland weeder		Lowland weeder	Upland weeder
	Lowland use	Upland weeder		
Purchase price (Tk)	600	-	500	450
Service life (yr)	5	5	5	5
Capacity (decimal/hr)	8.3	6.6	9.98*	5.0**
Working day/yr	80	60	80	60
Area of weeding (ha/yr)	21.50	12.82	25.85	9.72
Depreciation, Interest, Tax and other cost (Tk/yr)	160	-	131	124
Labour wedge (Tk/yr)	12,000	9,000	12,000	9,000
Maintenance cost, Tk/yr, (3.5% of the purchase price)	21	-	18	16
Total cost, Tk/yr	12,181	9,000	12,149	9,140
Two time weeding cost by weeder (Tk/ha)	1,133	1,404	940	1,880
Two time weeding cost by labour (Tk/ha)	3,705	4,014	3,705	4,014
Benefit due to weeder use (Tk/ha)	2,572	2,610	2,765	2,134
Average benefit considering lowland and upland weeding (Tk/ha)	2,591		2,450	
Benefit of developed weeder over two weeders (Tk/ha)	2,591 − 2,450 = 141			

N.B: Labour price = Tk. 200/man-day, *Islam and Haque (1985) and **BARI (2006)
a) Annual depreciation, D = P − S / L, Where, D = depreciation, Tk/yr, P = purchase price of the thresher, Tk
S = salvage value, Tk, L = Working life of the threshers, yr
b) Interest on Investment, I = P + S / 2 *xi*, Where, i = rate of interest
c) Tax, insurance and shelter cost, T = 3 % of purchase price.

conditions over the two weeders used in separate conditions.

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Development and Laboratory Performance of an Electronically Controlled Metering Mechanism for Okra Seed

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Abstract

Sowing is one of the most important operations in crop production that involves factors like correct seed rate, appropriate depth of placement and required seed spacing for maximizing crop yield. Mechanical or pneumatic seeders separate the seed and place them at the required spacing. However, they are bulky, require high energy and produce more vibration during operation and are expensive. An electronically controlled metering mechanism was developed with an attempt to make the drills/planters simpler without compromising precision in seed placement. It consisted of a proximity sensor, pulse generator, BCD counter (IC 4510), timer (IC 4093), relay unit, thumb wheel, DC motor and cup type seed metering unit. The electronic circuit controlled the operation of the DC motor which in turn rotated the metering unit to deliver the seed. Performance of the developed metering mechanism was evaluated in the laboratory using greased belt technique for okra seed with different levels of seed to seed spacing and forward speed of operation. The developed metering mechanism performed well and delivered the seed very close to the target seed to seed spacing with higher quality of feed index. The miss and multiple indexes were zero. Seed placement was more accurate at slower speeds and larger target seed to seed spacing.

Introduction

Agriculture is the mainstay of the Indian economy, contributing about 18.5 percent in GDP and supports about 115 million farm families. India contributes about 13 percent to the world's vegetable production and occupies first rank in Cauliflower production and second in Onion and Cabbage (Anonymous, 2008). Okra (Abelmoschus esculentus) is one of the major vegetable crops and is cultivated mainly for its immature fruits which are generally cooked as a vegetable. The total area under cultivation during the year 2006-07 was 0.38 million hectares. The annual production of okra is 3.68 million tonnes with an average productivity of 9.30 tonnes/ha. The crop is used in the paper industry as well as for extraction of fiber. The major okra producing states are Uttar Pradesh, Bihar, West Bengal, Orissa, Assam, Andhra Pradesh and Karnataka. It is grown during February-March, June-July and October-November. The seed rate varies from 10-12 kg/ha (rainy season) to 18-20 kg/ha (summer season). Plant spacing is kept lower (45 cm × 30 cm) for summer than for the rainy season (60 cm × 45 cm). The crop is generally sown manually by dibbling 3-4 seed per hole and, 2 weeks later, thinning is performed to maintain only 2 plants/hill (Kumar, 2009), which is quite a labor intensive operation.

Sowing equipment plays an important role, especially in labor scarcity areas. The use of seed-cum-fertilizer drills, not only conserves energy but also saves about 20 percent of the seed and increases the yield by 15 percent through better placement of seed and more effective utilization of fertilizer (Bansal *et al.*, 1983). Plate (horizontal, vertical and inclined type) and pneumatic seed metering devices are more common on vegetable planters. These metering devices are driven from the lug wheel through a chain-sprocket arrangement. This drive arrangement requires regular maintenance besides making the drills bulky. Pneumatic planters, though very precise, are expensive, produce more vibration and consume very high energy for operation. To have more precise seed to seed spacing with greater accuracy and higher efficiency, an electronically controlled metering mechanism can prove to be a very good option.

Kocher *et al.* (1998) and Panning *et al.* (2000) used an opto-electronic sensor for measuring the seed spacing in the laboratory for seeds larger than 3 mm in diameter. The unit did not work for seeds smaller than 3 mm in diameter. Lan *et al.* (1999) used an opto-electronic sensor system to measure seed spacing uniformity with different types of seed. Raheman and Singh (2003) developed a sensor, based on a light interference technique for sensing the seed flow from the metering mechanisms of a seed drill and planter. The sensor included an infra-red emitter, a phototransistor, a voltage divider network, IC4033B and a seven segment display unit and was mounted on the seed delivery tube. The performance of the sensor was studied for wheat, mustard and maize seeds in the laboratory. The sensor successfully sensed the seed droppings for mustard and wheat seed with a maximum error of 18 percent. Error within 10 percent was found for maize seeds because of more time gap between two consecutive seed droppings.

In view of very limited work on electronic metering mechanism, it was decided to develop an electronic controlled DC motor driven cup type metering device to be used in planters for accurate placement of seed. This would also eliminate the mechanical power transmission system commonly used to drive the seed metering device from the lug wheel, thus, making the drills/planters simpler.

Materials and Method

Determination of Physical Properties

Physical properties of okra seed (variety- Parbani Kranti) namely size, shape, sphericity, thousand seed weight, bulk density, angle of repose and moisture content were determined using standard procedures. For estimating the size, one hundred seeds were taken randomly and their dimensions were measured along all the three major planes. The measurement was replicated five times and their means computed. Sphericity and shape were characterized using the seed dimensions. Test weight was determined for five random samples of one thousand seed using an electronic balance with 0.01 g sensitivity. Bulk density was determined taking the ratio of weight to volume of seed using a thousand cubic centimeter measuring cylinder. The angle of repose was also determined using standard technique.

Development of Metering Mechanism

An electronically controlled seed metering mechanism was developed that consisted of a cup type seed metering unit, circular ring with sensing surface and an electronic assembly to control the operation.

Seed Metering Unit

The seed metering unit (**Fig. 1**) consisted of a mild steel circular plate 64 mm diameter with 8 mild steel cups on its periphery. The major dimension of okra seed was 5.71 mm, therefore, the diameter and depth of seed cups were 6 mm. The spacing between two cups was 25 mm. The cups were inclined at an angle of 450 from vertical to insure dropping of seeds into seed tube. A mild steel shaft 15 mm diameter and 120 mm length, supported on ball bearings for smooth rotation, was used to mount the circular plate with cups. A semi-circular hopper was made from 5 mm thick mild

Fig. 1 Details of seed metering mechanism

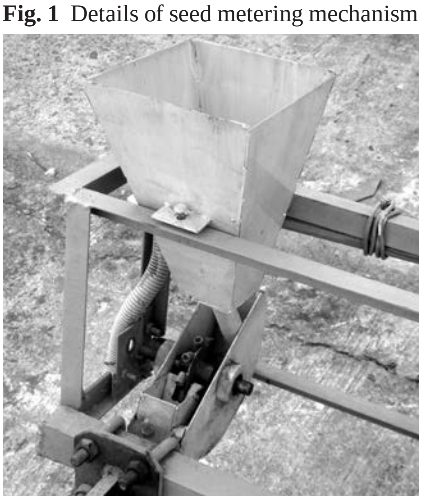
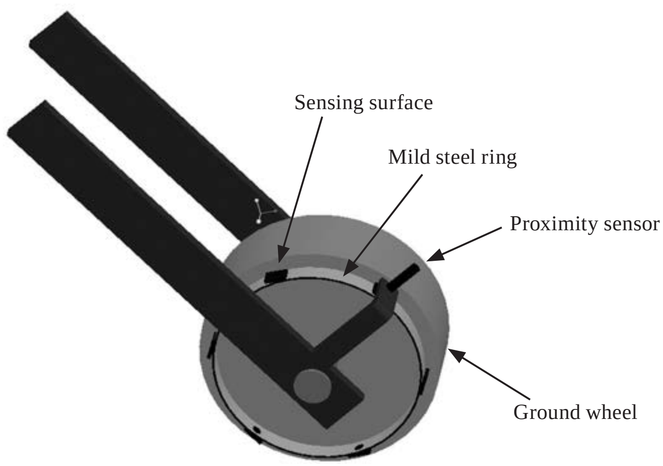


Fig. 2 Details of ring with sensing surface



steel sheet for housing the seed metering plate with cups. A seed box was also fabricated and fitted above the seed metering unit to hold a sufficient quantity of seed.

Ring with Sensing Surface

A circular ring, 286.5 mm diameter, simulating a lug wheel of a drill/planter was fabricated, using mild steel flat of size 25 × 5 mm (**Fig. 2**). The diameter of the ring was selected in such a manner so as to obtain its circumference in multiples of minimum within-row seed spacing which was 15 cm for this study. The circumference of this ring was 900 mm. Mild steel iron pieces, 50 mm long and 25 mm wide, were attached on the periphery of this ring at equal spacing as sensing surfaces. The total number of sensing surfaces were 6, 3 and 2 for target seed to seed spacing of 15 cm, 30 cm and 45 cm, respectively. This ring was attached by the side of a heavy wheel 385 mm diameter for its proper operation.

Electronic Assembly

The electronic assembly (**Fig. 3**) mainly consisted of proximity sensor, electronic circuit box and a DC motor. A 12 V tractor battery was used as the power source for the electronic circuit.

Proximity Sensor

A proximity sensor, popularly known as proximity switch, was used to generate an electrical pulse when it sensed the sensing surface mounted on circular ring.

Electronic Circuit Box

An electronic circuit box that included a pulse generator, BCD counter (IC 4510), Timer (IC 4093) and Relay unit was used. A pulse generator generated an electrical pulse for switching the relay unit "ON". The number of electrical pulses generated was counted and compared by the BCD counter, which also generated a time sequence to control the whole operation. A timer was used to control the duration of operation of a DC motor. The relay unit, when ON, controlled the flow of current from the tractor battery to the DC motor. A thumb wheel and screw control knob is provided on the front panel of the electronic box for setting a digital number from 0 to 9. When electrical pulses generated became equal to the preset digital number, then only the DC motor was allowed to operate and drive the seed metering device.

D. C. Motor

A 12 V, 42 rpm DC motor was used to operate the seed metering plate with cups. The DC motor was connected in parallel across the BCD counter. The duration of rotation of the motor was fixed in such a manner that its shaft rotated equal to the peripheral distance between the two seed cups in a single impulse of DC current.

Working of Electronic Controlled Metering Mechanism

As the circular ring was rotated by the variable drive unit the sensing surfaces fitted on its periphery appeared before the proximity sensor, which was fitted in a vertical position just above the ring. The proximity sensor generated an electrical pulse after detecting each and every individual sensing surface. A digital number 1 was set with the help of a thumb wheel. The counter (BCD) compared the number of electrical pulse being generated with the preset digital number for equality and, when both were equal, it allowed the signal to put the relay unit 'ON'. When the relay unit was set 'ON' it allowed the flow of current from the tractor battery to the DC motor which, in turn, rotated the seed metering unit to drop the seeds on to the greased belt.

Test Set Up

The performance of the electronically controlled metering mechanism was evaluated in the laboratory using a greased belt test setup. The setup was a 280 mm wide endless canvass flat belt supported between two head pullies of 300 mm diameter and fitted 5 m apart. Four auxiliary pullies were also provided to support the weight of belt.

An electric motor (3.73 kW) with a suitable reduction unit was used to run the belt at required liner speed. A mild steel rectangular stand 750

Fig. 3 Components of an electronic circuit

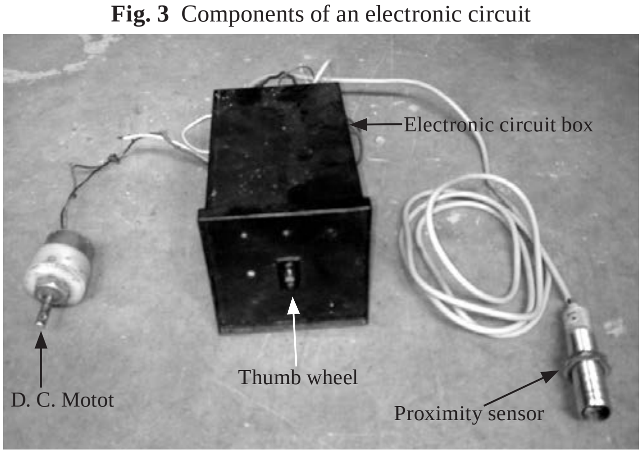
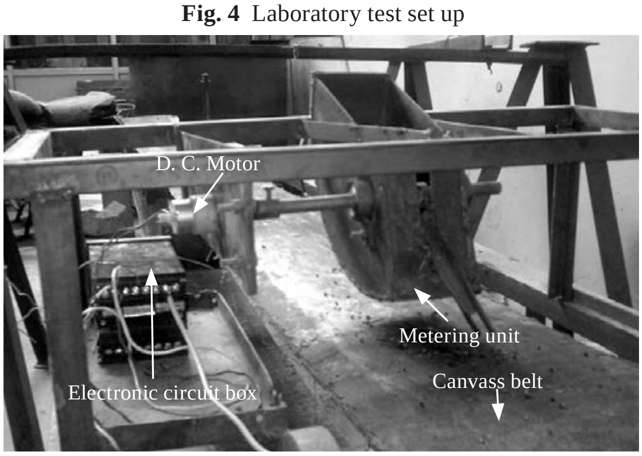


Fig. 4 Laboratory test set up



mm × 450 mm was used to mount the seed metering unit along with electronic assembly. The DC motor was fitted directly on the shaft of the seed metering unit. The drive to the circular ring with sensing surface and canvass belt was provided from the same shaft so as to derive equal linear speed in a similar manner as that of a lug wheel in drills/planters. **Fig. 4** shows the details of the test setup along with the electronic assembly.

Test Procedure

The hopper was filled with sufficient quantity of okra seed for one hundred percent cell fill. Grease was smeared on the canvass belt along its length for easy adhering of seeds. The canvass belt and circular ring with sensing surfaces were operated at required speeds through the electric motor. The developed metering mechanism was tested for three levels of theoretical within-row seed spacings of 15, 30 and 45 cm. The seed to seed spacing was varied by changing the number of sensing surfaces on the circular ring. Before actual conduct of test, a trial run was conducted for maximum achievable linear speed of the canvass belt. During the trial run, the minimum achievable speed limit was 1.0 km/h, whereas, the maximum speed limits were 1.85, 2.75 and 3.4 km/h respectively for above mentioned seed spacings. Therefore, the test was conducted for all the three mentioned speed limits.

During the laboratory test, the actual spacings between the two consecutive seeds dropped on the canvass belt were measured for a row length of 80 cm for each set of experiment. The number of replication was four for all the combinations of seed spacing and operating speeds. For evaluating seed spacing uniformity the Miss Index, Multiple Index, Quality Feed Index and Precision Index were determined as suggested by Kachman and Smith, 1995. The quality of feed index was the percentage of spacings that were more than half but no more than 1.5 times the theoretical spacing. It was the measure of how often the spacings were close to the theoretical spacing. Other parameters like number of seeds per meter length and seed damage were also determined. The data so obtained were analyzed statistically using a Completely Randomized Design.

Table 1 Physical properties of okra seed

Parameters	Average
Physical dimensions (Average for 500 seeds)	
1) Length (l), mm	5.61
2) Breadth (b), mm	4.75
3) Thickness (t), mm	3.79
Geometric mean, $(lbt)^{1/3}$, mm	4.60
Sphericity, $\{(lbt)^{1/3}\}/l$	0.74
Shape of seed	Oblong
Bulk density, g/cc	0.55
Angle of repose, degree	27.5
Moisture content, % wb	8.90
Initial germination, %	83

Results and Discussion

Physical Properties

Physical properties of okra seed (Variety- Parbani Kranti) were determined and the results have been presented in **Table 1**. On an average the seeds were 5.61 mm long, 4.75 mm wide and 3.79 mm thick with sphericity as 0.74. Based on sphericity, the shape of seed was oblong. These dimensions were used for deciding the dimensions of the cup of the seed metering unit. The average weight of one thousand seed was 65.80 g with a bulk density as 0.55 g/cc. The angle of repose ranged from 25.5 to 28.5 degree with an average value of 27.5 degree. The moisture content of seed was 8.86 percent. The initial germination of the okra seed was 83 percent.

Performance on Seed to Seed Spacing

The seed to seed spacing was measured during laboratory test for all the combinations of target seed spacing and operating speeds and the average value is shown in **Table 2**. At 15 cm target seed spacing,

Table 2 Estimated performance parameters for electronically controlled metering mechanism

Target spacing, cm	Speed of operation, km/h	Average observed seed spacing, cm	Standard deviation	Quality of feed index	Multiple index	Miss index	Precision	CD (P=0.05)	Significance level
15	1.00	15.3	0.895225	100	0	0	5.96817	1.187577	NS
	1.50	15.2	0.988345	100	0	0	6.588965		
	1.85	15.3	1.101466	100	0	0	7.343107		
30	1.00	30.4	1.03619	100	0	0	3.453968	0.9696435	NS
	2.00	30.8	1.225452	100	0	0	4.08484		
	2.50	31.0	1.402341	100	0	0	4.674471		
	2.75	30.9	1.897631	100	0	0	6.325438		
45	1.0	45.2	0.825903	100	0	0	1.835341	1.021149	NS
	2.0	45.1	1.170854	100	0	0	2.601898		
	3.0	45.4	1.528992	100	0	0	3.397761		
	3.4	46.1	2.005258	100	0	0	4.456128		

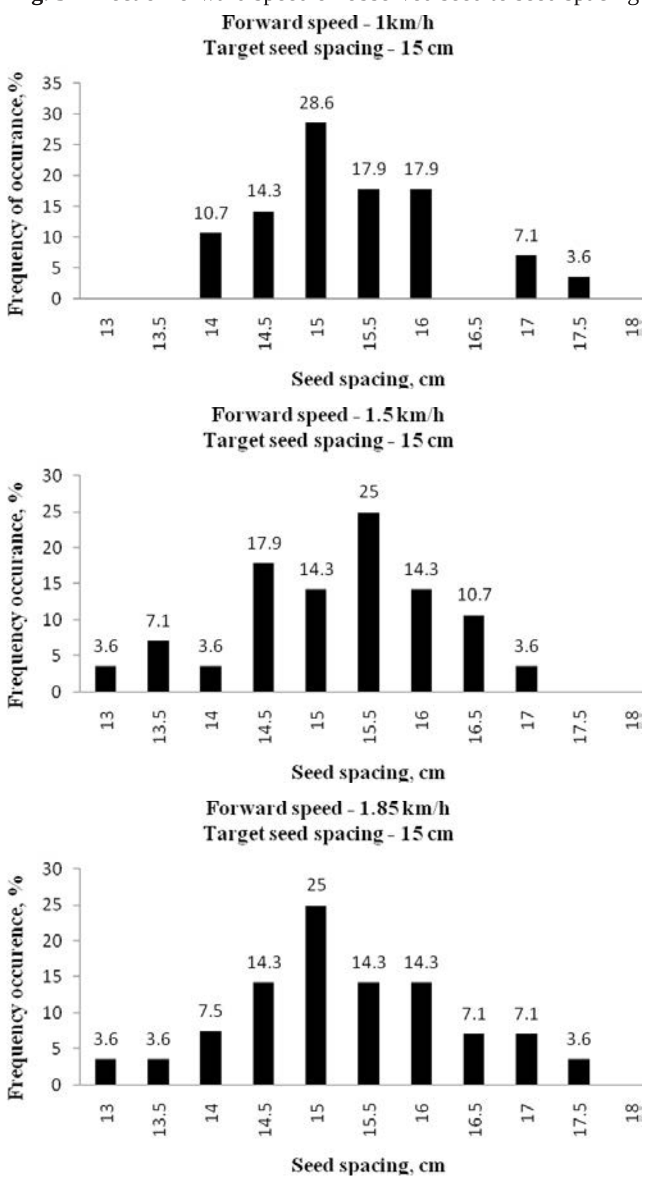
the average observed seed to seed spacings were 15.3, 15.2 and 15.3 cm at forward speeds of 1.0, 1.5 and 1.85 km/h, respectively. Similarly for 30 cm target seed spacing, the observed seed spacings were 30.4, 30.8, 31.0 and 30.9 cm at 1.0, 2.0, 2.5 and 2.75 km/h forward speed, respectively. When the number of sensing surfaces was adjusted to achieve a target seed spacing of 45 cm, the seed to seed spacings were 45.2, 45.1, 45.4 and 46.1 cm at forward speeds of 1.0, 2.0, 3.0 and 3.4 km/h, respectively. The percent frequency occurrence of observed seed to seed spacing for various target spacing and forward speeds have been presented in **Figs. 5** to **7**. The results indicate that the developed metering mechanism was able to deliver seed very close to the target seed to seed spacing with little variation ranging from a minimum of 0.22 to maximum 3 percent. The percentage variation was lowest for 45 cm target seed to seed spacing compared to 15 cm and 30 cm. Results also indicated higher accuracy at lower forward speeds for all the combinations of treatments.

Using the complete set of target seed spacing and operating speeds, various performance parameters were also estimated utilizing laboratory data on observed seed to seed spacing (**Table 2**) as suggested by Kachman and Smith, 1995. The precision (which is the coefficient of variation of spacing different than the usual statistical coefficient of variation and has been used as a measure of variability) indicated higher variability among the observed seed to seed spacing with an increase in operating speed for all the experiments. However statistical analysis showed no significant variation for observed seed to seed spacing with the increase in operating speeds at the 5 percent significance level. The reason for observed variations may be due to bouncing of the seeds. The miss and multiple indexes were zero, which indicated neither missing nor multiple seeds. The quality of feed index was 100 percent for all the combinations of parameters indicating that the observed seed to seed spacings delivered by the seed metering mechanism were, most of the time, very "close" to the target seed to seed spacing.

Number of Seeds per Meter Length

The number of seeds delivered per meter length during laboratory test was counted (**Table 3**). At 15 cm target seed spacing, the number of seeds per meter length varied between 5 and 7 seeds with an average of 6 seeds for all the three levels of forward speeds. For 30 cm target seed spacing, the average observed number of seeds per meter length was 4, 3, 3 and 4 seeds at forward speeds of 1.0, 2.0, 2.5 and 2.75 km/h, respectively. Similarly for 45 cm target seed spacing, the average number of seeds per meter length was 2 seeds for all the levels of forward speed. The results indicate that the metering device was able to singulate the seeds and delivered the desired number of seeds per meter length quite accurately for all

Fig. 5 Effect of forward speed on observed seed to seed spacing



the levels of target seed spacing and forward speeds of operation. However, it delivered very accurately for 45 cm target seed to seed spacing compared to other required seed spacings. The reason could be due to more time available for proximity sensor to sense two consecutive sensing surfaces which in turn might be helping in proper functioning of the electronic circuit and metering unit.

Fig. 6 Effect of forward speed on observed seed to seed spacing

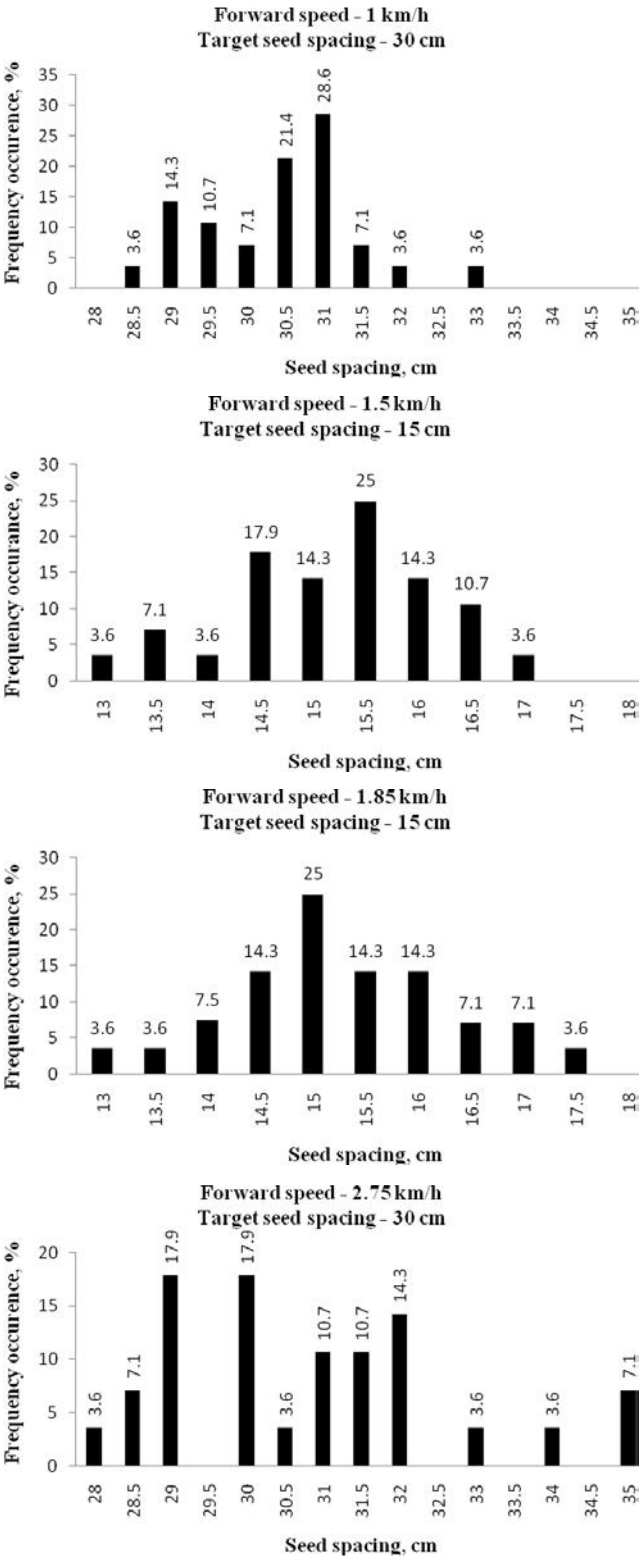
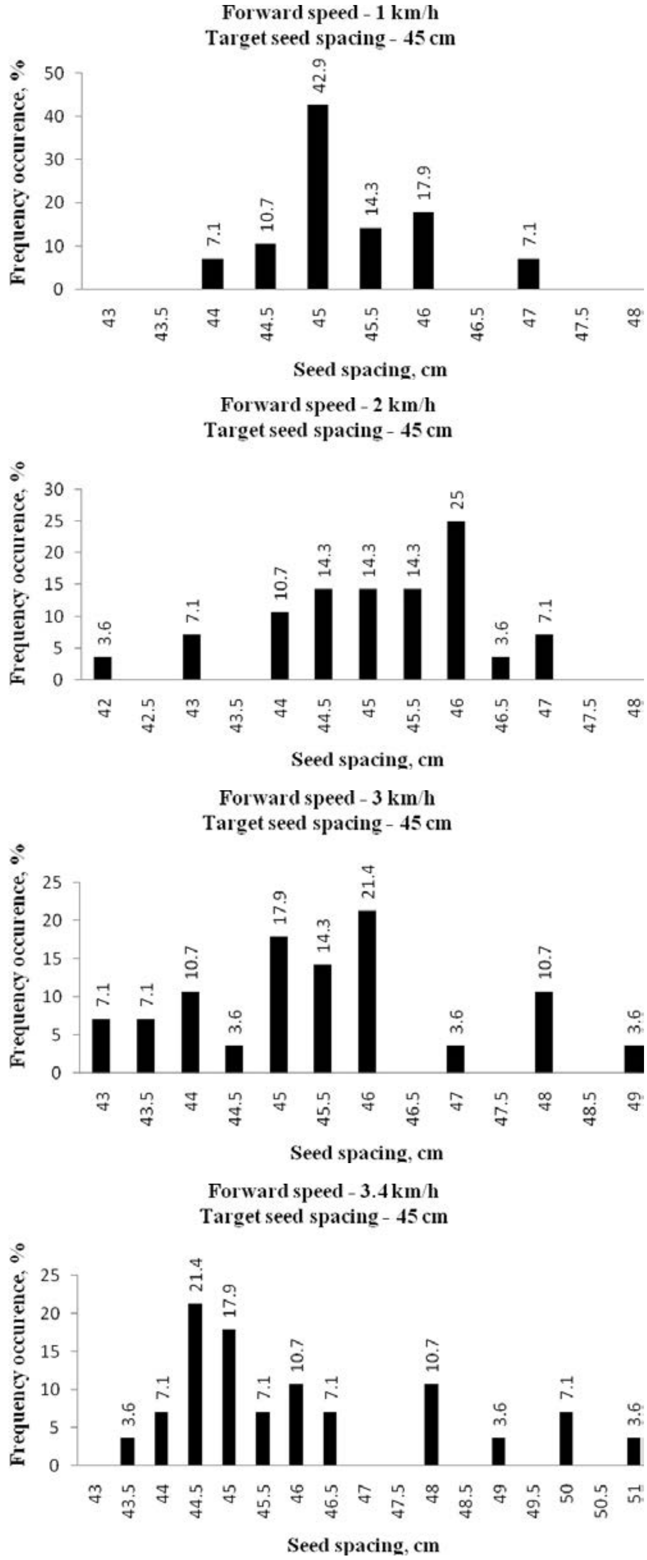


Fig. 7 Effect of forward speed on observed seed to seed spacing



Seed Damage

The external (visible) and internal seed damage was determined by visualizing as well as conducting germination tests for the seeds received after passing through the metering device. The results of the germination tests (**Table 4**) showed that the germination percentage was equal to the initial value of seed germination indicating no internal seed damage caused by the metering device. Visual observation of seed also showed no mechanical injury.

Conclusions

Based on the results of laboratory test, it can be concluded that the developed electronically controlled seed metering mechanism performed satisfactorily and delivered the seeds quite closer to the target seed spacings. The quality of feed index was higher, with miss and multiple indexes of zero. No mechanical damage to seed was observed. The developed metering device gave better results for higher seed to seed spacing. The developed seed metering mechanism did not work beyond the maximum achievable speed limits for the entire target seed spacings.

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Table 3 Number of seeds per meter length under laboratory test

Particulars		Target seed spacing (15 cm)			Target seed spacing (30 cm)				Target seed spacing (45 cm)			
Forward speed, km/h		1.0	1.5	1.85	1.0	2.0	2.5	2.75	1.0	2.0	3.0	3.4
Number of seeds dropped per meter length	R1	5	6	7	3	3	3	3	2	2	2	2
	R2	6	6	6	3	3	3	4	2	2	2	2
	R3	7	6	6	4	3	3	4	2	2	2	2
	R4	7	6	6	4	4	3	4	2	2	2	2
Mean		6	6	6	4	3	3	4	2	2	2	2
SD		0.96	0	0.50	0.58	0.50	0.0	0.50	0	0	0	0

Table 4 Germination percent of the Okra seed

Particulars	Replications	Number of seeds in sample	Germinated seed	Germination, %	External seed damage
Initial seed germination	1	100	84	84	Nil
	2	100	83	83	
	3	100	84	84	
	4	100	84	84	
	Mean	100	-	84	
Seed germination after passing through metering device	1	100	84	84	Nil
	2	100	82	83	
	3	100	85	84	
	4	100	84	84	
	Mean	100	-	84	

Anthropometric Dimensions of Turkish Operators of Loading Tractors Used in Forestry and the Design of the Workplace

by

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Abstract

This study aims to determine the principles of anthropometric location so that loading machine operators in Turkey are able work in a healthier, safer, and more productive environment. For this research, anthropometric data on the operators, measured with the photometric method, were first analyzed using statistical methods and then compared with the results of similar studies carried out in Turkey and other countries throughout the world. Second, the anthropometric compatibility between loading machines and their operators was examined in order to determine the parameters of the tractor control area suitable for Turkish operators. This study used 145 machines and operators working in loading and sorting operations in forest reserves within the Western Black Sea Region. Operators studied had an average stature of 1,744 mm, knee height of 562 mm, sitting height of 886 mm, and buttock to knee length of 597 mm. The most important data for the design, popliteal height, and hip breadth, were, respectively, found to have a 5 percent value of 433 mm and a 95 percent value of 412 mm. Although the results were similar to those of previous studies carried out in Turkey, there were considerable differences in anthropometric values between various nations. The study found that, in the region, the vertical adjustment length of operator seats was ± 49 mm and the horizontal adjustment length was ± 39 mm. In selecting loading machines for operator use, attention should be paid to the compatibility of operator control areas with the anthropometric characteristics of operators so that operators can use loading machines in a healthier, safer, and more productive manner. Operator seats already in use should be substituted with ones that are vertically and horizontally adjustable, and that are compatible with operators' anthropometric characteristics.

Introduction

In Turkey's forest reserves, agricultural tractors are used to carry loading equipment, while loading machines are extensively used for the loading and stacking of forest products. Loading holds an important function in efficiently transporting the raw wood material. The design of loading machines should, therefore, take into account the physical characteristics of machine operators so that they can increase their performance in production phases, carry out their many complex tasks with in such a way that they do not grow tired as quickly and are better able to ensure their safety (Liljedahl *et al*., 1996).

The arrangement of the control instruments in tractors used in Turkey are determined in accordance with the anthropometric characteristics of people from the country where the machine's patent agreement is signed. Current tractor seat dimensions were determined by the International Standard Organization (ISO), which took anthropometric characteristics into consideration. However, the dimensions determined by ISO are not fully compatible with the anthropometric dimensions of operators in Turkey (Sabanci, 1981 and Dizdar, 2004).

It is critical to consider the operators' body dimensions so that they are able to effectively control their machines, have better peripheral vision, easily monitor their surroundings, and comfortably move inside their vehicle (Eby and Kantowiz, 2006). Operator seats inside machines used in lumber extraction should properly support an operator's legs, buttocks, and back. Additionally, they should provide for greater maneuverability of both the body and the vehicle (Gellerstedt *et al*., 1999).

The harmony between the design of control instruments and the operator considerably affects the success and productivity of an operator. The element of a machine most important to the work and health of an operator is the seat. The breadth, length, height from the foot, level and shape of the seat, and the position of the seat in relation to control instruments are essential considerations for anthropometric location.

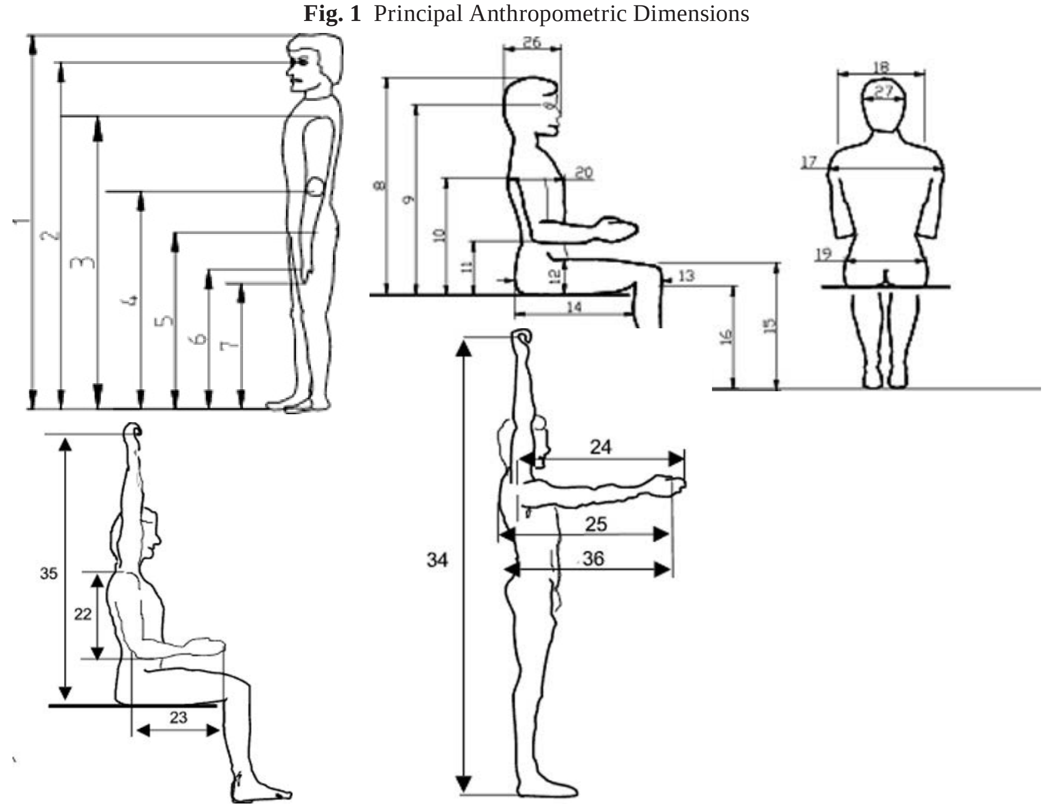
Previous studies have determined that there was no ergonomic suitability between anthropometric location and the control areas of tractor operators in Turkey (Sabanci, 1981; Has, 1999 and Kocturk and Avcioglu, 2006). In addition, there were a number of studies in other countries in on the compatibility of location design in control areas of tractors with the anthropometric dimensions of operators (Juliszewski, 1990; Gullberg, 1996; Yisa, 2002 and Hsiao *et al*., 2005). A study carried out in Nigeria examined the anthropometric dimensions, sitting positions, and location features of tractor operators, and determined that operators faced a number of inconveniences as a result of the fact that the location of seats and control organs were not ergonomically sound (Adekoya, 1993).

This study aims is to determine principles of anthropometric location so that operators of loading machines used in Turkey are able to work in a healthier, safer, and more productive manner. To do so, this study first examines the operators' anthropometric data, measured with the photometric method, and then analyzes the data using statistical methods. These data are then compared with the results of previous studies carried out in Turkey and other countries. Finally, the study examines the anthropometric compatibility between machine and the operator,and determines the appropriate parameters of the tractor control areas for Turkish operators.

Materials and Methods

This study was carried out on 145 loading machines and operators working in forest reserves of the Western Black Sea Region. The measurements for the study were made on International, Ford and Massey Ferguson agriculture tractors used to carry loading equipment, and Komatsu and Hidromek loading machines. Anthropometric measurements for height and diameter measurements were made with an anthropometer, tape measure and electronic caliber gage. Perimeter

Fig. 1 Principal Anthropometric Dimensions



measurements were made with an elastic tape measure, while weight was measured with an accuracy of within 100 grams. For the photometric method, the study utilized a Sony camera (5 megapixels), a chair with etched markings, and two red and white traffic cones. In order to determine the dimensions of tractors, the study used manufacturer guides to the machines and a tape measure. Computer measurements were then made using the 2006 Photometrix Iwitness 3-D drawing program.

The anthropometric variables measured in the study were determined according to methods of Pheasant (1998) and Sabanci (1999). Definitions of 37 anthropometric dimensions (including weight) measured in the study are given below in **Fig. 1** and **Table 1**.

In this study, a method for stereo-

Table 1 Descriptive and Percentile Values Relating to Anthropometric Measurements of Operators (mm)

	Antropometrik ölçüler	% 50	% 5	% 95	Std. Dv.	% Cv	England	Brazil	India	Sabanci	Ozok
A_1	Stature	1,744	1,667	1,821	46.8	2.68	1,740	1,700	1,620	1,691	1,681
A_2	Eye height	1,617	1,548	1,686	42.2	2.61	1,630	1,595	1,510	1,582	1,572
A_3	Shoulder height	1,439	1,376	1,502	38.3	2.67	1,425	1,410	1,345	-	1,382
A_4	Elbow height	1,104	1,039	1,169	39.5	3.58	1,090	1,045	1,025	-	-
A_5	Hip height	882	816	949	40.5	4.59	920	880	865	-	751
A_6	Fingertip height	689	612	767	47.3	6.86	655	625	585	-	624
A_7	Knuckle height	782	713	851	42.1	5.39	755	720	685	-	-
A_8	Sitting height	886	833	940	29.6	3.68	910	880	840	895	887
A_9	Sitting eye height	759	715	803	27.0	3.56	790	775	740	1,163	776
A_{10}	Sitting shoulder height	576	533	619	26.2	4.55	595	595	555	597	594
A_{11}	Sitting elbow height	241	193	289	29.2	12.08	245	230	205	256	-
A_{12}	Thigh thickness	177	141	213	21.9	12.41	160	150	135	-	-
A_{13}	Buttock-knee length	597	558	636	23.7	3.97	595	595	555	568	610
A_{14}	Buttock-popliteal length	470	413	526	28.5	7.34	495	480	465	445	-
A_{15}	Knee height	562	517	607	27.6	4.91	545	530	510	529	503
A_{16}	Popliteal height	493	443	543	28.3	6.15	440	425	415	411	-
A_{17}	Shoulder breadth (biacromial)	370	333	407	22.5	6.09	400	385	355	397	-
A_{18}	Shoulder breadth (bideltoid)	397	364	431	20.4	5.13	465	445	410	-	-
A_{19}	Hip breadth	365	318	412	28.6	7.85	360	340	310	354	356
A_{20}	Chest depth	251	209	293	25.6	10.18	250	235	170	228	-
A_{21}	Abdominal depth	249	209	289	24.5	9.82	270	245	185	-	-
A_{22}	Shoulder-elbow length	370	339	401	18.8	5.07	365	365	355	351	-
A_{23}	Elbow-fingertip length	471	427	514	26.8	5.69	475	475	460	454	-
A_{24}	Upper limp length	835	769	902	40.7	4.87	780	785	755	846	828
A_{25}	Shoulder-grip length	741	676	807	40.0	5.40	665	670	710	-	-
A_{26}	Head length	193	180	207	8.2	4.22	195	190	180	-	184
A_{27}	Head breadth	147	138	155	5.1	3.51	155	150	145	-	161
A_{28}	Hand length	188	180	196	5.0	2.65	190	185	185	-	191
A_{29}	Hand breadth	95	80	111	9.5	9.94	85	85	85	-	105
A_{30}	Foot length	271	255	286	9.4	3.46	265	260	250	255	261
A_{31}	Foot breadth	99	89	109	6.1	6.14	95	100	95	96	101
A_{32}	Span	1,770	1,675	1,865	58.0	3.28	1,790	1,755	1,705	-	1,711
A_{33}	Elbow span	883	799	967	51.4	5.82	945	925	880	-	-
A_{34}	Vertical grip reach (standing)	2,098	1,975	2,220	74.6	3.56	2,060	2,020	1,995	-	-
A_{35}	Vertical grip reach (sitting)	1,235	1,125	1,344	66.6	5.40	1,245	1,220	1,190	-	-
A_{36}	Forward grip reach	824	749	899	45.7	5.54	780	765	725	-	-
A_{37}	Body weight	74	59	89	9.4	12.62	75	66	49	69	66

photometric measurement was also used in order to resolve any problems resulting from taking anthropometric measurements in difficult field conditions (Robinette and Hudson, 2006). Final anthropometric measurements were subsequently determined using the 2006 Photometrix Iwitness 3-D computer drawing program by overlapping photographs instead of taking measurements in the field one by one (**Fig. 2**).

When measuring anthropometric dimensions, a total of ten photographs were taken of each operator, with five taken of the operator in a seated position and five taken of the operator in standing position. The series of photographs were taken starting at a single point and continued at intervals of roughly 45o angles around the operator, taking 5 photographs at each point. The photographs were taken in such a way that it was possible to precisely see the operator and reference points. The operators were asked to pose in two different positions in order to get the required anthropometric measures with the least number of photographs. In the first position, the operator stood with one arm hanging loosely by the side of the body and the other arm raised in the air; photos were taken in this position starting from the right and moving to the left. In the second position, the operator remained seated, with one arm bent at the elbow and the other extended in front of the body, while one foot remained planted on the ground while the other leg was bent forward.

In addition to the measurements obtained through the photographs, the distances between the fingertips and elbows of operators were measured in the field, with operators standing with their both arms raised. Moreover, stature and the distance between the elbow and finger tips were measured and noted to check for accuracy in the measurements. All of the anthropometric dimension measurements of length were made in millimeters, while weight was measured in kilograms.

The study determined the number of samples by making use of the anthropometric data. Pre-measurements were therefore made on 29 operators in the study group. The number of samples was calculated using the variation of the values at the sitting height, the standard deviation of which was the highest (30.5) among the all measurements made in the seated position. The number of samples (N) which was calculated as 143 was taken as 145 in the study.

$N = [\ (K1 \times S) / (d)]^2$

K1: 1,96 (t-value according to confidence interval % 10-90)

$N = [\ (1.96 \times 30.5) / (5)]^2$

d: sensibility (5 mm)

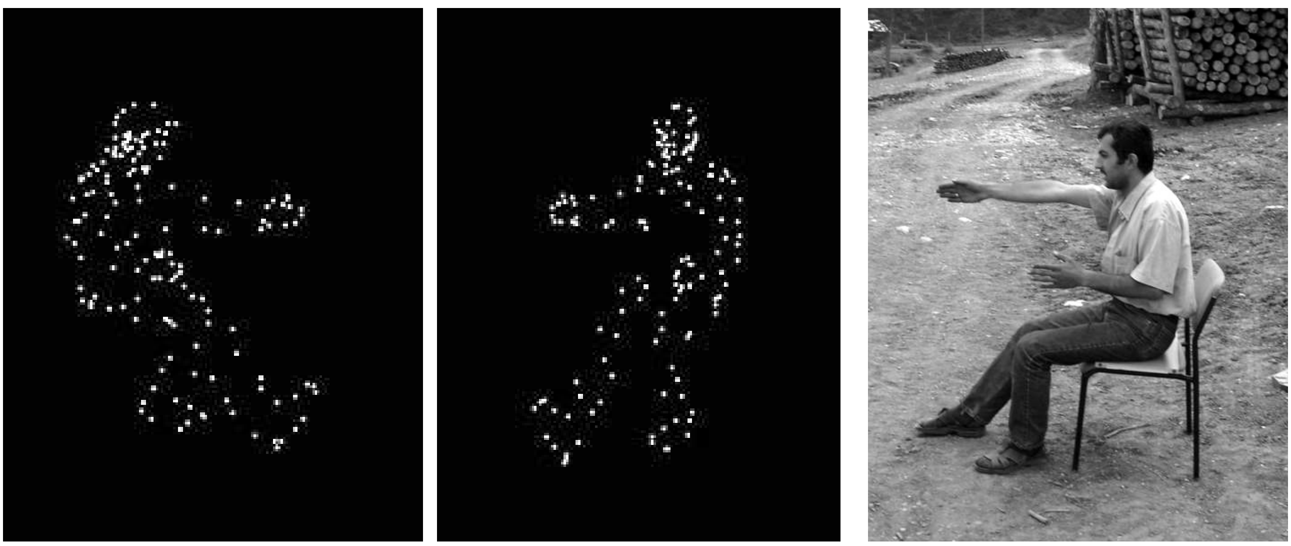
$N = 143$

S: 30.5 mm (the highest SD)

The study first determined the mean, standard deviation, variation coefficient, and the lowest and highest values of anthropometric measures belonging to the 145 loading machine operators. The anthropometric dimensions of operators in the region were compared with the results of studies carried out in Turkey and in other countries. The results were compared with those of previous studies, such as those of Sabanci (1999) and Ozok (1981) from Turkey, Pheasant (1998) from England, Brazil Instituto Nacional de Tecnologia (INDT) (1989) from Brazil, and Gite and Yadav (1989) from India. Through this comparison, this study sought to demonstrate the difference between the anthropometric measures obtained in this study and those in different parts of Turkey and other countries.

The compatibility of anthropometric measurements and machine-seat dimensions was subsequently

Fig. 2 Three-Dimensional View of an Operator



evaluated. The breadth, depth, and height of existing tractor seats, the height of any backrests and armrests, and the distance between the two armrests were measured. The compatibility of these values with the percentile anthropometric values was then evaluated. Next, the vertical and horizontal adjustment dimensions of the seats were calculated. The dimensions and parameters of operator seats in loading machines that could be used in the region were then determined, taking into account both data obtained in this study and world standards (ISO, 3411; Gellerstedt *et al.*, 1999). Measures for vertical adjustment of seats were determined using the values of variation in sitting height (vertical distance from the sitting surface to the crown of the head). Likewise, measures for horizontal adjustment were determined using the values of variation in buttock to knee length (Roubeck *et al.*, 1975; Sabanci, 1999). Making use of the operator dimension measures mentioned above, the adjustment dimensions of seats were calculated as follows:

Length of the vertical adjustment;

$X_d = 2 \times ss_d \times t$

(t: 1.645 for 5 % and 95 %)

Length of the horizontal adjustment;

$X_y = 2 \times ssy \times t$

ss_d: standard deviation of the dimension taken into consideration for horizontal adjustment.

ss_y: standard deviation of the dimension taken into consideration for vertical adjustment.

For a person working with an agriculture tractor in a seated position, limb movements were restricted to a clear, confined space. No clearly defined standard method existed for determining the dimensions of this operator control area (Sabanci, 1999). This study, however, determined optimum and maximum areas for eyes, hands, and feet in both the horizontal and vertical control area. The values from measurements and calculations for tractor control areas were obtained in accordance with the research methods implemented by Zander (1972) and Sabanci (1981).

Results

Table 1 presents the descriptive and percentile values of 37 anthropometric dimensions and the body weight of loading machine operators involved in the study (see **Fig. 1**).

From **Table 1**, one can see operators had an average stature of 1744 mm, knee height of 562 mm, sitting height of 886 mm, and buttock to knee length of 597 mm. The 5 percent value of popliteal height was 433 mm, and the 95 percent value of hip breadth was 412 mm. Popliteal height and hip breadth are the measurements most critical for use in design. The highest variations in the data were found in body weight (12.62), elbow height (12.08) and calf thickness (12.41).

These anthropometric dimensions were compared with those obtained in previous studies in Turkey and other countries. In Europe and the United States, taking into account anthropometric measurements, has been regular practice for years. However, few studies have been carried out in Turkey on this subject (Sabanci, 1999; Dizdar, 2004). In this context, comparisons were made with similar research in Turkey, England, Brazil and India (**Table 2**). The values of anthropometric dimensions obtained from this study were compared with the mean values of these prior studies carried out in England (Knight, 1984), Brazil (INDT, 1989), India (Gite and Yadav, 1989), and Turkey (Sabanci, 1981 and Ozok, 1981).

Table 2 shows, for instance, that the average stature of operators is 1,744 mm in England, 1,700 mm in Brazil, 1,620 mm in India, 1,691 mm (Sabanci) and 1,681 mm (Ozok) in Turkey, and 1,744 mm in the group examined by the present study.

Table 2 Seat Dimensions for Various Machines

Seat dimensions	Mean	Standard deviation
Seat breadth	47	4.30
Seat height	46	1.30
Seat depth	35	3.84
Backrest height	40	10.74

Table 2 describes the average values of seat height, breadth, depth, and backrest height of the seats of loading machines used in the region. Only 10 percent of the tractors examined in this study had armrests.

The average seat dimensions of tractors and anthropometric measures obtained from operators were compared. It was found that the

- 5 percent value of buttock-popliteal length and seat depth were, respectively, 413 mm and 350 mm
- 95 percent value of hip breadth and seat breadth were, respectively, 412 mm and 470 mm
- 5 percent value of popliteal height and seat height were, respectively, 443 mm and 460 mm
- 5 percent value of sitting shoulder height and backrest height were, respectively, 533 mm and 400 mm.
- 5 percent value of elbow height while seated was 241 mm

From these figures, it appears that seat breadth, seat depth, and backrest height of the seats in existing machines were ergonomically suitable for the people in the region; however, seat height was not compatible. As seen below, the vertical and horizontal adjustment dimensions of seats were calculated taking into account the operator dimensions described above. Standard deviation values of seated height for vertical adjustment and buttock to knee length for horizontal adjustment were calculated as shown here:

Length of vertical adjustment;

$X_d = 2 \times 29.6 \times 1.645 = 97$ mm

Length of horizontal adjustment;

$X_y = 2 \times 23.7 \times 1.645 = 78$ mm

Calculations revealed that the limits of vertical and horizontal adjustment were 97 mm (~ 10 cm) and 78 mm (± 4 cm), respectively. Operator seat dimensions of loading machines used in the region and their parameters are presented in **Fig. 3**, taking into account both the data collected in this study and current world standards (ISO, 3411 and Gellerdtedt *et al*., 1999).

Table 3 presents the results of calculations made with the mean values of anthropometric data. They were applied in order to determine tractor operators' eye, hand and foot control areas. Eye, hand and foot, vertical and horizontal, optimum, maximum-I and maximum-II control areas can be determined and workplace designed in accordance with the average dimensions (**Fig. 4**).

Discussion

This study measured the 37 anthropometric dimensions and body weight normally examined in the anthropometric studies carried out worldwide (Pheasant, 1998). The anthropometric dimensions measured in this study differ slightly from the past anthropometric studies (Sabanci, 1981; Ozok, 1981) carried out in Turkey in terms of the number of dimensions measured and results obtained.

By comparing the mean values of the anthropometric dimensions found in this study and those of previous studies (Sabanci, 1981; Ozok, 1981 and Kayis, 1990) carried out for different productive sectors, the following values were found:

- Average body weight was 69 kg in Sabanci, 66 kg in Ozok, 63 kg in Kayis, and 74 kg in this study

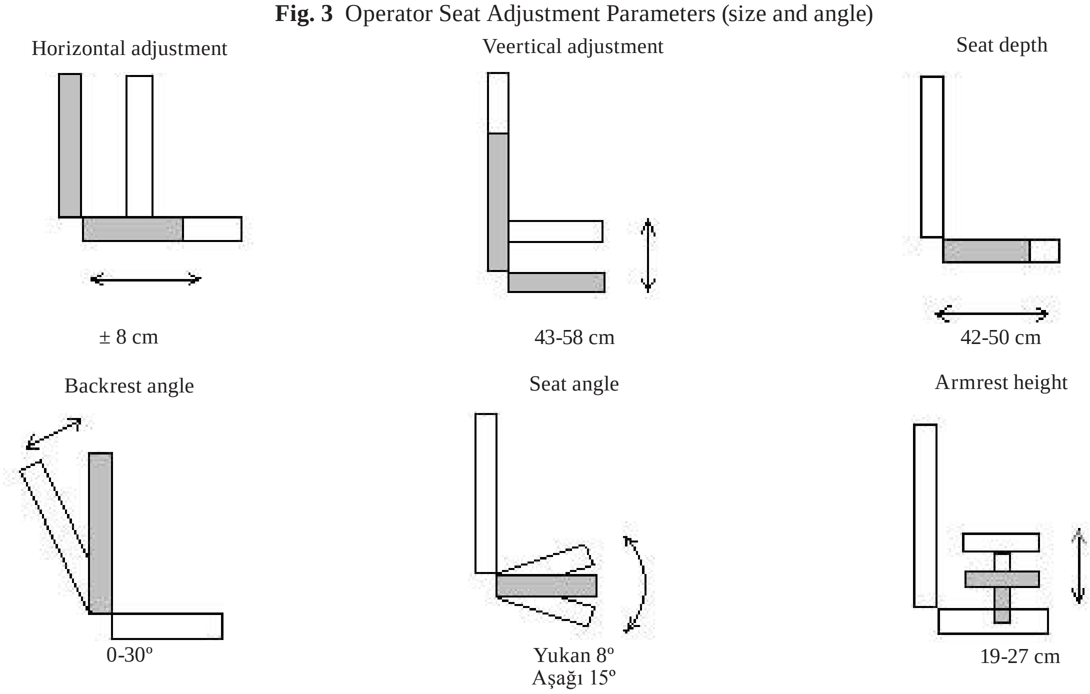
When analyzing the anthropometric studies carried out in Turkey, it is clear that similar results were obtained despite small differences between the mean values obtained. As seen in daily life, the stature and limb length of people from the same region - even people from the same family of similar age and gender —can differ greatly between one another. (Babalik, 2005).

Comparing the mean values of anthropometric dimensions in this study and those from prior studies in England (Knight, 1984), Brazil (BINT, 1989) and India (Gite and Yadav, 1989) reveals the following differences in values obtained:

- Average stature was 1,740 mm in England, 1,700 mm in Brazil, 1,620 mm for India, and 1,744 mm in this study
- Average seated height was 910 mm in England, 880 mm in Brazil, 895 mm in India, and 886 mm in this study
- Average knee to buttocks length was 595 mm in England, 595 mm in Brazil, 555 mm in India, and 597 mm in this study
- Average hip breadth was 360 mm for England, 340 mm for Brazil, 310 mm for India, and 365 mm in this study
- The value of upper limb was 780 mm for England, 785 mm for Brazil, 755 mm for India, and 835 mm in this study
- Average body weight was 75 kg in England, 66 kg in Brazil, 49 kg in India, and 74 kg in this study.

When evaluating the results of these studies, there were considerable differences between the anthro-

Fig. 3 Operator Seat Adjustment Parameters (size and angle)



pometric values found in various nations. For example, the measurements for theaverage stature of the in England were significantly greater than those in Brazil and India. It was discovered that the values closest to those of the people of the region examined in this study were those from the study carried out in England, while the values of the Brazilians and Indians most greatly differed. Anthropometric dimensions differ according to nation, region, gender, age, body structure, nutrition, physical activity, and even economic and social status (Singleton, 1983). The science of anthropometry requires constant investigation and evaluation of body dimensions in order to determine the anatomical differences between individuals and groups (Helander, 1997; Kroemer, 1997 and Dizdar, 2004).

The first ergonomic studies on tractors were concerned with seat dimensions, and the aim of these studies was to facilitate operators' work, reduce health risks, and utilize operators' energy more efficiently (Dincer, 1977). The study determined that seat breadth, seat depth and backrest height were compatible with the body dimensions of the people in the region, but that seat height was incompatible.

Table 3 Measured and calculated values for operator control areas

Control area	Anthropometric dimensions	From **Table 1** require calculate	Design value
Eye	Eye height	A9	759 mm
	Chest depth	A20	251 mm
	Visual angels, vertical		Opt: 30º, Max: 70º
	Visual angels, horizontal		Opt: 70º, Max: 180º
Hand	Sitting shoulder height	A10	576 mm
	Distance of the shoulder centre	a*	98 mm
	Shoulder-elbow length	A22	370 mm
	Elbow-grip length	A23 − (A28 / 2)	377 mm
	Elbow-fingertip length	A23	471 mm
	Upper arm angle-on the shoulder		Opt: 61º, Min: 38º
	Upper arm angle-under shoulder		Opt: 188º, Min: 168º
	Elbow angle		Opt: 38º, Min: 22º
	Distance of the elbows	b*	327 mm
	Shoulder breadth	A18	370 mm
	Lower arm angle, horizontal		Opt: 30º, Max: 70º
	Arm back angle, horizontal		Opt: 30º, Max: 70º
	Arm length	A24 − a	737 mm
	Functional arm length	A22 + b	697 mm
Foot	Hip centre height	c*	114 mm
	Hip distance	d*	143 mm
	Upper leg length	e = (A14 + A8) / 2 − d	390 mm
	Lower leg length	f = (A15 + A16) / 2	470 mm
	Heel-malleolus length		69 mm
	Foot length	A30	271 mm
	Hip breadth	A19	365 mm
	Distance of the heels		227 mm
	Distance of the knees		327 mm
	Foot breadth	A31	99 mm
	Leg length	e + f	918 mm
	Foot angle		9º, up, 18º
	Lower leg-knee angle, horizontal		20º, down, 23º

* Sabanci (1981)

The anthropometric dimensions of operators in this study revealed that the 5 percent value of buttock to popliteal length was 413 mm, 95 percent value of hip breadth was 412 mm and the 5 percent value of popliteal height was 443 mm. A study in India, on the operators of agriculture tractors revealed that the 5 percent value of buttock to popliteal length was 379 mm, 95 percent value of hip breadth was 256 mm, and that the 5 percent value of popliteal height was 366 mm (Mehta *et al.*, 2008). The fact that there are considerable differences among the anthropometric dimensions of tractor operators of the two nations requires that operators drive tractors with different seat and operator control dimensions.

Tractor seats can be adjusted to fit the anthropometric dimensions of operators through vertical and horizontal adjustment. This study found that the vertical adjustment length of operator seats in the Western Black Sea region was ± 49 mm and horizontal adjustment length was ± 39 mm. Sabanci (1981) observed that the vertical adjustment length was 50 mm and horizontal adjustment length was 45 mm. Clearly, the adjustment parameters of operator seats are similar to each other.

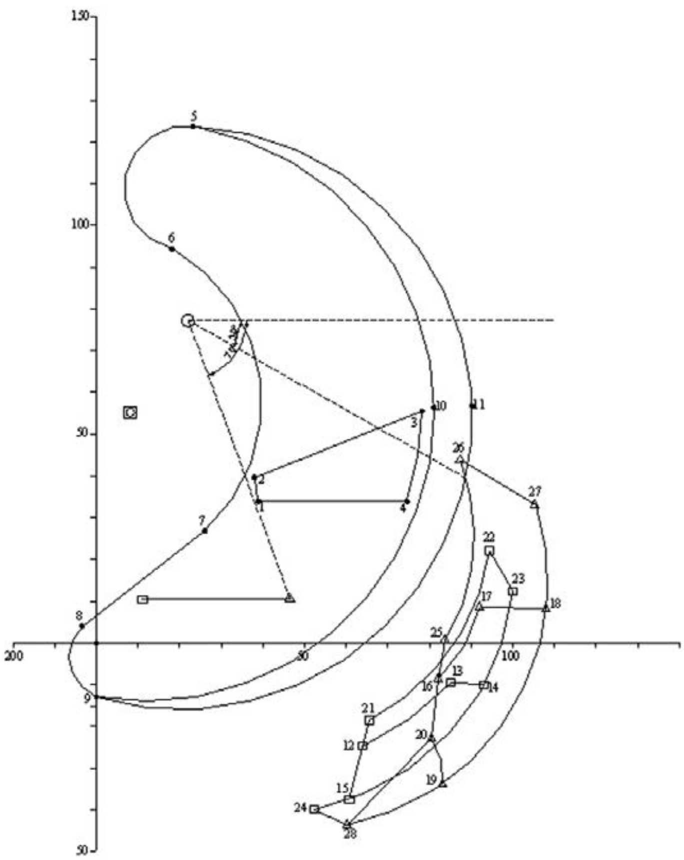
Furthermore, this study found that there was no armrest on the seats of most (90 %) of machines in the region and, for those that did have them, armrest height was not appropriate. Past studies found that, for agriculture tractor operators in India, the appropriate height for an operator's armrest was 300 mm (Tewari and Prasad, 2000). In this study, however, it was concluded that the armrest should be adjustable, providing a range of between 190 and 290 mm. If the armrests can be adjusted in accordance with the control mechanism, an operator will be able to work for a longer period of time without getting tired

(Sabanci, 1999), thus, reducing the operator's energy consumption.

Tractor operator workplaces have significant effects on the operation of tractors; if the control instruments are out of the control area, operator workload increases and while safety decreases. When control instruments are out of the area easily reached by an operator, the interaction between the operator and the machine are adversely affected, therefore, decreasing performance. In ergonomic terms, if extra energy must be spent reaching control instruments, this, in turn, increases the perceptional, mental and, particularly, the physical workload of an operator, negatively affecting the performance of the operator's system (Sabanci, 1981). In a study on the cabin conditions of the operators of rubber-tired loaders and excavators, for example, it was observed that cabins generally provided moderate or little comfort, but that cabin comfort increased with developments in seat design (Kujit-Evers *et al.*, 2003).

Toyokowa (1993) studied the symptoms of exhaustion for tractor operators, and discovered that the inappropriate location of control instruments was the only cause of operator exhaustion. Likewise, a study in Italy on various tractors determined that the control instruments that were difficult to use, and which were inappropriately arranged in relation to other ergonomic conditions such as noise and vibration, created unnecessary difficulties in working conditions of operators and had adverse effects on their productivity (Febo and Pessina, 1994).

Fig. 4 Vertical Design of Operator Workplace Area [Optimum Control Area; Eye (1-2-3-4), Hand (5-6-7-8-9-10) and Foot (12-13-14-15, 16-17-18-19-20)]



Conclusions

When selecting loading machines, attention should be paid to the compatibility between operator control areas and the anthropometric characteristics of operators so that operators can use the loading machines in a healthier, safer, and productive manner. Operator seats currently in use should be replaced by seats that are vertically and horizontally adjustable, and which are compatible with the anthropometric dimensions of operators. Anthropometric characteristics that affect the physical stress of operators, as well as critical ergonomic factors such as noise and vibration, should be continually evaluated so that loading operations can be performed in a healthier and more productive manner.

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Performance of 3.75 kW Diesel Engine Using Different Blends of Jatropha Oil

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Abstract

Petroleum consumption and its prices are increasing day by day. Moreover, the petroleum reserves in the world are rapidly depleting. Due to these reasons, there is a need for an alternative fuel for the compression ignition (CI) engine. The performance of a 3.75 kW diesel engine was evaluated using different blends of jatropha oil (JO) with diesel (HSD) as a fuel. The fuel samples were prepared by blending jatropha oil with diesel in the ratio of 0:100, 10:90, 20:80, 30:70 and 40:60. The performance parameters of diesel engines such as specific fuel consumption, exhaust gas temperature and brake thermal efficiency and exhaust emission parameters such as carbon monoxide, nitrogen oxides, smoke opacity and noise levels were measured and analysed at different engine loadings to optimise blending ratio of jatropha oil with high speed diesel (HSD) for use as an alternate fuel for compression ignition engines. The specific fuel consumption of the engine increased with increase with percentage of jatropha oil blended with diesel. The specific fuel consumption of the engine increased in the range of 1.56-14.4 % and 4.76-20.63 % at 80 and 100 % load, respectively, as compared to HSD with blending of jatropha oil in diesel from 10 to 40 %. The increase in blending of jatropha oil (JO) in HSD from 0 to 40 % increased exhaust gas temperature from 4.0 to 8.8. The highest brake thermal efficiency of the engine was 26.9 % at full load on the engine with HSD and it decreased with increase in percentage of jatropha oil in fuel blends at different loads on engine. The decrease in brake thermal efficiency was low (1.5 %) up to 20 % blending of jatropha oil in HSD and it increased sharply beyond 20 % blending.

The blending of jatropha oil with HSD increased the CO emission. The increase in CO emission at 100 % load on the engine was 3.6, 4.2, 79, 79.6 % for 10, 20, 30 and 40 % blending of jatropha oil in HSD. The NOx emission increased with increase in engine load for all the fuels, i.e. HSD and blended fuels. The NO_x emissions were reduced by 8.5, 28.2, 35.2 % with 20, 30 and 40 % blending of JO in HSD. The smoke opacity of the engine at lower loads decreased with increase with jatropha oil blending in HSD. There was no significant effect of blending of jatropha oil into diesel on noise levels of the engine at lower loads. The engine noise increased from 91.2 to 97.1 dB (A) with increase in load on the engine. Therefore, it may be

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concluded that the performance of CI engine with 10 % blending of jatropha oil was quite close to diesel fuel and performed better from exhaust emission norms.

Introduction

The petroleum reserves in the world are rapidly depleting. There are various political and economical reasons that affect the petroleum supply. Moreover, consumption and prices of petroleum are increasing day by day. All these factors lead to search for alternatives to petroleum based fuels specially diesel and petrol. Though there are a wide variety of alternative fuels available, the research has not yet provided the right renewable fuel to replace or substitute diesel. Thermodynamic tests based on engine performance have established feasibility of using a variety of alternative fuels such as compressed natural gas (CNG), hydrogen, alcohols, bio-gas, bio-mass and edible and non-edible vegetable oils. They can cut foreign exchange and contribute towards protection of the earth from the threat of environmental degradation. Vegetable oil based fuels can be utilized as diesel alternatives for compression ignition engines as their properties are close to diesel. Many vegetable oils such as karanj oil, rice bran oil, soybean oil, rapeseed oil, mahua oil, linseed oil have been used to run internal combustion engines either directly by blending with diesel or after esterification (Murugesan, 2009). Vegetable oils offer the advantage of freely mixing with alcohols and these blends can be used in existing diesel engines without any modifications. On volume basis, heat content of vegetable oils is almost comparable to diesel oil owing to greater densities. Heat value decreases with increasing unsaturation as a result of fewer hydrogen atoms (Raju *et al*., 2000).

Jatropha curcas oil often known as "Ratanjot Tel" in the north India, is also known as wild castor oil. The jatropha oil has various advantages and the plant can be grown in wasteland. In India, it is found in semi wild conditions and grown in fields. It is a large shrub with thick branches and numerous large leaves attaining a height of 3-4 m in three years. The plants are not browsed by goats or cattle. The jatropha plant has few insect or fungal pests and is not a host to many diseases that attack agricultural plants. It requires little irrigation and can be successfully cultivated both in irrigated and rainfed conditions. It can tolerate high temperatures and can be grown very well under low fertility and moisture conditions. The plants grow quickly forming a thick bushy fence in a short period of 6-9 months. Its seeds resemble castor seeds in shape, but are smaller and brown in colour. It is obtained from the dried ripe seed of the flax plant. Its viscosity is more than most of the vegetable oils. Jatropha curcas appears to be a new and feasible alternative fuel for diesel.

Some investigations were carried out on use of esters prepared from blends of a variety of vegetable oils like jatropha oil, karanj oil, rice bran oil, rapeseed oil and soybean oil. The esters of these oils have been successfully used in internal combustion engines and their performance has been reviewed by many researchers. The technology for preparation of esters has not been standardised, resulting in the variation of properties of esters for use in internal combustion engines. The by-products generated in the process are also not being effectively utilized resulting in high cost of production of esters. Considering the various advantages of jatropha oil as an alternative fuel, this investigation was carried out to evaluate the performance of a small capacity diesel engine using different blends of jatropha oil with diesel as fuel.

Materials and Methods

Instrumentation System

A 3.75 kW naturally aspirated, water cooled, 4 stroke, direct injection compression ignition engine (Fieldmarshal make) was used for the experiment. The SAJ test plant with digital timer and AG-20 series eddy current dynamometer were used for carrying out experiments with the high speed diesel (HSD) and jatropha oil blended with HSD. The volumetric fuel measuring unit (SAJ, Pune make) used in the study consisted of a vertical glass measuring tube or pipette fitted with a top manifold and bottom manifold assembly. The SFV-75 model was used for measurement of fuel consumption of the engine with in the range of 25-75 cm^3.

A four digit digital R.P.M. indicator, Crystal 1,030 (SAJ, Pune make), was used for measurement of engine speeds. Digital indictors and indicating controllers model Analogic-101 were used for continuous process control applications for measuring temperatures. The pressure was measured with a pressure indicator. The pressure logic consisted of power supply card, the CPU card and the display and keyboard card.

A microprocessor controlled, multi-functional portable Sensonic 5,000 flue gas analyzer was used for measurement of exhaust gas emissions. The equipment had electrochemical sensors. Together with sensors and catalytic devices it was used for measurement of gas concentration (CO and NO_x) using the electrochemical cells. An AVL 437 smoke meter was used for measuring smoke opacity of exhaust emissions. The modular precision sound level meter type 2,231 (Bruel & Kjaer, Denmark) was used to measure the sound level of the engine.

Preparation of Fuel Blends and Engine Test Set Up

Different blends of high speed

diesel (HSD) and jatropha oil (JO) were premixed on a volume basis and stored in separate auxiliary fuel tanks. Pure high speed diesel (HSD) and four jatropha oil blends were used: 100 % high speed diesel (HSD), 90 % diesel with 10 % jatropha oil (JO 10), 80 % diesel with 20 % jatropha oil (JO 20), 70 % diesel with 30 % jatropha oil (JO 30) and 60 % diesel with 40 % jatropha oil (JO 40). The substitution of jatropha oil with diesel beyond 40 % was not done in this study because it was shown during the trial run that at 50 % blending of jatropha oil with HSD that the engine operation was not smooth and engine sound was abnormal.

The selected 3.75 kW water cooled, 4 stroke, CI engine was loaded with an AG 20 eddy current dynamometer during the experiment. The experimental test set up for the study is shown in **Fig. 1**. Temperature sensors were placed at the engine water inlet, engine water outlet, engine air inlet, engine air exhaust, dynamometer water inlet, dynamometer water outlet, engine lubricating oil at crank case, engine fuel in cylinder. The temperature values were shown on the display board. The ambient dry bulb temperature and wet bulb temperature were also recorded during the experiment. The atmospheric pressure was measured with a pressure indicator and shown on the display panel.

Evaluation of Performance of the Engine Using Blended Fuel

Torque measurement was made with a strain gauge load cell along with speed measurement from a shaft mounted with a sixty tooth wheel and magnetic pick-up. The engine was tested for performance evaluation using various fuel blends at loads of 0, 20, 40, 60, 80 and 100 % with corresponding power of 0, 0.74, 1.48, 2.22, 2.96 and 3.75 kW, respectively. The various performance parameters such as power, fuel consumption, temperatures (engine inlet and outlet, exhaust gas, dynamometer water inlet and outlet, engine fuel and lubricating oil), barometric pressure, exhaust emissions (CO and NO_x), smoke opacity and sound levels were measured and recorded.

Initially, the engine was run on HSD. The power was adjusted for minimum load and shown on the display board. The total time elapsed for 25 ml fuel consumption was shown on the display screen. These steps were repeated for 20, 40, 60, 80 and 100 % of rated load. The experimental trials following the steps were repeated for fuel blends ratio of JO : HSD for 10 : 90, 20 : 80, 30 : 70 and 40 : 60.

The fuel consumption (l/h) was calculated by the formula

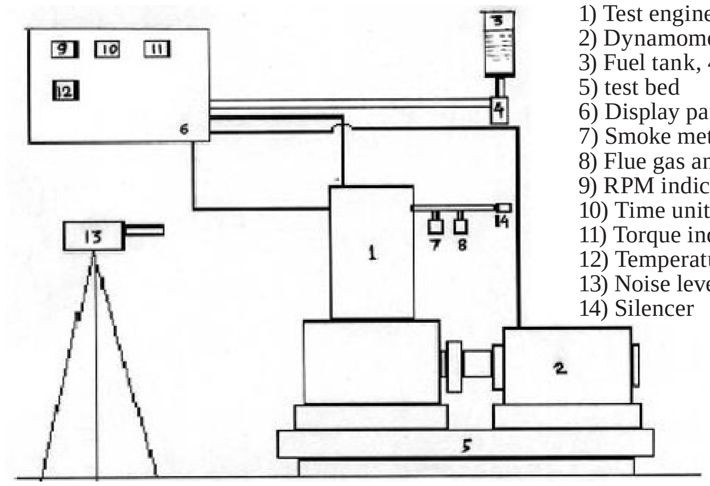
Fuel consumption = (25 × 3600) / (time in s × 1000)

The specific fuel consumption (SFC) in kg/kWh was calculated by the equation

SFC = (fuel consumed in kg × 3600) / (load in kW × time in s)

The brake thermal efficiency (%)

Fig. 1 Experimental set up for performance evaluation of 3.75 kW diesel engine with fuel blends



1) Test engine
2) Dynamometer
3) Fuel tank, 4) Pipette
5) test bed
6) Display panel
7) Smoke meter
8) Flue gas analyser
9) RPM indicator
10) Time unit indicator
11) Torque indicator
12) Temperature indicator
13) Noise level meter
14) Silencer

Fig. 2 Variation of fuel consumption with brake power of engine for various fuel blends

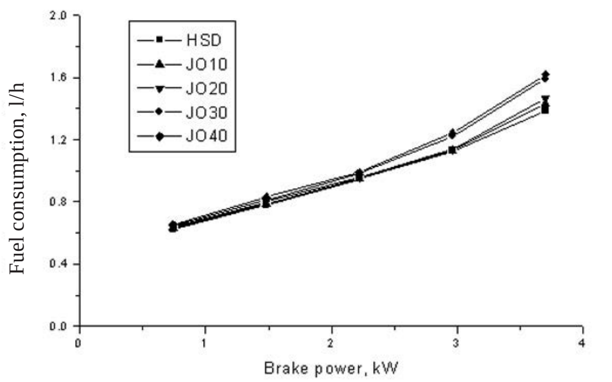


Fig. 3 Variation of specific fuel consumption with brake power of engine for various fuel blends

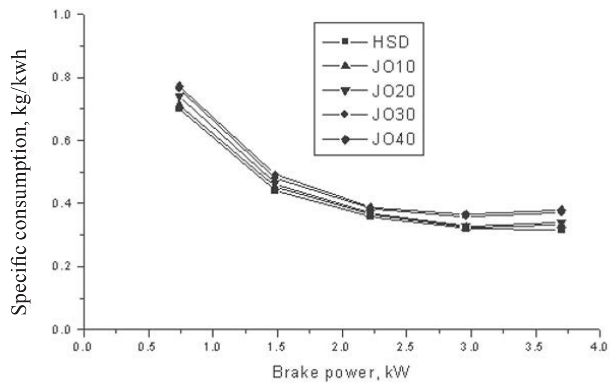


Table. 1 Properties of jatropha oil (JO) and HSD

Property	Jatropha oil (JO)	High speed diesel (HSD)
Density, kg/m^3	920	840
Calorific value, kJ/kg	39,700	42,490
Flash point, ºC	227	61
Fire point, ºC	250	69
Cloud point, ºC	−5	−19
Viscosity, cst	49.93	4.59
S content, % mass	0.13	0.25

was calculated using the equation

η_{th} = *(brake power in kW × 100) / (calorific value of fuel × mass of fuel used)*

The sound level of the engine was measured by placement of the instrument at a height of 1.2 m above the ground and at a distance of 1 m from the engine centre for free-field conditions. The equivalent sound pressure level (dB) was recorded during each measurement trial.

Results and Discussion

The fuel properties of jatropha oil and high speed diesel (HSD) used in the study are reported in **Table 1**. This shows that viscosity of jatropha oil is around 10 times higher as compared to high speed diesel (HSD). Therefore, the jatropha oil (JO) was blended with high sped diesel (HSD) in different proportions for testing. The densities of jatropha oil and high speed diesel (HSD) were 920 and 840 kg/m^3, respectively. Thus, the blending of jatropha oil led to an increase in the density of the fuel.

The effect of blending of JO in HSD on fuel consumption of the engine is shown in the **Fig. 2**. The hourly fuel consumption of the engine increased with increase in percentage of jatropha oil (JO). For 100 % load, the fuel consumption was 1.39, 1.44, 1.47, 1.59, 1.62 l/h for HSD, JO 10, JO 20, JO 30, JO 40, respectively. The blending of jatropha oil into HSD increased fuel consumption of the engine from 1.4 to 14.5 % for 10 to 40 % blending of jatropha oil as compared with 100 % HSD fuel.

The effect of various fuel blends on the specific fuel consumption (sfc) of the engine at different brake power is shown in **Fig. 3**. In general, the specific fuel consumption of the engine increased with increase in blending of jatropha oil in HSD. The specific fuel consumption decreased as the load on engine increased. The minimum specific fuel consumption was 0.315 kg/kWh at maximum rated power of the engine with HSD fuel, whereas, it was minimum at 80 % of rated power of the engine for blended fuels. The minimum specific fuel consumption was 0.325, 0.329, 0.357 and 0.366 kg/kWh for JO 10, JO 20, JO 30 and JO 40 fuels, respectively. At full load, the specific fuel consumption of the engine was 0.330, 0.340, 0.370 and 0.380 kg/kWh for 10, 20, 30 and 40 %, respectively, blending of jatropha oil into diesel. The increase in specific fuel consumption was up to 4.8 % at 20 % blending of jatropha oil in HSD at 80 % of rated power. The variations in sfc up to 20 % blended fuels at part load and 10 % blended fuels at rated power were observed within the limit (± 5%) recommended in IS 10,000 (part IV) (Indian Standard).

The increase in specific fuel consumption of the engine at 80 % load was 1.56, 2.81, 11.56 and 14.4 % for 10, 20, 30 and 40 % blending of jatropha oil, respectively, as compared with the diesel fuel. At 100 % load, the specific fuel consumption was 0.315, 0.330, 0.340, 0.370 and 0.380 kg/kWh for HSD, JO 10, JO 20, JO 30, JO 40, respectively. The increase in the specific fuel consumption was 4.76, 7.94, 17.46, 20.63 % for 10, 20, 30 and 40 %, respectively, blending of jatropha oil with HSD at 100 % load on the engine.

The observed variation of exhaust gas temperature with brake power during the engine testing

Fig. 4 Variation of exhaust gas temperature with brake power of engine for various fuel blends

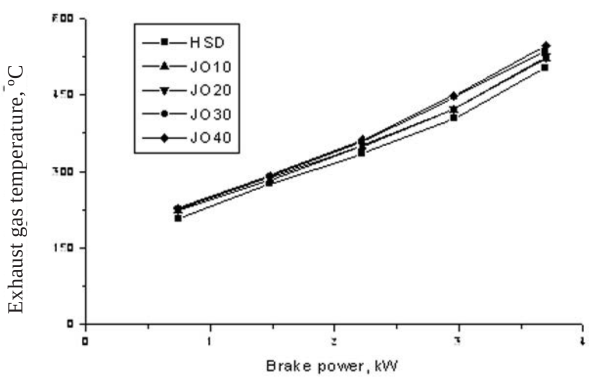
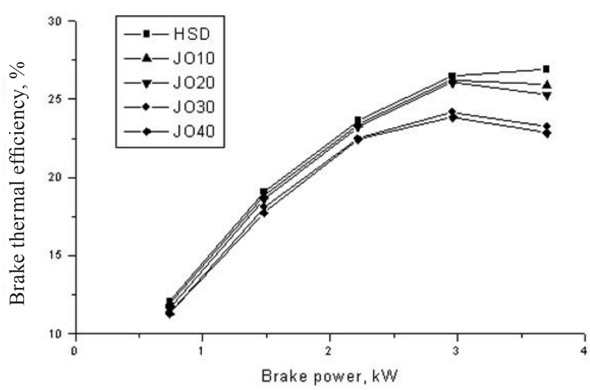


Fig. 5 Variation of brake thermal efficiency with brake power of engine for various fuel blends



with blended fuels is shown in **Fig. 4**. The exhaust gas temperature increased with the increase of engine load for all the fuel blends. A similar trend was observed by Murugesan *et al.* (2009) for vegetable oils. The exhaust gas temperatures at rated engine power were 502, 522, 525, 536, 546 ºC for HSD, JO 10, JO 20, JO 30, JO 40 fuel blends, respectively. This indicated that the increase in blending of jatropha oil (JO) in HSD from 0 to 40 % increased exhaust gas temperature of the engine from 4.0 to 8.8 %. The increase in exhaust gas temperature with increase in JO blending indicated delay in combustion due to change in fuel properties required to enhance injection timing.

The variation of brake thermal efficiency with brake power of the engine for different fuel blends is shown in **Fig. 5**. The highest brake thermal efficiency of the engine was 26.9 % at full load on the engine with HSD and it decreased with increase in percentage of jatropha oil in fuel blends at all loads on the engine. A similar trend was observed by Pramanik (2003) and Singh *et al.* (2008). For blended fuels, the brake thermal efficiency of the engine increased with increase in engine load up to 80 % and then decreased beyond 80 % load on engine. The maximum brake thermal efficiency of the engine was 26.2, 26.1, 24.2, 23.8 % for JO 10, JO 20, JO 30, JO 40 fuel blends, respectively. The decrease in efficiency was 0.87, 1.47, 8.65, 9.95 % for JO 10, JO 20, JO 30, JO 40 fuel blends, respectively. The decrease in brake thermal efficiency was low (1.5 %) up to 20 % blending of jatropha oil in HSD and it increased sharply beyond 20 % blending. The thermal efficiency at full load on the engine was 25.9, 25.3, 23.3, and 22.9 % and decreased by 3.64, 5.98, 13.50, and 15.10 % for JO 10, JO 20, JO 30, JO 40 fuel blends, respectively as compared to HSD fuel.

In general, the blending of jatropha oil with HSD increased the CO emission (**Fig. 6**). The CO emission at 100 % load on the engine was 4,307, 4,462, 4,486, 7,725, 7,735 ppm for HSD, JO 10, JO 20, JO 30, JO 40 fuels, respectively, and the increase in emission was 3.6, 4.2, 79, 79.6 % with 10, 20, 30 and 40 % blending of jatropha oil in HSD. The minimum CO emission level was observed with diesel fuel and maximum with 40 % jatropha oil blended fuel. The maximum CO emission was at maximum power on the engine with all the fuels. Blending of jatropha oil into HSD beyond 20 % increased CO emission sharply from the engine.

The NO_x emission increased with increase in engine load for all the fuels, i.e. HSD and blended fuels (**Fig. 7**). NO_x emissions of the engine reduced with increase in blending percentage of jatropha oil in HSD and were lower than those with diesel fuel. The reduction of the NO_x emission was possibly due to lower calorific value of blended fuels. The low NO_x emission could also have been due to low temperature during combustion. A similar trend was observed by Murugesan *et al.* (2009). The NO_x emissions of the engine at full load were 655, 559, 439 and 396 ppm for HSD, JO 20, JO 30, JO 40 fuel blends, respectively, and were reduced by 8.51, 28.15, 35.2 % with 20, 30 and 40 % blending of JO in HSD.

The effect of jatropha oil blending on smoke opacity of the engine is shown in **Fig. 8**. There was a decrease in smoke opacity with increase in jatropha oil blending in HSD at lower loads on the engine. There was an increase in smoke opacity with increase in JO blending at higher loads. At 80 % load, the smoke opacity of the engine was 51.6, 51.8, 52.0, 58.6, 64.6 % for HSD, JO 10, JO 20, JO 30, JO 40 fuels, respectively, with increase of 0.4, 0.8, 13.6, 25.2 %, respectively, as compared to HSD. At 100 % load, it was 91.0, 92.0, 93.3, 98.6, 99.4 % for HSD, 10, 20, 30, 40 % blend, respectively, with an increase of 1.11, 2.53, 8.35, 9.23 % as compared to HSD.

The variation of equivalent sound pressure level (Leq) with brake power of the engine for various fuel blends is shown in **Fig. 9**. There was no significant effect of blending of jatropha oil into diesel on noise lev-

Fig. 6 Variation of carbon monoxide emission with brake power of engine for various fuel blends

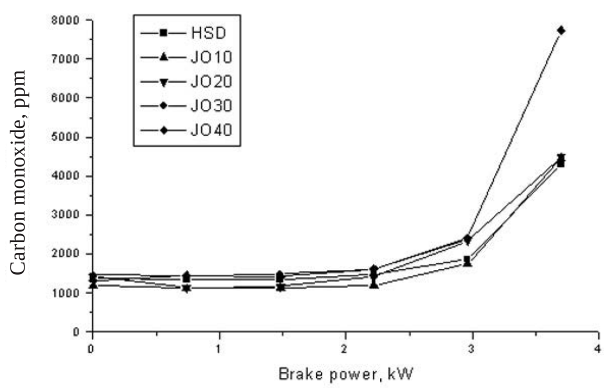
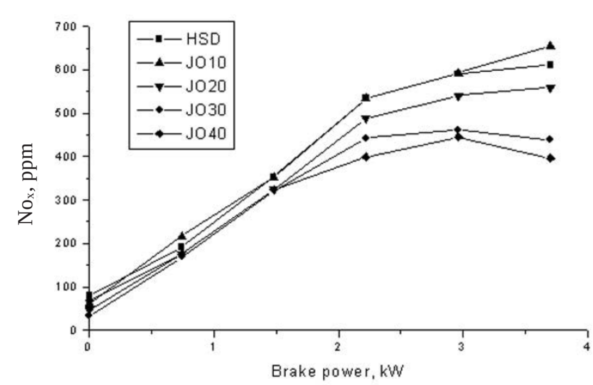


Fig. 7 Variation of NOx emission with brake power of engine for various fuel blends



els of the engine at lower loads. The engine noise increased from 91.2 to 97.1 dB (A) with increase in load on the engine. The sound levels of the engine at 80 % load were 94.7, 94.9, 95.6, 96.0, 96.5 dB (A) for HSD, JO 10, JO 20, JO 30, JO 40 fuel blends, respectively. Similarly, at 100 % load on the engine, it was 97.1, 97.3, 97.4, 98.5, 98.6 dB (A) for HSD, JO 10, JO 20, JO 30, JO 40 fuel blends, respectively.

Finally, it may be concluded that the performance of CI engine with 10 % blending of jatropha oil was quite close to diesel fuel and it also performed better from exhaust emission norms.

Conclusions

The following major conclusions were drawn from the study.

1. The specific fuel consumption of the engine increased with increase with the percentage of jatropha oil blending with HSD. The specific fuel consumption decreased with increase in load on engine up to 80 % and then increased at 100 % load on engine for different blended fuels.
2. The exhaust gas temperature increased with the increase in percentage blending of jatropha oil in the fuel. The maximum exhaust gas temperature at rated power was 546 ºC with 40 % blending of jatropha oil as compared to 503 ºC for HSD.
3. The brake thermal efficiency of the engine decreased with increase in percentage of jatropha oil blending into diesel. The maximum thermal efficiency of the engine was 26.9 % at rated power and 26.2 % at 80 % of the rated power for diesel fuel and 10 % jatropha oil blended fuel, respectively.
4. The carbon monoxide levels in the exhaust gas of the engine decreased up to 10 % blending of jatropha oil with HSD and then increased with further increase in jatropha oil blending percentage.
5. The emission of oxides of nitrogen in exhaust gases increased with the increase in engine load. The NO_x emission in exhaust gases decreased from 8.51 to 35.2 % as compared to HSD with 20 to 40 % increase in jatropha oil blending in diesel.
6. The smoke opacity of exhaust gases increased with the increase in load on the engine for blended fuels. The smoke opacity of exhaust gases was lower at 20-60 % load on the engine with blended fuels as compared to HSD and higher at 80 and 100 % load on the engine.
7. The noise level increased with increase in load for both HSD and blended fuels. There was no significant difference in noise levels of the engine with blended fuels at lower loads.
8. The CI engine can be operated without affecting the performance of the engine with 10 % blending of jatropha oil with high speed diesel.

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Fig. 8 Variation of smoke opacity with brake power of engine for various fuel blends

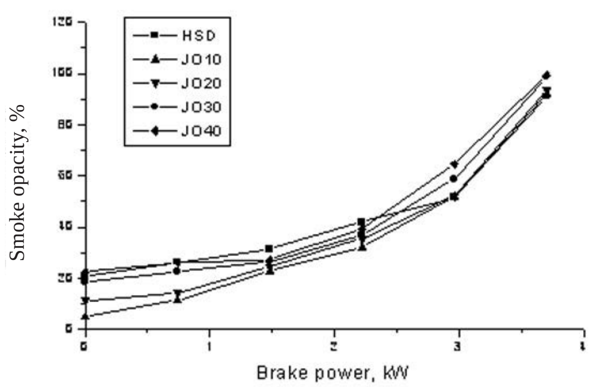
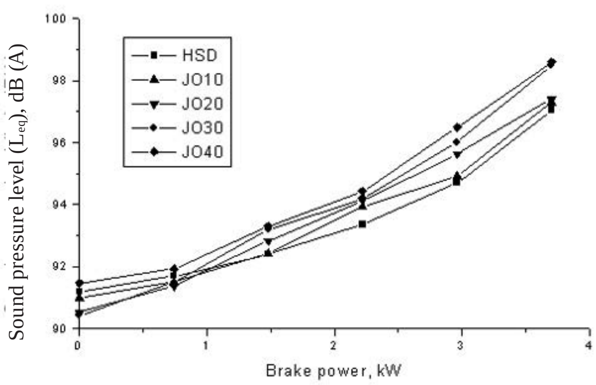


Fig. 9 Variation of equivalent sound pressure level with brake power of engine for various fuel blends



Engineering Studies on the Performance of Paddy and Rice Separator

by
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Abstract

The performance of paddy and rice separator was theoretically and experimentally investigated as a function of change in separator sieve angle, paddy moisture content and feeding ratios. The separator performance was evaluated in terms of separator productivity, crack percentage, separator efficiency and energy requirements. The theoretical analyses revealed that the optimum sieve speed of 0.5 m/sec. (200 rpm) is recommended to prevent riding of material on the sieve surface. The experimental results revealed that the separator performance was in the optimum region under the following conditions:

- Sieve angle of 15 deg.
- Paddy moisture content of 14 %.
- Feeding ratio of 0.80.

Introduction

Agricultural policy depends on the successful application of technology through mechanizing the agricultural processes of strategic crops. One of the most important economical crops is rice as it contributes to the international income in addition to the local consumption in feeding and industrial aspects. Thus, up-to-date technology through the different stages of rice production is an important area to be examined. One of these stages is rice milling. Major operations carried out in a modern rice mill are cleaning, husking, separation of paddy and rice, whitening and grading. Separation of paddy and rice on the oscillating type separator takes place due to the difference in specific gravity and surface characteristics of paddy and rice. The separating machine separates the product of the husking operation into three parts: brown rice, paddy and a mixture of the two. Brown rice is fed to the whitening machine, paddy is sent back for husking and the mixture is returned to the separator. El-Raie (1987) studied the technical parameters of the flat sieve such as dimensions of sieve, speed of the crankshaft of the screening unit and the ideal distribution of the holes on the sieve sheet. This information is very important in designing and developing the specific machines suitable for Egypt. Ahmed (1988) mentioned that the slope of the separating sieve is a controlling factor of the effectiveness of separation. This effectiveness improves significantly as the sieve is also a controlling factor in determining the maximum possible feed rates to the winnowing machine. As the sieve slope increases, the maximum possible feed rates, provided the other factors are kept the same, are greatly reduced. He also found that the maximum efficiency values of separation are quite different depending on the number of oscillations per minute. Ahmed *et al*. (1993) developed a winnowing machine designed in such a way as to change the parameters affecting the separation effectiveness such as the sieve oscillation, amplitude, sieve angle and feed rate for using a threshed wheat crop by the locally made stationary thresher. They added that, a separation effectiveness of 97 % was obtained at sieve oscillations of 500 cycle/min, sieve angle of 2 deg and feed rate of 30 kg/h cm. at a grain/straw ratio of 1:3.

Amin (2003) studied some engineering parameters affecting cleaning and separating efficiency for each machine type (vibratory and rotary machines) such as type of motions (vibrating or rotary speeds), cells shape (rectangular, square and circle), position of rectangular cell (parallel or perpendicular with speed direction), sieve inclination and sieving time. He found that the efficiency increased by increasing sieving time and oscillating and rotary speed. Awady *et al.* (2003) developed and tested a separating and cleaning machine for winnowing rice crop for better efficiency and reduced loss. The cleaning machine consisted of frame, grain hopper, oscillating dual-screen assembly, a centrifugal blower and electric motor. The eccentric and support linkages of screen assembly caused it to oscillate, moving the grain over the flat screen. During the operation, grain was loaded onto the hopper and fed into the oscillating screen through the bottom opening and regulated by the slide gate. The upper screen separated the impurities that were bigger than the grain, and the lower screen separated those that were smaller and dust. El-Sahrigi *et al.* (2004) designed and constructed a separating and cleaning unit able to separate various types of medicinal and aromatic seeds and their associated foreign matter by making simple adjustments according to the type of seeds, their physical properties and associated impurities. They also tested the performance of a cleaning unit under the following main factors: the frequency of the sieve unit, feed rate, air velocity and slope of the sieve unit. The maximum seed cleanliness and separation effectiveness were 99.01 and 89.75 %, respectively, and were obtained at a frequency of 10.50 Hz, feed rate of 300 kg/h, slope of 13 deg, and air velocity of 3.2 m/s.

This research has covered theoretical and experimental analyses of some engineering and operational parameters affecting the performance of a paddy and rice separator for the purpose of maximizing separator efficiency and minimizing energy requirements.

Materials and Methods

The main experiments were carried out at Zagazig Milling Company, Sharkia province, to study the effect of some engineering and operating parameters on the performance of a paddy and rice separator.

The Rice Variety

Rice Giza 172 (short grain) variety was used. The physical and mechanical properties of the paddy and brown rice are shown in **Table 1**.

The Separator

The new SATAKE paddy separator Ps 120E was used with the following specifications:

Output capacity:

long grains 5.2 -5.6 ton/h

short grains 6.0-7.2 ton/h

Required power: 0.75 kW

The Instruments

- A repose angle meter was used for measuring the angle between the base and the slope of the grains.
- A digital instrument was used for measuring the friction angle of the grains on metal sheet surface with an accuracy of 0.01 degree.
- An electric digital balance was used for measuring the mass of grains samples with an accuracy of 0.1 mg.
- A digital power meter was used to measure the required power for operating the separator.

Separator Adjustment

A vibrated sieve was used to remove paddy from brown rice. Agitation of the separating sieve resulted in displacement of the paddy and rice over its surface. The paddy and rice were agitated to obtain optimal separation. The paddy and rice were uniformly distributed over the sieve surface and moved towards the delivery end of the sieve. The sieve was agitated by multiple system linkage kinematic characteristics of linear motion of the driving link and the crank-connecting rod mechanism for small values of r/L (crank shaft length/connecting rod length). The following relationships apply (Klenin *et al.*, 1985):

$X = r\,(1 - \cos \omega t)$

$\dot{X} = \omega r \sin \omega t$

$\ddot{X} = \omega^2 r \cos \omega t,$

where X = Instantaneous displacement, cm

$\dot{X}$ = Motion velocity, cm/s

$\ddot{X}$ = Acceleration of motion, cm/s^2

ω = Angular velocity, rad/s

R = crank shaft length, cm

The Following are Forces Acting on the Paddy and Rice Lying on a Sieve:

1. W is the force due to the weight of the paddy and rice directed downward.
2. F_i is the inertia force acting in a direction opposite to that of the mass acceleration force. The magnitude of the force F_i is obtained as follows:

   $$F_i = m\,\ddot{X} = m\,\omega^2 r \cos \omega t,$$

   where m = mass of paddy and rice.
3. F_f is the friction force between the paddy and rice and the sieve surface acting in a direction opposite to motion direction.
4. R is the reaction force of the working surface on the rice acting in a direction normal to the surface. The sieve is set horizontal or inclined to the horizontal plane with the angle of inclination selected from the condition:

   $\alpha \leq \phi$, where

   α is the Angle of sieve with the horizontal, and

   ϕ is the friction angle between the mixture and the sieve surface.

According to the conditions given above, the material will not slide over the sieve when it is stationary. When the sieve is agitated at a par-

Table 1 Physical and mechanical properties of paddy and brown rice

Rice	Length	Width	Thickness, mm	Mass of 1,000 kernels g	Repose angle, deg.	Friction angle, degree(θ)	Coefficient of friction
Paddy	7.70	3.22	2.45	24.90	40-45	48	1.11
Brown	5.57	2.92	2.25	21.40	3-38	40	0.84

ticular frequency and amplitude, a motion is imparted to the material relative to the sieve surface. The possible types of motion of the material are only sliding motion over the sieve towards the delivery end, and in the reverse direction or loss of all contact between the material and the sieve surface.

Motion of Material over the Sieve Surface at the Delivery End from A to B

Motion at the delivery end is possible when the resultant of all forces acting the material is greater than the friction force (**Fig. 1**) that is:

$W \sin \alpha + F_I \cos \alpha \geq F_f$

$F_f = R \tan ø = \mu R$, where

F_f is the the friction force

ø is the friction angle

μ is the coefficient of friction, and

R is the normal force

To determine force R projecting all the forces in a direction normal to the sieve:

$R = W \cos \alpha - F_i \sin \alpha$

Then motion of the seed at the exit may be expressed by the following inequality:

$W \sin \alpha + F_i \cos \alpha \geq \mu W \cos \alpha - \mu F_i \sin \alpha$ or

$mg \sin \alpha + m \omega^2 r \cos \alpha \geq \mu mg \cos \alpha - \mu m \omega^2 r \sin \alpha$

$$\omega_1 = \sqrt{\frac{g(\mu \cos\alpha - \sin\alpha}{r(\cos\alpha + \mu \sin\alpha)}} \quad \text{or}$$

$$N_1 = \frac{60}{2\pi}\sqrt{\frac{g(\mu\cos\alpha - \sin\alpha)}{r(\cos\alpha + \mu\sin\alpha)}}$$

Sliding motion of the material up and down the sieve surface from B to A:

Fig. 2 shows that the movement of material from B to A is possible when*:*

$F_i \cos \alpha - W \sin \alpha \geq F_f$, where

$R = W \cos \alpha + F_I \sin \alpha$

Then the motion of material, in this case, may be expressed by the following inequality:

$F_i \cos \alpha - W \sin \alpha \geq \mu W \cos \alpha + \mu F_i \sin \alpha$

$m \omega^2 r \cos \alpha - mg \sin \alpha \geq \mu mg \cos \alpha + \mu m \omega^2 r \sin \alpha$

$$\omega_2 = \sqrt{\frac{g(\mu \cos\alpha + \sin\alpha}{r(\cos\alpha - \mu \sin\alpha}} \quad \text{rad/s}$$

$$N_2 = \frac{60}{2\pi}\sqrt{\frac{g(\mu\cos\alpha + \sin\alpha)}{r(\cos\alpha - \mu\sin\alpha)}} \quad \text{rpm}$$

Fig. 1 Motion of material over the sieve surface at the delivery end from A to B

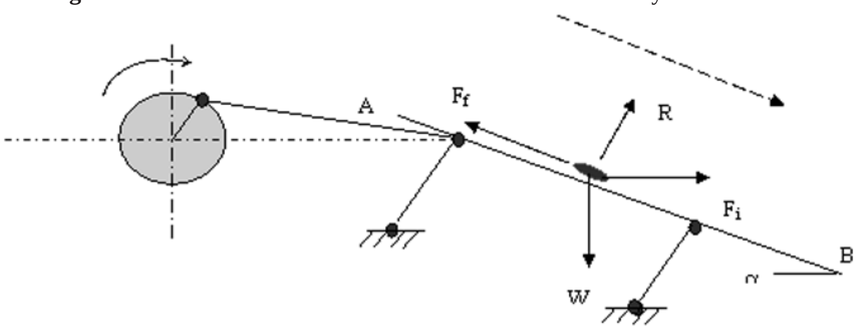
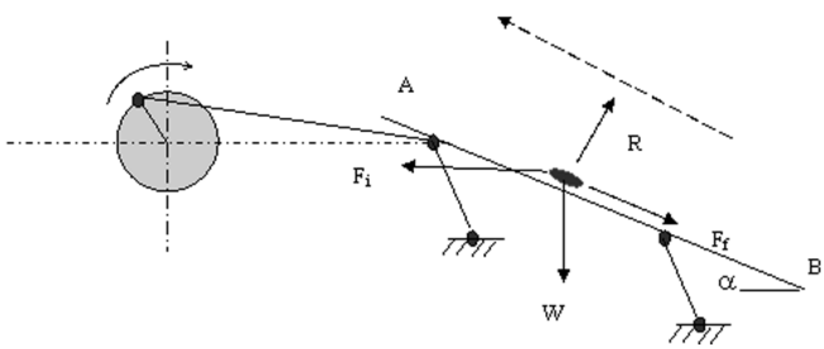


Fig. 2 Sliding motion of the material up and down the sieve surface from B to A



Riding of material on the sieve surface:

The motion of material in this case may be expressed by the following inequality:

$W \sin \alpha + F_i \cos \alpha \geq F_f$

But in this case $R = 0$

$\therefore F_f = 0$

$F_i \cos \alpha \geq - W \sin \alpha$

$m \omega^2 r \cos \alpha \geq - mg \sin \alpha$

$$\omega_3 = \sqrt{\frac{-g\sin\alpha}{r\cos\alpha}}, rad/s$$

$$N_3 = \frac{60}{2\pi}\sqrt{\frac{-g\sin\alpha}{r\cos\alpha}}, rpm$$

It follows from these equations that the mass of material moves along the sieve toward the exit ($N > N_1$) and to the opposite direction ($N > N_2$).

Sieving and separating are more successful under the following conditions:

$N_3 > N > N_2$ *and* $N_2 > N_I$

where N is the optimum sieve speed.

Under the above conditions and by using the previous equations, the separator was adjusted so that the angle of sieve with the horizontal was determined to be less than the friction angle between the material and the sieve surface $\alpha \leq ø$. Therefore, $\alpha \leq 40$ deg.

Additionally, optimum sieve speed was calculated to be:

N = 200 rpm = (3.33 Hz); corresponding to a linear sieve speed of 0.5 m/s.

Experimental Procedures

The experiment included four sieve angles with the horizontal (10, 12, 15 and 20 deg); four paddy moisture contents (10, 12, 14 and 16 %) and five feeding rate ratios (0.50, 0.60, 0.70, 0.80, and 0.90).

The feeding rate ratio was the ratio of the actual feeding rate, Q, to (Q_{max}) the maximum feeding rate (7,200 kg/h).

Values Calculated from the Data:

1) *Separator productivity*, in Mg/h, was the ratio of the mass of brown rice (Q_{br}), in Mg, to the time (t)

consumed in the separating operation, h.

2) *Crack percentage* was the ratio of the mass of cracked grains in the brown rice sample, in g, to the mass (T) of the total sample, in g.

3) *Separator efficiency* of the paddy and rice separator was measured using the following equation (Modi 1972):

$$\zeta s = \frac{x_b(x_s - x_p)(x_b - x_s)(1 - x_p)}{x_s(1 - x_s)(x_b - x_p)^2}$$

where x_b is the weight of brown rice fraction in the collection from the brown rice outlet of the separator;

x_p is the weight of brown rice fraction in the collection from the paddy outlet of the separator;

x_s is the weight of brown rice fraction in the sum of the collection from the brown rice outlet and paddy outlet of the separator.

4) *Energy requirements* were calculated by the following equation:

Energy requirements, *Wh/Mg = Required power (W) / Separator productivity (Mg/h)*

The required power was measured using the digital power meter.

Results and Discussion

Factors affecting Separator Productivity

Separator productivity is significantly affected by sieve angle, paddy moisture content and feeding ratio.

Fig. 3 shows that increasing sieve angle from 10 to 15 degrees, measured for feeding ratios of 0.5, 0.6, 0.7, 0.8 and 0.9, increased separator productivity by 16.02, 16.29, 37.01, 38.34 and 29.78 % at constant paddy moisture content of 14 %. For further increase in sieve angle from 15 to 20 degrees, separator productivity was decreased by 3.84, 7.07, 24.53, 24.47 and 17.20 % under the same previous conditions. The increase in productivity by increasing sieve angle from 10 to 15 deg was attributed to the uniform sliding of brown rice towards brown rice outlet. Decrease in productivity by increasing sieve angle from 15 to 20 deg was attributed the fast sliding of brown rice towards the paddy entrance.

Fig. 4 shows that increasing moisture content from 10 to 16 %, measured at separator ratios of 0.5, 0.6, 0.7, 0.8 and 0.9, decreased separator productivity by 11.76, 13.53, 13.41, 4.53 and 3.15 % at a constant sieve angle of 15 deg. This was due to the heavy weight of high moisture content material, which caused brown rice to fall into the outlet of paddy and mixture.

Figs. 4 and **5** show that increasing feeding ratio from 0.5 to 0.9 increased separator productivity by 68.54 % at a constant paddy moisture content of 14 % and constant sieve angle of 15 deg. This was explained by the fact that, when feed-

Fig. 3 Effect of sieve angle on the separator productivity at different feeding ratios and a constant moisture content of 14 %

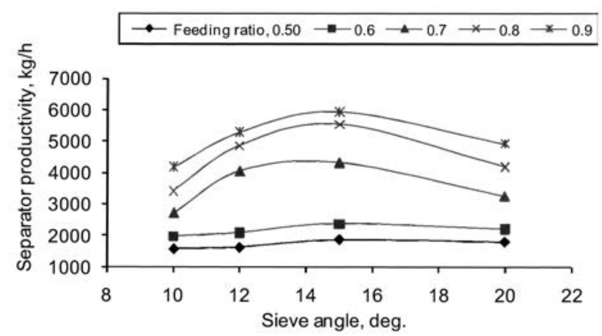


Fig. 4 Effect of paddy moisture content on the separator productivity at different feeding ratios and a constant sieve angle of 15 deg

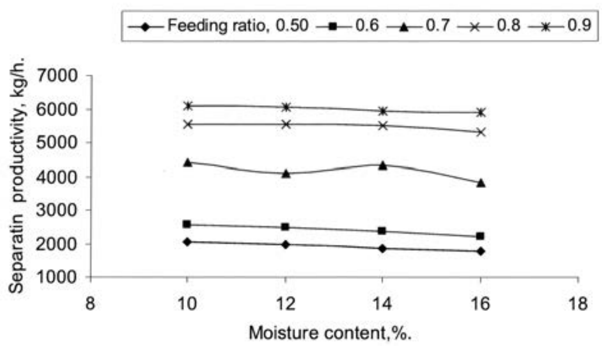


Fig. 5 Effect of sieve angle on the crack percentage at different feeding ratios (constant moisture content 14 %)

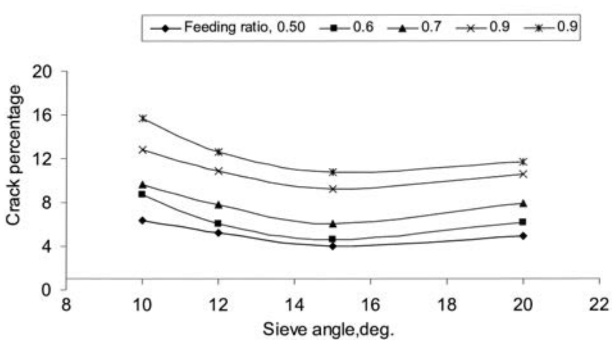
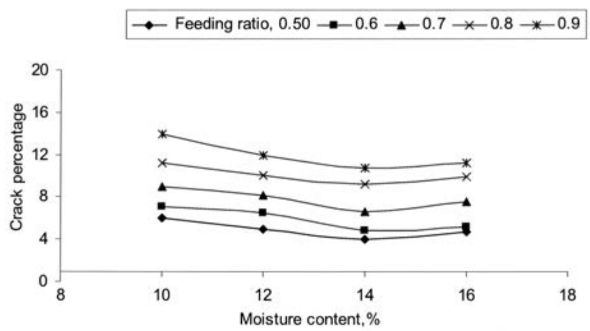


Fig. 6 Effect of paddy moisture content on the crack percentage at different feeding ratios (sieve angle, 15 deg.)



ing ratio increased, material covered the surface of the sieve uniformly which give smooth movement of material towards their outlets.

Factors Affecting Separator Crack Percentage

Representative crack percentage values versus both sieve angle and paddy moisture content are given through the various feeding ratios in **Figs. 5** and **6**.

Fig. 6 shows that increasing sieve angle from 10 to 15 degrees, measured for feeding ratios of 0.5, 0.6, 0.7, 0.8 and 0.9, decreased separator crack percentage by 37.5, 47.12, 36.45, 28.12 and 31.21 % at a constant paddy moisture content of 14%. For further increase in sieve angle from 15 to 20 degrees, separator crack percentage increased by 18.36, 25.80, 22.78, 13.20 and 7.69 % under the same previous conditions. Any increase or decrease in sieve angle more or less than 15 deg. tended to increase crack percentage due to the non uniform movement of materials on the surface of sieves.

Fig. 6 also shows that increasing moisture content from 10 to 16 %, measured at separator ratios of 0.5, 0.6, 0.7, 0.8 and 0.9, decreased separator crack percentage by 21.66, 26.76, 15.55, 11.60 and 20.00 % at a constant sieve angle of 15 deg. The decrease in brown rice moisture content tended to increase crack percentage because dry grains are more sensitive to impact with each other.

Figs. 5 and **6** show that increasing feeding ratio from 0.5 to 0.9 increased separator crack percentage by 62.96 % at a constant paddy moisture content of 14 % and constant sieve angle of 15 deg. This was due to the high density of materials on the sieve surface that increased pressure and impact forces between grains.

Factors Affecting Separator Efficiency

The most critical factors affecting separator efficiency were sieve angle, paddy moisture content and feeding ratio as shown in **Figs. 7** and **8**.

Fig. 7 shows that increasing sieve angle from 10 to 15 degrees, measured for feeding ratios of 0.5, 0.6, 0.7, 0.8 and 0.9, increased separator efficiency by 16.01, 16.27, 37.01, 38.33 and 29.78 % at constant paddy moisture content of 14 %. For further increase in sieve angle from 15 to 20 degrees, separator efficiency decreased by 3.84, 7.02, 24.53, 24.47 and 17.20 % under the same previous conditions. Higher or lower values of sieve angle more or less than 15 deg. decreased separator efficiency because of the decrease in separator productivity in both cases.

Fig. 9 shows that increasing moisture content from 10 to 16 %, measured at separator ratios of 0.5, 0.6, 0.7, 0.8 and 0.9, decreased separator efficiency by 11.76, 13.54, 13.41, 4.52 and 3.15 % at a constant sieve angle of 15 deg. This attributed to the decrease in separator productivity by decreasing paddy moisture content.

Figs. 7 and **8** show that increasing feeding ratio from 0.5 to 0.9 increased separator efficiency by 44.08 % at constant paddy moisture content of 14 % and constant sieve angle of 15 deg. Increasing feeding ratio increased separator productivity, which in turn increased separator efficiency.

Factors Affecting Separator Energy Requirements

Separator energy requirements are more sensitive to different factors such as: sieve angle, paddy moisture content and feeding ratio.

Fig. 9 shows that increasing sieve angle from 10 to 15 degrees, measured for feeding ratios of 0.5, 0.6, 0.7, 0.8 and 0.9 decreased separator energy requirements by 20.84, 20.06, 39.76, 43.70 and 37.17 % at a constant paddy moisture content of 14 %. For any further increase in sieve angle from 15 to 20 degrees, separator energy requirements were non-significantly increased by 1.34, 3.79, 8.65, 23.25 and 15.41 % under the same previous conditions. Lower values of sieve angle less than 15 deg. increased energy due to the slow movement of paddy on the sieve surface, especially with high feeding ratios, while higher values of sieve angle, more than 15 deg, also increased energy due to the de-

Fig. 7 Effect of sieve angle on the separating efficiency at different feeding ratios (constant moisture content 14 %)

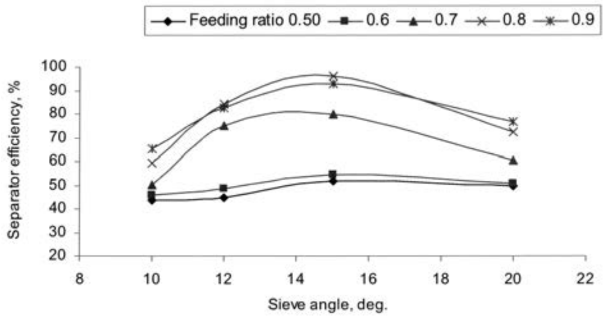
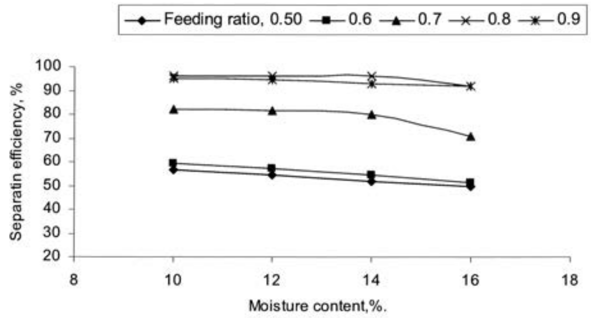


Fig. 8 Effect of paddy moisture content on the separating efficiency at different feeding ratios (sieve angle, 15 deg.)



crease in separator productivity.

Fig.10 shows that increasing moisture content from 10 to 16 %, measured at separator ratios of 0.5, 0.6, 0.7, 0.8 and 0.9, increased separator energy requirements by 17.81, 20.03, 17.21, 14.57 and 13.99 % at constant sieve angle of 15 deg. Increasing paddy moisture content increased energy because the elastic conditions of high moisture content material lead to irregular motion.

Figs. 9 and **10** show that increasing feeding ratio from 0.5 to 0.9 increased separator energy requirements by 64.87 % at constant paddy moisture content of 14 % and constant sieve angle of 15 deg. The increase in feeding ratio led to an increase in material mass on the sieve surface, which in turn tended to increased energy.

Conclusion

Optimization of paddy and rice separator variables is of great importance to minimize both cracks and energy and maximize performance efficiency.

The theoretical and experimental results revealed that, for obtaining high grade rice production, the following parameters must be used:

- Sieve speed of 0.5 m/s (200 rpm)
- Sieve angle of 15 deg.
- Paddy moisture content of 14 %
- Feeding ratio of 0.8.

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Fig. 9 Effect of sieve angle on the energy requirements at different feeding ratios at a constant moisture content of 14 %

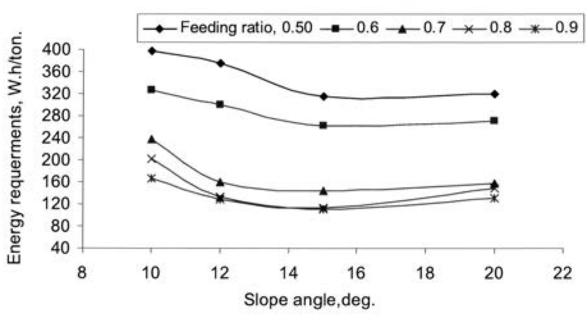
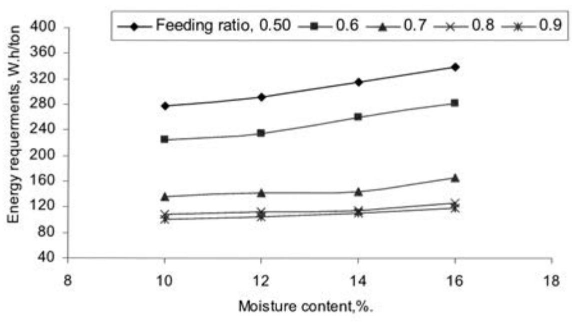


Fig. 10 Effect of paddy moisture content on the energy requirements at different feeding ratios with a sieve angle of 15 deg



ABSTRACTS

The ABSTRACTS pages is to introduce the abstracts of the article which cannot be published in whole contents owing to the limited publication space and so many contributions to AMA. The readers who wish to know the contents of the article more in detail are kindly requested to contact the authors.

945

Researching on some influence parameters on the operational quality of the sugarcane waste spreader serving the sugarcane cultivation in Tay Ninh: Nguyen, Associate Prof., Vice-Rector of Nonglam University, Ho Chi Minh City, VIET NAM; **Ton That Hai**, Researcher, same.

The Study of these influence parameters on the operational quality of the sugarcane waste spreader would aim to advance the optimum operation regime for the machine and to form a solid basis for further studying. Serving this aim, the studying would be carried out as follows: the fabrication of the machine, the study object and the means to access to the actuality of the production, the testing at site. The result of the study is to form the new plan to design an appropriate machine; to determine the regression equation to represent the operation quality of the machine (The regularity of the spread sugarcane waste, the fuel consumption) in function of the speed of the feeding conveying chain (the feeding quantity), the forwarding speed of the combine, and the rotation of the spreading rotor. From the equation, a reasonable operation regime can be applicable for the machine: The feeding chain speed at 0.02242 m/s, the forwarding of the combine at 6.95 km/h and the rotation of the spreading rotor about 125.63 RPM, the attained regularity of the spread material more than 80% and a fuel consumption of 7.5l of D. O. / ha (in the testing conditions).

949

Study of Engineering Factors Affecting the Rate of Biogas Production: Ibrahim M. M., Prof. of Agril. Engg., Faculty of Agril, Mansoura University, Nile Delta, EGYPT; **Y. M. El-Hadidi**, same; **G.I. Rashed**, Prof. of phys. Sci., Faculty of Engg., same; and **Safia M. EL-GAyAr**, Senior Research in AEnRI, Giza, EGYPT.

This work aims to study the effect of some engineering parameters such as hydraulic retention times (20, 30 and 40 days), fermentation temperature levels (30, 35 and 40 ºC), agitation rates (1,200, 1,350 and 1,480 L/h) and total solid concentrations (6, 8 and 10 %) on biogas production during anaerobic fermentation to optimize the digester performance. The results showed that the maximum gas production was obtained at 20 days hydraulic retention times (HRT), 40 ºC fermentation temperature, 1,350 L/h agitation rate and 8 % total solids concentration (TS). At these levels maximum removal rate of chemical oxygen demand (COD), bio-chemical oxygen demand (BOD), total suspended solids (TSS) and volatile suspended solids (VSS) were obtained. The obtained results showed also that the biogas fertilizer (digested sludge) can be used as a free pathogenic fertilizer with high fertilizing value or as animal fodder additive.

956

Studying, Designing, Manufacturing and Testing the Scalding Machine of the Continous Poultry Processing System: Nguyen Van Hung, Faculty of Engg., Nong Lam University, Ho Chi Minh City, VIET NAM.

The aim of this study was to research the scalding machine of the continuous poultry slaughtering system 500 birds/hour. The scalding machine was designed, manufactured, and installed in the poultry slaughtering systems that have been operating stably in Khanh Hoa province (Huong Giang company) and Binh Thuan province (Celine Thai company). Experiments were executed to determine the effects of the two factors of scalding temperature and scalding time to product quality. Result of optimization illustrated the best operational regime with the scalding temperature at 65 ºC and the scalding time at 156 seconds. This project was achieved the Vifotec price (Vietnam Fund for Supporting Technological Creations) 2007.

1025

Experimental Study of Mixture Homogeneity in a High-viscous Flow Mixer: Mingjin Yang, Associate Prof. College of Engineering and Technology, Southwest University, Chongqing 400716, CHINA; **Tielin Shi**, Professor, School of Mechanical Science & Engg., Huazhong University of Science & Technology, Wuhan 430074, CHINA; **Xiwen Li**, Prof., same; **Shuzi Yang**, same.

Twin screw mixers are commonly used to mix a high-viscous flow fluid. An approach for the impact of mixing parameters of a twin screw mixer on mixture homogeneity of the mixing slurry is proposed in this study. Definition of density-based intensity of segregation is put forward to evaluate mixture homogeneity of the slurry. By means of Orthogonal Factorial Experiment Design technique and statistical analyses of range and variance, major parameters of the mixing process and their significance affecting mixture homogeneity of the slurry are obtained. Experimental and analysis results show that control factor of ingredient of the slurry significantly contribute to mixture homogeneity, and ingredient of coarse solid phase particle has bad performance of mixture homogeneity. Besides, a 3-D sampling model is presented for good representation of the samples being taken in this study.

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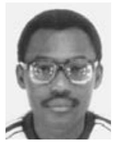
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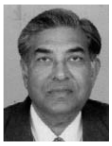
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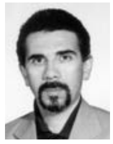
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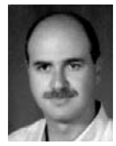
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Back Issues

AGRICULTURAL MECHANIZATION IN ASIA, AFRICA AND LATIN AMERICA

(Vol.40, No.3, Summer, 2009-)

Vol.40, No.3, Summer, 2009

◇ ◇ ◇

Vol.40, No.4, Autumn, 2009

◇ ◇ ◇

Vol.41, No.1, Winter, 2010

◇ ◇ ◇

Vol.41, No.2, Spring, 2010

◇ ◇ ◇

Vol.41, No.3, Summer, 2010

◇ ◇ ◇

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