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VOL.27, NO.4, AUTUMN 1996

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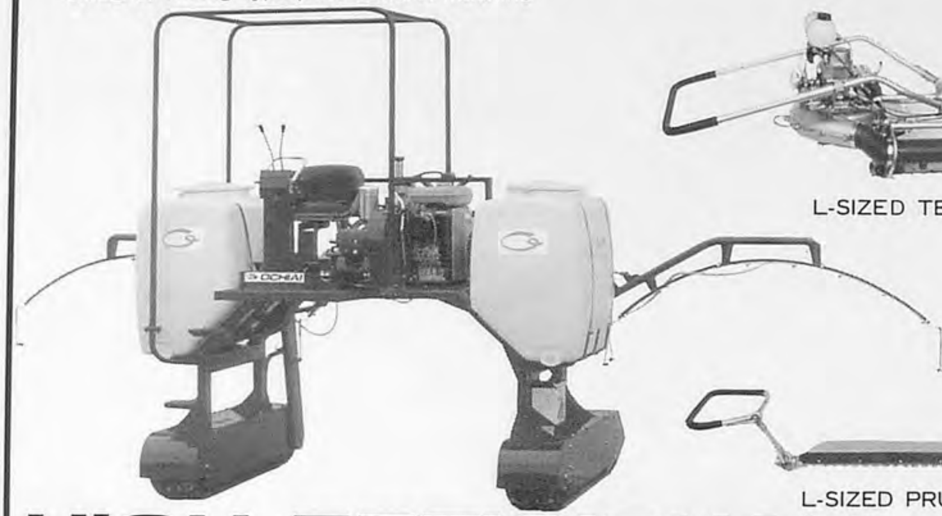

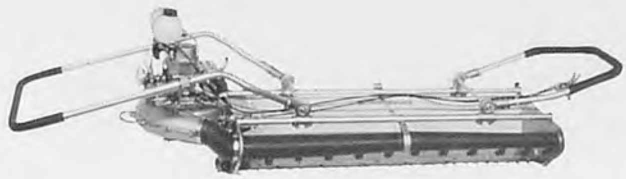

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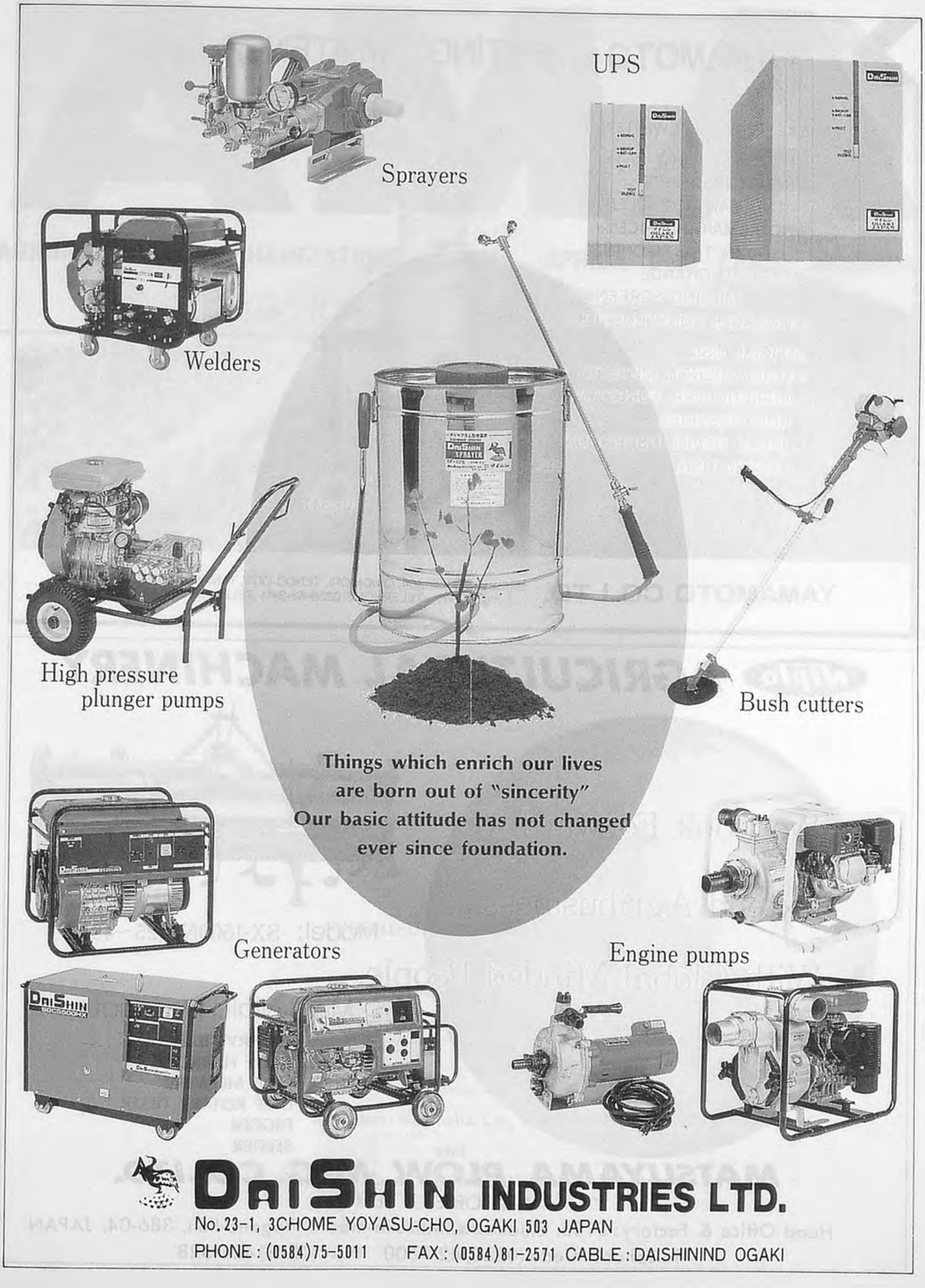
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EDITORIAL

Since early in the 1990's, the world demand for cereals has come to exceed its supply due to population increase. This is prominent especially in China which is the most populous country in Asia. China has a population of over 1.2 billion and it is ever increasing in spite of the government's policy to control population. Moreover, people has come to eat more meat with higher income brought by economical growth, which largely increased the consumption cereals feed as a matter of course. China eventually banned the export of cereals in 1994, soybeans in 1996.

The world market price of cereals has become unstable under these circumstances. The world natural resources usable per capita, like forests, farmland, is reducing year by year. We will not be able to support ourselves without raising land productivity. Agricultural machines are not only indispensable for us to do the most suitable work most timely with lightening the labor, but are playing a key role in raising land productivity, as once pointed out by Dr. G.W. Giles. The key item in increasing food production is to promote agricultural mechanization in developing countries.

We have been enough aware of usefulness of agricultural machines in view of labor saving. Now we should have a better appreciation for what agricultural machines can do to raise land productivity. In developed countries farm output had grown rapidly by raising the productivity. Its growth, however, has become slow-paced recently. There is a need for developing new and outstanding technology to further improve land productivity. In some parts of developing countries, rich land is changing into desert and eco-system is being badly pressured due to population increase, forests destruction and etc. Faced with many difficulties, we are to join our efforts and challenge the way to find a better solution.

Yoshisuke Kishida
Chief Editor

Tokyo, Japan
October, 1996

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AGRICULTURAL MECHANIZATION IN ASIA, AFRICA AND LATIN AMERICA

Vol. 27, No. 4, Autumn 1996

Different Responses of Tillage Practices in Reclamation of Dense Saline Sodic Soils

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Abstract

The productive land of Pakistan is being destroyed due to waterlogging and soil salinity/sodicity. Considerable work has been done for the reclamation of saline, saline sodic and sodic soils in the past. However, an appropriate solution for reclamation of dense saline sodic soils is still awaited. Previous research work was focussed on the application of chemical and biological amendments with narrow tine (N.T.) cultivator, thus giving slow response due to poor permeability of such soils to canal water. This study was, therefore, designed to employ deep tillage implements along with chemical and biological amendments for reclamation of dense saline sodic soils.

The physical properties of soil like shear strength, penetration resistance, soil bulk density and infiltration rate were measured as an index to soil amelioration process. The seedling emergence of *Jantar* (*Sesbania aculeata*) and wheat crops were recorded in order to assess the differential response of treatments applied. The study indicated that subsoiler was the most effective tillage implement followed by chisel plow in improving soil physical properties, the N.T. cultivator (control) gave the poorest response. The *Jantar* and wheat emergence were 2 to 3 times more in subsoiled and chiseled plots compared with control treatment.

Acknowledgement: Financial support from International Waterlogging and Salinity Research Institute (IWASRI), Lahore is greatfully acknowledged. Investigations to the current stage could not have taken place without such assistance.

Introduction

The twin menace of waterlogging and salinity is spoiling agricultural land at an alarming rate of 4.86 ha/h in Pakistan (IWASRI, 1989). About 5.73 million ha of salt affected soils occur in the country. Nearly 80% of the salt affected soils of Punjab and 56% of Pakistan are saline sodic (Muhammad, 1983), and are not easily reclaimable because of low permeability to water.

Reclamation of saline soils is easily accomplished by simple leaching with water. For reclamation of saline sodic soils, the use of certain chemical amendments is essential. However, reclamation of the dense saline sodic soils is extremely difficult. Such soils behave like stone with almost negligible permeability. In Pakistan as well as in foreign countries research has been conducted on reclamation of such impermeable soils. Most of this work pertains to the use of chemical and biological amendments whereas, the differential effects of tillage with other amendments have not been throughly investigated. The use of tillage implements has been considered fundamental in reclamation of saline sodic soils. An unthoughtful use of deep tillage implements may destroy the natural soil structure that has been slowly developed through centuries. Thus, selecting the type of implement, depth of its operation, desired level of soil tilth etc. need to be established for a successful reclamation of the problematic soils. The present study was accordingly planned to investigate the different effects of various tillage practices and their interactions with chemical amendment (gypsum) for reclamation of dense saline sodic soils.

Background

Hard Layer

According to conservative estimates, in Punjab alone, 1.94 million ha of land are dense saline sodic (Tariq, 1990). There are three distinct types of hard layers, i.e., plow pan, clay pan and sodic pan (Chaudhry, 1990). A plow pan develops as a result of continued plowing at a shallow depth through the years. This pan restricts water movement and, consequently, the salts are carried with the irrigation water. For reclamation of such soils, moldboard plowing is considered dangerous, since inverting such soils means bringing salted layer at the surface and enhancing the salinity level of the cultivated layer. However, chiseling coupled with leaching is very effective for these soils. A clay pan develops in the soil with a shallow layer (20-26 cm) of sandy or sandy loam soils at the surface followed by a clay layer. The movement of salts and smaller soil fractions are obstructed by the clay layer. Thus, a hard pan starts developing at the junction of loam and clay layers and gets harder with the passage of time. Moldboard plowing followed by disking helps mixing the loam and clay layers and reclaim the soil.

Dangerous of all is the sodic pan that develops in the sodic soils or through use of brackish well water for irrigation. In fact, sodic pan develops due to deflocculation of soil initiated by the sodium on exchangeable complex (Baver, 1928 and Harris, 1931). The deflocculation produces abundance of disintegrated silt and clay sediments which seal the soil pores on their downward movement and the soil voids almost cease to exist. Such a soil behaves like concrete with practically negligible permeability and soil is considered almost dead as regards a medium to support plant life. Such soils present a challenge to the agricultural scientists for their reclamation. Reclamation of such soils is the subject of research reported here.

Past Research

Previous researchers have intensively investigated the effects of chemical and biological amendments on reclamation of saline sodic soils (Muhammad et al., 1969a; Hussain and Hamid, 1978; Nishat, 1978 and Rashid et al., 1985) whereas the soil physical properties such as soil density, porosity, penetration resistance, soil roughness and infiltration characteristics have not been throughly investigated. Tillage practice is thought to play a vital role in changing the soil properties, therefore, proper selection of the implement for reclamation of soil is important. Tillage practices include subsoiling, chiseling, moldboard plowing or a combination of these followed by secondary tillage implements like cultivator, disk harrow etc. depending on the initial soil conditions. Some of the previous studies (Muhammad et al., 1988) have used subsoiling and there is no mention of other tillage implements either primary or secondary ones. In fact all the tillage implements have varied effects on soil physical properties and, therefore, their interaction with other chemical and biological treatments may produce useful results. This research had been planned to take into account the effectiveness and efficiency of various tillage implements combined with chemical and biological amendments to reclaim saline sodic soils and evolve a suitable package for the farming community.

Materials and Methods

Site Selection

During the first phase of the study a hunt for suitable site, with dense saline sodic soil, was initiated in the province of Punjab. Depending upon the level of salinity, the approach to the experimental area and cooperation of farmers, the site was selected in Faisalabad district.

Soil Hardness Profile

In order to assess the nature and depth of hard layer at the selected site, a pit (150 × 150 × 150 cm) was dug in the middle of the field. Soil resistance to penetration was measured using a cone penetrometer with a cone of 2 cm diameter. The measurements were made up to 150 cm depth with the increments of 15 cm.

The treatments and their levels included in the experimentation were as **Table 1**.

Application of Treatments

The field consisted of 1.20 ha of flat land that has never been

Table 1.

Treatments	*Levels*
1. Tillage practices	i. Subsoiling (40-50 cm deep)
	ii. Chisel plowing (30-35 cm deep)
	iii. Disk plowing (20-25 cm deep)
	iv. Tine cultivation (control) (8-12 cm deep)
2. Chemical amendment	i. 75% of gypsum requirement (GR)
	ii. 50% of gypsum requirement (GR)

under cultivation. The field was divided into two blocks of 0.60 ha each. The blocks were designated as A and B. Each block was further subdivided into 12 equal-sized plots each measuring approximately 8.63 × 57.94 m. These plots were randomly assigned four tillage treatments (subsoiling, chiseling, disk plowing and tine cultivation). Each treatment was replicated thrice. Tillage operations were performed in the designated plots. Gypsum at the rate of 50% and 75% of gypsum requirement (GR) was applied in blocks A and B, respectively. After tillage operations all the plots were flooded with water. When the field reached to field capacity, the seedbed was prepared with two passes of tine cultivator. Wheat variety LU 265 was seeded. In order to ascertain the effects of tillage treatments, chemical amendments and leaching, soil and crop parameters were studied.

Results and Discussion

The hardness profile of the site is presented in **Fig. 1**. The soil indicated maximum resistance of 4 000 g/cm^2 at a soil depth of 15 cm and the same decreased with the depth and reduced to 1 200 g/cm^2 at a depth of 150 cm. It appears from the graph that the harder patch of the soil existed between surface and 60 cm soil depth whereas the lower soil layers were relatively softer.

As infiltration rate of water through the soil was very low, the plot did not reach field capacity level and wheat crop could not be planted. *Jantar* was sown in the following April. However, this crop did not grow due to high salinity level. These two crops were again sown in the following year.

Emergence Counts of Wheat

Emergence counts were taken 20, 27 and 33 days after planting using one square meter steel frame, from randomly selected spots, in all the plots. The data were averaged over the two blocks as the differences in plant population of the two blocks varying in gypsum levels were insignificant. The wheat emergence was maximum in the subsoiled plots followed by almost similar emergence counts in chiseled and disked plots **(Fig. 2)**. However, the emergence remained poor in the tine cultivated plots. All this suggests that subsoiler was more effective in breaking the hard layer, enhancing leaching and rendering a desirable seedbed conducive to plant germination in a saline sodic soil. The differencial response of tillage treatments ranked subsoiling at the first position, chiseling/disk plowing at the middle and tine cultivation as poor in performance.

Emergence Counts of Jantar

In order to assess the effects of previously applied gypsum and tillage treatments on leaching of salts and, consequently, soil amelioration status, *Jantar* was planted. Emergence of the crop was recorded 27 days after planting and values are given in **Table 2**. Emergence counts indicate clear trends as regard the effect of tillage treatments. Emergence was 2 to 3 times greater in 75% GR than the plots having 50% GR. This suggests that additional gypsum was quite beneficial in soil amelioration process.

Subsoiled plots gave maximum emergence whereas, chiseled and disk plowed plots produced similar effects and the tine cultivated plots (control) was at the bottom of the list. The effects of tillage treatments were more pronounced in the block with higher gypsum content. In fact, emergence counts present a composite effect of soil amelioration resulting from various treatments and therefore, emergence provides a clear picture of the cause-effect relationship.

Table 2. Effects of Tillage Practices and Gypsum Levels on Emergence of Jantar (*Sesbania aculeata*)

Tillage implement	Emergence counts per square meter area	
	gypsum 75% of GR	gypsum 50% of GR
Subsoiler	83	27
Chisel plow	46	25
Disk plow	44	22
Cultivator	34	21

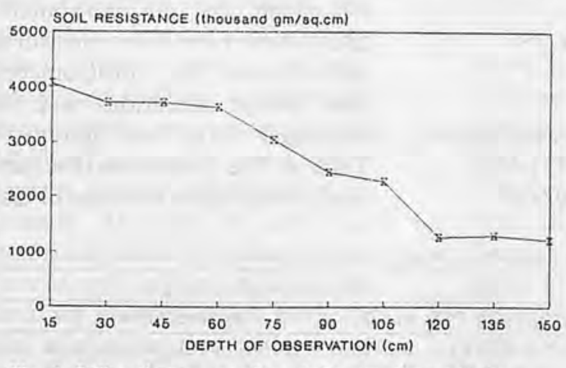

Fig. 1 Soil resistance profile of the experimental site.

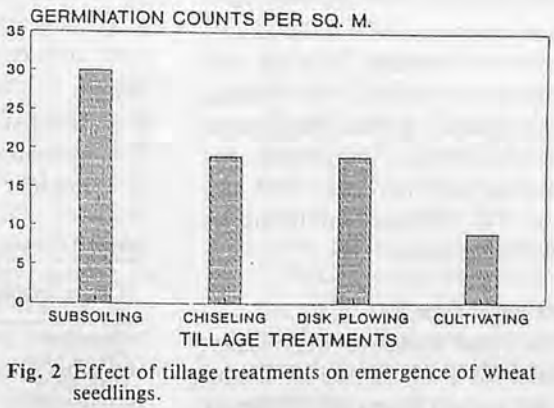

Fig. 2 Effect of tillage treatments on emergence of wheat seedlings.

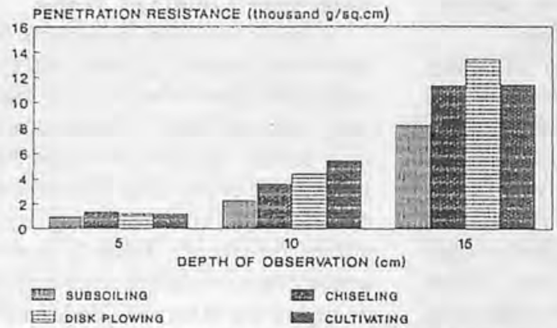

Fig. 3 Effect of tillage treatments on soil penetration resistance.

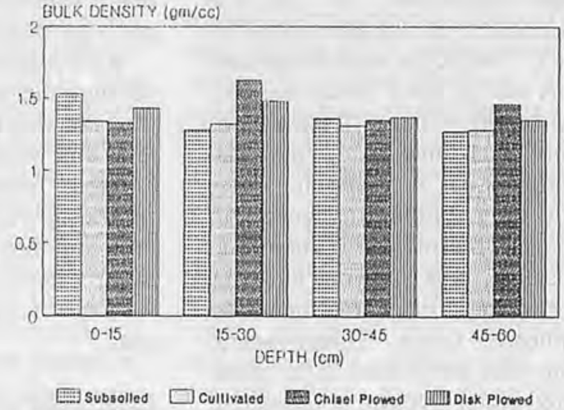

Fig. 4 Effect of tillage treatments on bulk density of soil.

Soil Resistance to Penetration

In order to assess soil tilth in differently tilled plots soil, penetration resistances were measured using 2 cm dia. cone penetrometer at 5, 10 and 15 cm soil depths. The soil resistance increased with the depth since the top layer was pulverized. The tillage effects were non-significant near the soil surface (5 cm) but more pronounced at 10 cm and 15 cm depths. The subsoiled plots indicated least resistance at 10 and 15 cm depths compared with tine cultivated and disked plots (**Fig. 3**). Soil resistances to penetration at 10 cm and 15 cm soil depths measured in subsoiled plots were 2 290 and 8 260 g/cm^2, respectively, while in cultivated plots resistance values at the said depths were 5 430 and 11 460 g/cm^2, respectively. The trend in the resistance profile indicated that subsoiler was quite effective in reducing soil resistance (hardness) compared with rest of the treatments. Disked plots showed maximum resistance of 13 440 g/cm^2 at 15 cm, perhaps, due to soil compaction caused by bearing areas of disks. Subsoiler appeared relatively better implement for reducing soil hardness and the same was reflected in terms of better emergence.

Soil Bulk Density

Dry soil densities were determined for 0-15, 15-30, 30-45 and 45-60 cm soil layers in differently tilled plots. No significant effect of gypsum was noted hence the density values were averaged over the blocks and the results are shown in **Fig. 4**. The densities were similar except for chiseled plots in which density was the highest for the soil layer at 15-30 and 45-60 cm and these were 1.8 and 1.4 g/cm^3, respectively.

Soil Shear Strength

Soil shear strength is a good indicator of seedbed preparation as regards the emergence of seedlings. Cohron Sheargraph apparatus was used to measure the shear stress of soil against three levels of normal stress. The measurements were made at randomly selected locations in differently tilled plots. Employment of linear regression analysis yielded the following shear strength equations (**Table 3**) using Coulomb's theory given below:

$$S = C + N \tan\varnothing$$

Where,

S = shear strength of soil, g/cm^2
C = cohesion of soil, g/cm^2
N = normal stress, g/cm^2
$\varnothing$ = angle of internal friction of soil, degrees

Among the tillage treatments, only the disk plowing effectively reduced soil cohesion. Soil cohesion was 15.64 and 14.59 g/cm^2 in disk plowed plots for 75% GR and 50% GR, respectively, while the cohesion values in cultivated plots for 75% GR and 50% GR were 112.6 and 114.0 g/cm^2, respectively. Maximum value of soil cohesion and internal friction in tine cultivated plots (control) adversely affected the emergence process in these plots. This partly explains the poor emergence in tine cultivated plots

Infiltration

Infiltration is a complex process that largely depends on the soil physical properties such as initial soil moisture content, previous wetting history, permeability changes due to surface water movement and air entrapment. Infiltration rates were measured with double ring infiltrometers after tillage operations and the averaged data are given in **Table 4**. The data shows that maximum infiltration rate was in disk

Table 3. Shear Strength Equations Using Coloumb's Theory

Tillage implement	Shear strength	
	gypsum 75% of GR	gypsum 50% of GR
Subsoiler	S = 77.10 + 0.29 N	S = 76.24 + 0.30 N
Chisel plow	S = 76.02 + 0.30 N	S = 85.52 + 0.31 N
Disk plow	S = 15.64 + 0.54 N	S = 14.59 + 0.54 N
Cultivator	S = 112.6 + 0.28 N	S = 114.0 + 0.27 N

Table 4. Infiltration Rates as Affected by Different Tillage Implements and Levels of Gypsum

Tillage implement	Average infiltration rate (mm/h)	
	75% GR	50% GR
Subsoiler	20.39	13.80
Chisel plow	17.36	11.73
Disk plow	20.57	18.50
Cultivator	18.57	10.75

plowed plots which was 20.57 and 18.50 mm/h in 75% GR and 50% Gr blocks, respectively. The data suggest that additional gypsum (75% of GR) helped in improving infiltration process. Perhaps, more sodium was removed with additional gypsum and the sealing effect due to exchangeable sodium activity reduced. As regards tillage implements, the rate of infiltration was generally better in deeply tilled plots.

Soil Chemical Properties

Soils were sampled up to 150 cm soil depth with 15 cm increments before and after the application of treatments for analysis of electrical conductivity (EC), pH, exchangeable sodium percentage (ESP) and sodium adsorption ratio (SAR). Analysis of the data revealed that the main effects of tillage practices as well as gypsum levels were non-significant. Non-significant effects of different tillage operations and gypsum quantities may be attributed to the spacial variability of soil itself within the experimental area. As the saline and non-saline spots were randomly distributed over the entire area, the actual treatment effects were, perhaps, masked by the soil variability. In such cases the measurement of soil physical properties may better explain the soil amelioration efficiency. In view of the non-significant effects of tillage practices as well as gypsum levels, the data for each chemical property was averaged across tillage and gypsum levels (blocks). It is obvious from **Table 5** that the soil

Table 5. Chemical Analysis of Soil Before and After Application of Treatments

Depth (cm)	EC (mmohs/cm)		pH		ESP		SAR	
	B	A	B	A	B	A	B	A
15	29.6	4.6	8.1	7.4	116.1	18.0	92.1	13.9
30	28.6	5.0	8.1	7.6	115.0	20.8	91.3	16.1
45	22.4	6.6	7.9	7.7	87.7	28.7	69.5	22.5
60	15.5	7.8	7.9	7.8	38.2	31.1	30.0	24.3
75	10.8	9.4	7.8	7.9	35.9	33.3	28.1	26.1
90	8.2	12.9	7.8	8.0	27.2	42.7	21.2	33.3
105	6.2	16.3	7.8	8.0	21.6	51.4	16.7	40.5
120	5.6	19.9	7.8	8.0	21.2	68.3	16.4	54.0
135	5.1	22.7	7.6	8.1	18.9	77.0	14.6	60.9
150	5.0	22.9	7.5	8.2	17.8	74.9	13.7	74.9

reclamation took place at quite an accelerated rate. For instance, EC decreased from 290.60 to 4.57 (84.6%), pH 8.1 to 7.43 (8.3%), ESP 115 to 18 (84.3%) and SAR 92.21 to 13.92 (84.9%) for 0-15 cm soil layer.

Conclusions

The following conclusions were drawn from the first year's experimentation:

1. The hard pans observed in sodic soils existed up to a depth of 150 cm or even deeper. The hard layer at the site was a sodic pan that results from the downward movement of deflocculated soil sediments.
2. Hardness profiles were drawn for the selected soil. the resistance to penetration, especially in the plow layer, was 4 000-6 000 g/cm^2. Such value is too high and results in a very low soil permeability.
3. Wheat emergence was maximum in the subsoiled plots followed by almost similar emergence in the chiseled and disk plowed plots. However, emergence remained poorer in tine cultivated plots (control). These data suggest a superiority of subsoiler among its counterparts. Perhaps subsoiler broke hard layer effectively, caused more salt leaching and thereby left seedbed conducive for seed germination.
4. Soil resistance to penetration after application of treatments indicated that the subsoiled plots produced the least resistance values at 10 and 15 cm depths. This helps in explaining a better emergence in the subsoiled plots.
5. Analysis of soil shear strength data yielded maximum values of soil shearing resistance and soil cohesion in disk plowed and tine cultivated plots, respectively. Increased values of soil shearing resistance and cohesion may partly explain the poorer emergence in the disked and tine cultivated plots.
6. Water infiltration rate increased in the plots with 75% GR compared with those of 50% GR. Perhaps, additional gypsum, removed more exchangeable sodium on the soil complex and consequently, reduced sealing effects caused due to deflocculation of sediments.
7. Generally deep tillage accelerated infiltration rate except in chiseled plots. Subsoiled and disked plots produced maximum infiltration rates.
8. Jantar emergence was 2 to 3 times greater in the plots with 75% GR compared with those of 50% GR. This suggests additional gypsum was quite beneficial in soil amelioration process.

9. Subsoiled plots had maximum plant population of Jantar, whereas, chiseled and disk plowed plots were similar in seedling counts and the averaged of tine cultivated plots (control) was the least.
10. Differences in the soil chemical properties created due to treatments were, perhaps, masked by the soil spatial variability.

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The World Food Prize

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Development of Minimum Till Planting Machinery

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Abstract

Tillage is a time, energy consuming and costly intensive production operation. Agricultural scientists at the Punjab Agricultural University. Ludhiana (India) have demonstrated through experiments for several years that wheat crop could be successfully raised with minimum till planting system without any yield loss, provided planting could be accomplished on time and weeds could be controlled chemically. For successful adoption of the minimumm till planting technology under varying field soil conditions, a tractor-drawn 9-row strip-till drill was developed. It cuts and tills a 7.5-cm wide strip in front of each furrow opener of the drill. The intensive on-farm research had shown that wheat crop could be planted under untilled field conditions by this machine and there was no loss of yield. With its use in the fields with heavy soil there was a fuel saving to the extent of 40 ℓ/ha of diesel, time saving of 65-70% and reduction in the cost of production of US$ 16/ha. The machine has been recommended by Punjab Agricultural University, Ludhiana for farm level adoption and is commercially available for US$1 000 each.

Introduction

Tillage is one of the major farm operations and is an important contributor to the total cost of production. Excessive tillage is energy and time consuming and costly operation. It is considered harmful to the soil structure. It also contributes to wind and water erosion of the soil. The rising cost of hydrocarbon fuels which are bound to be exhausted sooner or later, availability of herbicides coupled with the motive of timely sowing and reducing the cost of production has provided enough incentive to the researchers to the world over to investigate tillage operations more closely.

Much of the work has been done in the U.S.A., Canada, Australia, New Zealand, U.K. and some other European countries (1, 2, 3, 4, 5, 6, 8, 9, 10, 11). Research has been conducted under the titles of minimum-tillage, no-tillage, optimum tillage, soil compaction, etc. Results have shown that with the development and use of appropriate herbicides, the technique of zero tillage/minimum tillage/direct drilling has shown considerable potential for some crops under certain conditoins. Research conducted in the Dept. of Agronomy (12) at Punjab Agricultural University, Ludhiana had shown that wheat crop could be grown under minimum tillage conditions without any loss in yield. However, the major handicap in the adoption of this technology was the non-availability of a suitable planting machine. This paper discusses the details regarding the development and field evaluation of a suitable 9-row tractor-drawn seed drill for planting in untilled field, conducted at the Department of Farm Power and Machinery during the last decade.

Development of Minimum Tillage Attachment

In Punjab, wheat is grown in an a rea of over 3 m-ha and is planted by tractor or bullock drawn seed-cum-fertilizer drills. The first approach to the development of a suitable minimum till drill was to develop an appropriate attachment to an existing drill for adoption of the technology (13). With this end in view, five different types of commercially available seed cum fertilizer drills were tried for direct drilling of wheat in manually harvested paddy fields. This was intended to asses the problems arising from the use of commercially available drills under no-tillage regime. The major problems encountered were accumulation of straw and stubble in front of the tynes, formation of clods, poor coverage of seed and fertilizer leading to bird-damage to the seeds, excessive slippage of ground wheel due to uneven fields leading to skips in the placement of seed and fertilizer as well as higher power requirement for the operation. To overcome the above mentioned problems, it was decided to develop an attachment in front of furrow openers of the existing seed drill.

Coulter Attachment

After careful review of various no-till drills available in the literature and considering the specific requirements of wheat farmers in the Punjab, a disc-coulter attachment was designed and fabricated in the Department of Farm Power and Machinery, PAU, Ludhiana. Details of the planting machine with the coulter attachment were reported by Shukla, et al. (7). Major limitations of such a machine were its poor performance in medium and heavy soils. To overcome these problems, a rotary blade attachment was developed and evaluated in the same Department. Details of the work are given in the pages that follow.

Strip-till Seed-cum-fertilizer Drill

The drill is essentially a 9-row seed-cum-fertilizer drill with a rotary blade attachment for minimum soil manipulation running ahead of the normal furrow openers (**Fig. 1**). It is operated by a tractor of 35 or higher horse power (26.11 kW). The rotary attachment consists of a frame with a rotor with '9' flanges. Each flange has 6 C-type tines (blades). The spacing between the flanges is the same as the row spacing for the crop to be planted. Power to the rotor shaft is provided from the tractor PTO through a speed reduction gear box and chain and sprocket drive. The rotor revolves at a speed of 300 rpm corresponding to the rated PTO speed of 540 rpm. The rotary attachment is provided with a M.S. sheet cover to protect the power transmission system. It also helps to reduce the soil cover over the seed. The specifications of the strip-till drill are given in **Table 1**.

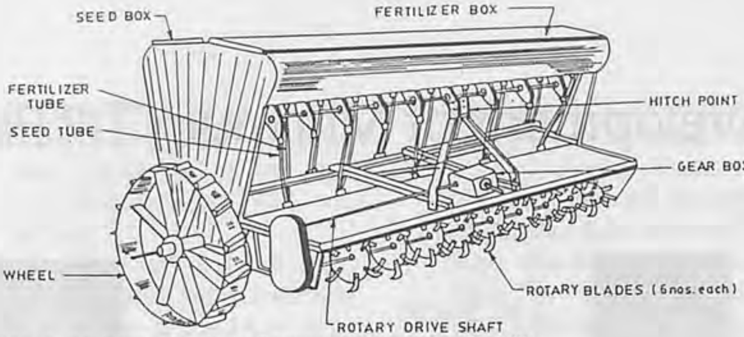

Fig. 1 An isometric view of tractor-drawn strip till drill.

Table 1. Main Specifications of the Strip-till Drill

Type	: Tractor-moounted
Power required (hp)	: 35 or above (26.11 kW)
No. of furrow openers	: 9
Type of furrow openers	: Reversible shovel type
Row spacing (mm)	: 204
Provision on drive wheel to reduce slippage	: Lugs
Seed metering device	: External fluted feed rollers
Fertilizer metering device	: Gravity type with adjustable orifices and agitators
Transmission system	: Chain and sprocket drive
Strip-till attachment	: Detachable type
Rotor shaft	: High pressure pipe
Internal dia (mm)	50
Thickness (mm)	10
Transmission shaft diameter (mm)	: 40
Flange No.	: 9
Flange dia. (mm)	: 230
Flange thickness (mm)	: 10
No. of blades on each flange	: 6
Blade type	: C type
Width of cut of each blade (mm)	: 37.5
Blade mounting	: Properly and uniformly staggered to give a helical cutting pattern for even load on the tractor
Type of gear in gear box	: Bevel
No. of teeth of drive	: 10
No. of teeth of driven	: 18

Experimental Procedure for Field Evaluation

The drill was used for planting wheat crop in heavy textured soils under paddy-wheat rotation. Wheat variety of WL-711 was sown. The experiments were conducted for four years continuously, i.e., 1985 through 1988-89 in the same field using randomized block design with three treatments (including control) and four replications. In each plot 18 rows of wheat crop were planted. The control treatment consisted of standard field preparation recommendation of Punjab Agricultural University for raising wheat crop in heavy textured soils. These included two operations by a disk harrow, two operations by a field cultivator and three operations by a planker. Post emergence weed control was done mechanically with a manually-operated wheel-type weeder.

In treatment T1 direct sowing

of wheat with minimum till seed drill without any preparatory tillage was performed. Post emergence weed control was done chemically. However, in treatment T2 post emergence weed control was done mechanically.

Experimental data regarding soil moisture content at the time of sowing, fuel consumption, labour requirement, germination count and yield were taken. The field data was analyzed and found quite encouraging.

Table 2. Average Field Data of No-till Experiments for Wheat Crop in Heavy Soil

Crop year	Treatment	Soil Moisture (percent)	Soil Bulk density before sowing (gm/cc)	Germination count (No./m^2)	Yield (t/ha)
1985-86	C	18.78	1.36	92.7	3.83
	T1	18.93	1.47	89.0	3.76
	T2	19.36	1.52	80.8	3.72
1986-87	C	22.20	1.26	74.7	4.10
	T1	23.63	1.31	49.7	3.99
	T2	22.26	1.36	60.7	3.82
1987-88	C	16.45	1.12	103.0	3.77
	T1	17.19	1.29	110.0	3.79
	T2	17.72	1.36	88.2	3.81
1988-89	C	19.72	1.21	104.9	3.94
	T1	21.09	1.32	117.3	4.05
	T2	21.31	1.32	111.6	3.62

Note: C - Conventional tillage and the weed control practices as recommended by PAU, Ludhiana.
T1 - Strip-till drill sowing and use of post emergence chemicals for weed control.
T2 - Strip-till drill sowing and post emergence weed control mechenically.

Table 3. Salient Data on Strip-till Drill Experiments (1991-92) for Sowing Wheat

Location	Date of sowing	Germination count (Plant/m^2)		Yield t/ha	
		Conventional till	Strip till	Conventional till	Strip till
Drainage Farm Deptt. of Soil & Water Engg. PAU, Ludhiana	23.11.91	217	227	3.00	3.24
Rice, Research Station, Rauni, Patiala	30.11.91	96	142	3.54	3.40
Rice Research Station Kapurthala	4.12.91	—	—	3.06	3.85
Deptt. of Agronomy Farm, PAU, Ludhiana					
Paddy-wheat	6.12.91	46/m	41/m	5.04	5.04
Cotton-wheat	12.12.91	60/m	61/m	4.71	4.61
Seed Farm Narain Garh					
Cotton-wheat	8. 1.92	—	—	1.96	1.97

Results and Discussion

Results of experiments are given in **Table 2**. The field had medium heavy soil with the soil moisture varying between 16.45% and 23.63% during different years. Average soil bulk density before sowing varied between 1.12 and 1.52 g/cc. in plots of different treatments during the four years of study. The average soil bulk density values in the control plots each year were less than T1 and T2 treatment plots. There was non-significant difference in average soil bulk density of 'T1' and 'T2' plots. The average germination count in plots of different treatments during the period varied from 49.7 to 117.3 plants/m^2. No definite correlation in germination count between different treatments was found. The average yield for various treatments varied from 3.62 to 43.10 t/ha. There was no significant difference in the yield at 5% level of significance among the different treatments.

Cone penetrometer readings were recorded for all the plots. The average cone index values in 'T1' and 'T2' plots were higher than those for the control or normal tillage plots. This is in conformity with the data on average bulk density recorded for different treatment plots. The effect of tillage as measured through cone index values was studied up to a depth of 20 cm.

Based on the promising results of the experiment, it was decided to undertake multi-location field trials at different research farms of PAU, Ludhiana. The experiments were conducted during the 1991-92 wheat season. These experiments were conducted at the Drainage Farm, Dept. of Soil and Water Engg., Rice Research Station, Rauni (Patiala), Rice Research Station, Kapurthala; Seed farm, Naraingarh and the Agronomy Farm of PAU, Ludhiana. Observations on date of sowing, germination count and yield data were recorded.

The results of the multi-location research trials are given in **Table 3**. The dates of planting at different locations were between Nov. 23, 1991 and Jan. 8, 1992. The late planting at Naraingarh Farm was due to delayed harvesting of cotton in the cotton-wheat rotation. The yield varied between 3 and 5.04 t/ha except in the case of Naraingarh Farm where it was 1.96 t/ha due to fairly late planting of wheat after the last picking of cotton and uprooting of the cotton sticks. It was observed that, in general, there was no perceptible difference in the yields between the two treatments except in the case of Rice Research Station, Kapurthala where yield in unprepared field using strip till drill was considerably higher than the conventional till field.

Energy input and cost of production of wheat under paddy-wheat rotation in heavy soil for conventional tillage and minimum tillage was also studied from 1985-86 to 1988-89. Under the minimum tillage system, the diesel fuel consumption in planting operation only (no separate seedbed preparation required) was 18 ℓ/ha while diesel fuel used for seedbed preparation sowing under conventional tilllage system was 60 ℓ/ha. Thus, there was a saving of diesel to the extent of 42 ℓ/ha. The total energy input for minimum tillage system with mechanical weed control was 17 264 MJ/ha and that with chemical weed control was 17 328 MJ/ha. In conventional tillage, under similar soil conditions, the energy input varied between 19 659 MJ/ha and 19 723 MJ/ha.

Hence, under the minimum tilllage system, energy saving under heavy soil condition was 2 395 MJ/ha. The total cost of wheat production in conventional tillage system under heavy soil conditions varied between U.S.$118.61 and U.S.$125.19/ha. In the minimum tillage system, the total cost of production varied between U.S.$102.29 and U.S.$108.87/ha. Thus, there was a saving of U.S.$16.32/ha in the cost of production.

In the experimental results, no significant difference in the yield was observed when the crop was planted on the same day for both the conventional and minimum tillage systems. However, with the latter system, a time saving of 65 to 70% in comparison with the former tillage planting was obtained. Thus, by adopting the minimum tillage planting technology for wheat, the timeliness of operation improved significantly resulting in an increase in the total yield of wheat.

In the Punjab State, wheat is grown in an area of over 3 m-ha. For every million ha sown under the minimum tillage system, a saving of U.S.$16.32 million in the cost of production can be achieved. The diesel saving for every million ha brought under this system will be about 42 million litres. Thus, there was a good potential of saving the precious petroleum fuel.

Conclusion

The tractor-drawn Strip-till Drill designed, developed and tested at PAU has already been included in the recommendations entitled Package of Practices 1993, of the Punjab Agricultural University, Ludhiana. A mechanically sound and reliable machine is now commercially available for about U.S.$1 000 each. It can be used for planting the wheat crop under different crop rotations.

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Modification and Field Performance Evaluation of a Windrower for Rapeseed in Pakistan

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Abstract

After making the necessary modifications into horizontal conveyor and overhead reel assemblies of JF Reaper Windrower, it was tested in the field to evaluate its performance for harvesting rapeseed at three forward speeds and two crop maturity levels. The machine showed the best performance at the combination of 3.03 km/h forward speed and 30% seed moisture level among all the combinations tested. The mechanical harvesting of rapeseed is economical and having less seed loss as compared to the manual system. Break-even area and pay-back period of the windrower are 32 ha and 2 years, respectively, at 300 h of annual use.

Introduction

Pakistan is facing a severe shortage of edible oil. Only 30% of edible oil requirement of the country is fulfilled by domestic production (Ahmad and Hanif, 1986) and the rest is imported, resulting in heavy expense of foreign exchange. Rapeseed and mustard are the main oil crops of the country covering approximately 27% of total domestic production of edible oil. Continuous efforts are being made to increase the production of these crops to achieve self-sufficiency in domestic edible oil production. However, little has been achieved as these crops compete with wheat and grain legume crops.

Harvesting is one of the most labor intensive operations in rapeseed production. The time for rapeseed harvesting in major rapeseed growing areas of Pakistan is mid-April. This is immediately followed by wheat harvesting. There is a severe shortage of labor during this period. The labor required for manual harvesting is 150 man-hours/ha as compared to only 60 man-hours for mechancial harvesting of rapeseed crop (Anwar et al. 1991). Therefore, it is essential to mechanzie rapeseed harvesting operation to shorten its harvesting time so that wheat harvesting can be started on time. Also, timely harvesting is important to obtain maximum yield from the crop (**Fig. 1**). Early harvesting results in poor seed development while delayed harvesting causes loss in yield due to shattering (Muhammad, 1986).

With this in view, the Barani Agricultural Research and Development (BARD) Programme of Pakistan Agricultural Research Council (PARC) started to develop a package of technology for rapeseed crop maximization in Pakistan, especially on rainfed lands. After a continuous effort for many years a prototype tractor front-mounted reaper windrower for rapeseed, known as JF Reaper Windrower, was developed by modifying the PECO reaper. Anwar et al. (1991) modified the prototype and tested it in the field. They reported a low field efficiency of 46.5% and high seed loss of 12.5% and suggested further modifications in the conveying system and reel assembly of the machine to improve its performance. This paper reports on the work done to further modify the

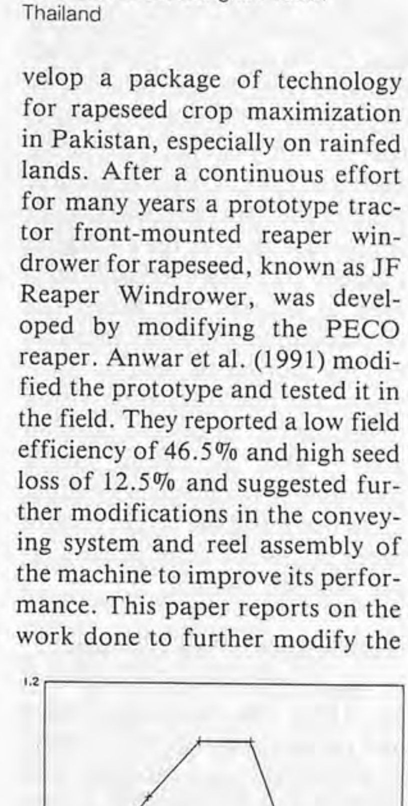
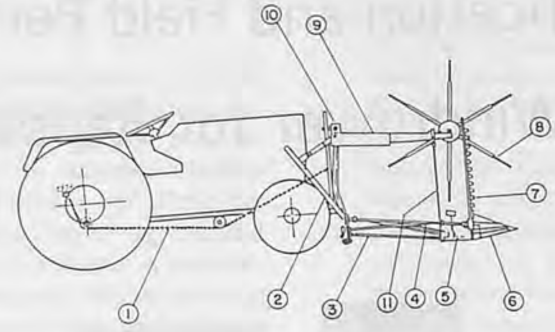


Fig. 1 Effect of harvest date on the yield of the rapeseed crop.

JF reaper windrower to make it suitable for local conditions, evaluate its field performance, compare its performance and economics of use with the conventional method, study the feasibility of adoption and recommend further necessary modifications, if any, to further improve its performance.



1 = Wire rope
2 = Reel height adjustor
3 = Horizontal cutter bar drive
4 = Vertical cutter bar drive
5 = Horizontal cutter bar
6 = Right side crop divider
7 = Vertical cutter bar
8 = Reel
9 = Reel arm
10 = Gear box
11 = Reel arm support

Fig. 2 Side view of JF reaper windrower.

Materials and Methods

Description of the Machine

The JF windrower is a tractor front-mounted, PTO-operated and hydraulically-controlled machine (**Fig. 2**). Its cutter bar is operated by PTO shaft through a pair of belt pulleys, a propeller shaft and cam. A vertical knife in front of the right hand side divider is provided to clear the entangled rapeseed crop. An overhead reel is used to support and gather the crop being cut and to lay the same on the horizontal canvas conveyor. The machine has provision to set the reel at desired position and to alter the cutting height according to requirement by either underneath stubble height control shoes or tractor three point linkage position control.

Modification of the Machine

As recommended by Anwar et al. (1991) the conveying system and reel assembly of the JF reaper windrower were modified and reinstalled on the machine. A conveyor width of 1 370 mm was selected for the newly designed conveyor for better conveying of the harvested crop. Accordingly, the conveyor roller, conveyor roller drive and vertical cutter bar drive were modified. Reel assembly was modified by providing six batts as compared to previously four batts for better guidance of the crop during harvesting.

Evaluation Criteria for JF Reaper Windrower

Two systems of rapeseed harvesting, (i) manual system i.e., manual cutting and collection of harvested crop and (ii) mechanical system i.e., mechanical reaping by JF reaper windrower and manual collection of harvested crop were considered. Parameters such as field efficiency, field capacity, losses of seeds and cost of operation were determined for both systems of harvesting. The performance of mechanical system in terms of these parameters was evaluated and compared with that of the manual system to study its feasibility of adoption. The break-even area and payback period of the JF reaper windrower were also determined.

Field Test and Data Collection

Pre-harvest parameters such a size of the field, crop condition i.e., variety, plant height, plant density, mode of planting, crop yield and pre-harvest labor requirements were noted. Machine operation parameters such as forward speed, effective width of cut, fuel consumption, cutting height or stubble height, different time elements to calculate field efficiency and effective field capacity as well as seed losses during machine operation and manual harvesting such as shattering, drop, conveying and uncut losses were also measured.

Cost of Operation

The total cost of harvesting of the machine was calculated by the following relationship (Hunt, 1973):

$$AC = FC + A/C\,(R\&M + L + O + F + T) \qquad (1)$$

where,

AC = Annual cost of operating a machine, US$/year
FC = Annual fixed cost, US$/year
A = Annual area harvested, ha
C = Field capacity, ha/h
R&M = Repair and maintenance cost, US$/h
L = Labor rate, US$/man-h
O = Oil cost, US$/h
F = Fuel cost, US$/h
T = Cost of tractor used by the machine, US$/h
= FC/h + R&M/h (for tractor)

Results and Discussions

General Observations on the Machine

Being tractor front-mounted and hydraulically-controlled machine, the operator gets better visibility of the job and field condition. However, due to wider horizontal conveyor and heavy weight of the reaper, transportation of the equipment posed some difficulty. Maneuverability of the machine in the field was easy as harvesting of rapeseed is done in dry soil condition. The machine could be lifted from 0.3 to 0.5 m using the tractor hydraulic power through steel ropes and suspension system. So there was no difficulty encountered while taking the turns or crossing the small bunds in the field. Though it satisfactorily handled the tall crops up to 1.75 m, some difficulty was encountered to handle the heavy green crop during field testing.

It was observed that no manual harvesting was required for head land opening before machine harvesting. It was also observed that the machine could easily cut the crop close to the bunds. Lowering and raising of the cutting table can easily be achieved from the operator's seat through the hydraulic control lever. The cutting height can also be adjusted easily through hydraulic control lever position up to a maximum of 350 mm. The presence of green weeds in the field did not cause any serious problem because the crop was cut well above the ground level.

General Field Information Data Recorded During Testing

The JF windrower was tested on rapeseed crop, Canola type Shiralee variety, at the experimental fields of the National Agricultural Research Centre, Islamabad. Test plot size of 0.20 ha was selected with three replications for each of three forward speeds of the tractor in 2nd, 3rd and 4th low gears. The average forward speed in these three gears were 2.28, 3.03 and 4.01 km/h, respectively. The average plant density, plant height and cutting height were 49 plants/m^2, 1.67 m and 0.33 m, respectively. The average cutting width of machine was 1.8 m.

Table 1. Field Efficiency, Field Capacity and Seed Losses at Different Speeds and Moisture Content Level

Forward speed (km/h)	30% Moisture content level					15% Moisture content level				
	Field efficiency (%)	Field capacity (ha/h)	Seed losses (% of yield)			Field efficiency (%)	Field capacity (ha/h)	Seed losses (% of yield)		
			ML	GB	Total			ML	GB	Total
2.28	80	0.33	7.2	3.5	10.7	85	0.33	16.6	4.9	21.5
3.03	83	0.44	8.5	3.5	12.0	80	0.45	17.2	4.9	22.1
4.01	68	0.63	8.9	3.5	12.4	82	0.67	18.5	4.9	23.4

ML = Seed losses during machine operation and GB = Gathering and bundling losses.

Field Efficiency and Field Capacity at Different Speeds and Moisture Levels

The average field efficiency of the machine was 80, 83 and 68% at 30% seed moisture content and 85, 80 and 82% at 15% seed moisture content when operating the machine at 2.28, 3.03 and 4.01 km/h, respectively **(Table 1)**. At 30% seed moisture content windrower has the lowest field efficiency of 68% at 4.01 km/h forward speed due to maximum time loss (182 seconds) in declogging the machine while the machine has the maximum efficiency of 83% at 3.03 km/h speed which was due to minimum declogging time of the machine at that speed. The average field efficiency of the machine at 15% seed moisture content was almost constant at all forward speeds because the crop was dry and time loss in declogging was almost the same in all cases.

The average effective field capacity of the machine was 0.33, 0.44 and 0.63 ha/h at 30% seed moisture content and 0.33, 0.45 and 0.67 ha/h at 15% seed moisture content when operating the machine at 2.28, 3.03 and 4.01 km/h, respectively **(Table 1)**. The average field capacity of the machine increased with the increase of both forward speed of travel and dryness of the crop. The increase in field capacity due to the increase in machine forward speed was highly significant at 0.01 level. The increase in field capacity with an increase in crop dryness was also significant at 0.01 level.

Seed Losses at Different Speeds and Moisture Content Levels

Seed loss due to machine operation (ML) increased as the machine forward speed increased at both seed moisture levels. The average machine loss increased from 7.2 to 8.9% of crop yield at 30% seed moisture level and from 16.6 to 18.5% of crop yield at 15% seed moisture level as the machine forward speed increased from 2.28 to 4.01 km/h, respectively **(Table 1)**. The total machine loss was significantly affected by seed moisture content at 0.01 level but the effect of machine forward speed was not statistically significant at any level. Gathering and bundling losses (GB) at both the moisture levels were 3.5% and 4.9% of yield, respectively. They were consistent at all speeds because these operations were carried out manually. The total average seed loss increased from 10.7 to 12.4% of crop yield at 30% seed moisture level and 21.5 to 23.4% of crop yield at 15% seed moisture level as the machine forward speed increased from 2.28 to 4.01 km/h, respectively **(Table 1)**.

Optimum Operating Condition of the Machine

The best possible combination of the JF reaper windrower was at 3.03 km/h forward speed and 30% moisture level of the crop as it corresponds to a reasonably high average field efficiency (83%) and comparatively lower seed losses (8.5%). Hence, this operating combination was selected as the optimum operating condition of the reaper. The average seed loss after incorporating the modifications into horizontal conveyor and reel assemblies were reduced from 12.5% to 8.5% and the average field efficiency increased from 46.5% to 83%.

Manual Harvesting System

The traditional sickle was used for manual harvesting of the crop. Tests were carried out at 30 and 15% seed moisture content level of the Canola type, *Shiralee* variety, of rapeseed in a test plot of 0.07 ha. The average crop density, plant height and crop cutting height were 57 plants/m^2, 1.68 m and 0.38 m, respectively. The average seed losses during reaping operation at 30 and 15% moisture levels were 9.5 and 11.8% of crop yield, respectively. The gathering and bundling losses at both the moisture content levels were 3.5 and 4.9% of crop yield, respectively. These losses are the same as those found for mechanically harvested crop. The average labor requirement at 30 and 15% moisture levels for reaping of crop were 96.0 and 93.2 man-hours/ha. The labor requirement for gathering and bundling operations at both moisture content levels was 50 man-h/ha.

Comparison of Manual and Mechanical Harvesting System

The performance of the JF windrower at 3.03 km/h operating speed was compared with that of the manual system in terms of field capacity, seed losses, labor requirements and cost of harvesting at 30% moisture content of the crop.

Field Capacity, Seed Losses and Labor Requirements

The average field capacity of the JF reaper windrower in reaping was 0.44 ha/h as compared to 0.01 ha/h of manual system.

The average seed losses during operation in mechanical harvesting system were 8.6% of crop yield as compared to 9.5% of crop yield in manual system **(Table 2)**. Gathering and bundling losses were almost constant at 3.5% for both system.

The average labor requirement in mechanical system was 6 man-h (skilled and unskilled)/ha as compared to 96.0 man-h/ha (unskilled) in case of manual harvesting system **(Table 2)**. The labor requirement for gathering and bundling operations for both systems were consistent at 50 man-h/ha. The total average labor requirement in mechancial system, including gathering and bundling operations was 56 man-h/ha (skilled and unskilled) as compared to 146 man-h/ha (unskilled) in the manual system.

Harvesting Cost

The cost of reaping with the JF windrower at 300 h (30 days × 10 h/day) annual use was US$13.80/ha as compared to US$21.80/ha for manual reaping as shown below:

1. Annual fixed cost of windrower (US$/year) = 358.90
2. R&M cost @7% of P/100 h use (US$/h) = 0.90
3. Tractor cost (US$/h) = 2.20
4. Labor cost for harvesting (US$/ha)
   Unskilled @US$0.23/man-h = 0.70
   Skilled @US$0.30/man-h = 0.90
5. Fuel and lubrication cost (US$/ha) = 2.40
6. Field capacity of the machine (ha/h) = 0.44

Cost of mechanical harvesting (US$/ha) [((1)/300 h + (2) + (3))/(6) + (4) + (5)] = 13.80

Cost of manual reaping (US$/ha), [96 man-h/ha × 0.23 $/man-h] = 22.10

The value of seed loss was US$38.47/ha in mechanical reaping while that in manual reaping was US$43.30/ha. The results indicate that the cost of mechanical harvesting/ha as well as the value of seed lost during reaping operation were lower as compared to that of the manual system.

Table 2. Comparison of Mechanical and Manual Harvesting Systems

Seed Losses (percent of yield)	
Manual system	
Shattering	6.6
Drop	2.9
Conveying	—
Total	9.5
Mechanical	
Shattering	3.3
Drop	—
Conveying	5.2
Total	8.5
Labor Requirement (man-h)	
Manual	
Unskilled	96.0
Skilled	—
Total	96.0
Mechanical	
Unskilled	3.0
Skilled	3.0
Total	6.0
Harvesting Cost (US$/ha)	
Manual	
300 h	21.8
250 h	21.8
200 h	21.8
150 h	21.8
Mechenical	
300 h	13.8
250 h	14.2
200 h	15.1
150 h	16.4

Break Even Area for JF Reaper Windrower

Using equation (1), the cost/ha was plotted against annual area harvested to determine the break even area for the JF windrower **(Fig. 3)**. The break even area for a single crop per year was 32 ha.

Payback Period for JF Reaper Windrower

In Pakistan, the usual span of harvesting season for rapeseed crop is 30 days. Assuming maximum 10 h/day use of the reaper, the maximum hours of work could be 300 h in a year and the machine can harvest up to 120 ha. The payback period of the machine is calculated as shown below:

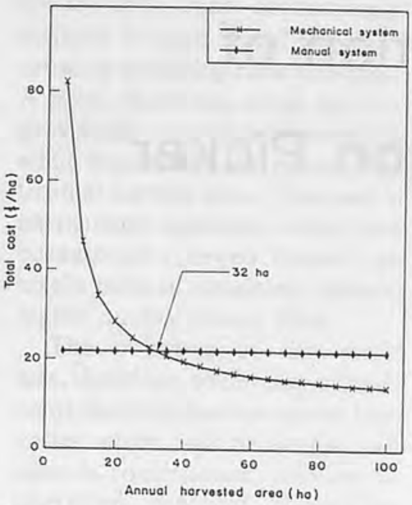


Fig. 3 Payback period of the JF reaper windrower.

1. Estimated price of machine, US$: 1 272.70
2. Effective field capacity, ha/h: 0.40
3. Cost of manual reaping, US$/ha: 21.80
4. Cost of mechanical reaping at 300 hours annual use, US$/ha: 13.80
5. Saving per hectare [3-4], US$/ha: 8.10
6. Annual area harvested [300 h/year × 0.4 ha/h], ha: 120.0
7. Net saving per season/year [5 × 6], US$: 973.10
8. Payback period [1/7] at 300 h annual use, years: 2.0

With an annual use of 250, 200 and 150 h, the cost of the windrower can be paid back in 2, 3 and 4 years, respectively.

Conclusions

The following conclusions were drawn from the result of the tests conducted:

1. Based on the combinations tested it can be concluded that the combination of 3.03 km/h of machine forward speed and 30% seed moisture content gave better performance of the machine with field capacity of 0.44 ha/h, field efficiency of 83% and seed loss of 8.5%.
2. The average machine losses after incorporating the modifications into horizontal conveyor and reel assemblies were reduced from 12.5% to 8.5% and the average field efficiency increased from 46.5% to 83%.
3. The average labor requirement of the mechanical reaper and manual reaping systems was 6 and 96 man-h/ha, respectively. The labor required for gathering and bundling operation in both cases was 50 man-h/ha.
4. The total cost for harvesting with mechanical reaper with tractor was US$13.80/ha as compared to US$21.80 for manual reaping. The break even area for annually harvesting a single crop was 32 ha.

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Field Performance Evaluation of a Manually-operated Cotton Picker

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Abstract

A hand-operated mechanical cotton picker was designed and field tested. The picker consists of an endless chain equipped with picking fingers that engage and remove cotton from the bolls. A projecting overhand prevents the stripped cotton from dropping off. The device was light enough to be carried and manipulated by one hand of an operator, yet wholly adequate in its structural aspects to stand up under continuous, vigorous use to which it was tested. The machine removed 90-95% of the cotton from the plants and put approximately 98% into the sack; the remaining 2% which fell on the ground was not significant ($p<0.05$) compared to machine harvest loss. The trash content in mechanically-harvested and hand-harvested cotton were comparable and the difference was about 17%.

Introduction

The difficulties of producing a cotton crop do not end with the cultivation of the plants to a point where the ripe cotton is ready to be picked, but extend, in no small degree, to the operation of removing the cotton. Many expedients have been devised and are described in Smith and Wilkes (1976). Generally, most cotton stripper/harvester designs are based on roller action on either side of the plant but Batchelder et al. (1961) have shown that stripping rolls covered with strips of nylon brush and solid steel stripping rolls harvested the least quantity of vegetative trash.

Past research and development efforts on cotton harvesting machines have concentrated on optimizing the size and speed of harvest without removing the cotton most expeditiously and efficiently. In effect, available cotton harvesting machinery designs have no provisions for discriminating between the cotton boll and foreign materials. The efficiency of harvesting and quantity of foreign materials in a mechanically-harvested cotton depend mainly on plant parameters (height, spacing, size and condition) and mechanical adjustment of the harvester rather than the operator (Wanjura, 1979; Wanjura and Baker, 1979). This being so, it is not unusual to have a harvest of cotton full of cotton boll carpels (burs), dried leaves and stalks (sticks), thus creating further processing problems. It has been estimated that machine-stripped seed cotton generally contains 200 to 600 kg (as much as 80%) of foreign matter (Baker et al. 1977). Although several types of seed cotton cleaning and seed extracting machines (bur machines, stick machines or combination bur and stick machines) are used at cotton gins to remove the foreign matter before ginning, they no doubt add to the cost of the lint material. Apart from the economic consideration, the presence of foreign matter puts a lot of stress on the machine components with attendant machine down time. Besides, the use of cotton cleaning machines has its own problems, too. The quantity of sellable fibres in lint cleaner wastage are often of substantial value. Mangialardi and Cocke (1979) retrieved about 5.4 kg of lint/218 kg bale from waste from two lint cleaners operated in tandem.

Current cotton harvesting machines are too sophisticated and expensive for the small-holder farmers who grow the bulk of cotton in developing countries. This is probably the reason why cotton harvesting has remained the least mechanized crop in Africa. Although hand picking of cotton is the best insurance against these nuissance, it is painfully slow. Besides, attracting people to the arduous task of hand picking of cotton has been increasingly difficult in most recent years because of dwindling rural activities. A need, therefore, exists for improvement over hand-harvesting while minimizing the quantity of trash at harvest time. This need is particularly apparent at this time because of renewed interest by textile mills in obtaining cleaner, higher quality cotton fibre.

The objective of this work was, therefore, to develop a hand-controlled mechanical cotton harvester which will be simple and light in construction, efficient in operation, practical, comprising but few parts and, therefore, inexpensive to manufacture.

Materials and Methods

A schematic drawing of the cotton picker design is shown in **Fig. 1** while a pictorial representation of the picker in use is shown in **Fig. 2**. The harvester consists of a rectangular casing, comparatively large at the rear end and diminished toward its front end. At the small end is a projecting overhanging portion which prevents the picked cotton bolls from dropping off. Extending across both ends of each harvester casing are two shafts with ends disposed in anti-friction bearings carried by the walls of the casing. Each shaft carries a sprocket wheel on which is mounted an endless chain equipped with fingers that engage and remove cotton from the bolls. A third shaft at the rear end is equipped with a spike teeth bushing arrangement at its middle that serves to knock the cotton from the fingers into a sack unit attached to the discharge opening at the base casing. Some of the spikes are arranged to work between the fingers and others at opposite sides of the fingers. The top wall of the casings are detachably slotted into closed position through grooves on the side walls so that the interiors of the harvester mechanisms may be conveniently inspected when occasion demands. The casings are constructed of light plywood and thin metal sheets. For the purpose of actuating the harvester, a crank handle is connected to the right end of the sprocket-carrying rear shaft.

In as much as most types of cotton-picking machinery have no provisions for discriminating between leaves, stalk and cotton bolls, the nature of this device is such that it puts the judgement of the operator under probation i.e., the efficiency of harvest is mainly operator dependent. To increase his efficiency as an operator, all that he has to do is to present the mouth piece of the device to a ripe cotton boll and continuously crank the handle, whereupon the chain fingers will remove cotton from the bolls and the spikes will remove the cotton from the fingers. It would show a lack of judgement to present the mouth piece to an unripe cotton boll or to a stalk full of leaves.

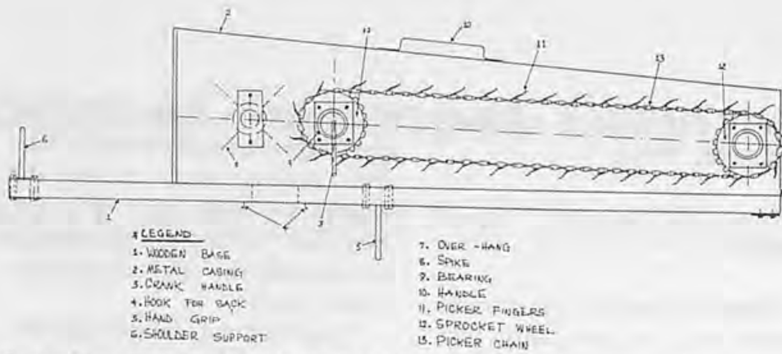

Fig. 1 Cross sectional view of cotton picker.


Fig. 2 Pictorial illustration of cotton picker in field operation.

Field Tests

The initial concept of the machine was first field-tested in 1989. Subsequent field tests in 1991 and 1993 were made after some minor amendments indicated in the first trial were incorporated. Usually about 140 kg of cotton was harvested for each test. In each instance, it was not expedient to operate at a particular speed, rather the mouth piece was directed to every cotton boll until it was either extracted, fell to the ground or found impossible to extract. The harvest loss due to the stripping operation was divided into that portion that fell on the ground (ground loss) and the cotton left on the plant (plant loss). Harvestable yield was the sum of harvested yield, ground loss and plant loss. Harvest losses were expressed as a percentage of harvestable yield. The foreign materials from a 100 g sample were removed by hand and separated into burs (stem) and sticks (stalk and leaves) to determine the fractions according to the standard ginning laboratory methods (Shepherd, 1972).

Results

Table 1 shows the category of ground loss, i.e., the cotton that fell on the ground and which decreased significantly with modifications in the design. In the first prototype, the stripping fingers were not long enough to reach the receptacle of the cotton boll. Besides, the tips were too broad and during operation, knocked off the cotton bolls rather than rake them into the chamber, causing the ground loss to be substantial. An increase in branch moisture content reduced ground harvest loss. This may be due to the fact that at lower moisture content, the bolls become overriped such that they get easily detached at the slightest contact. This trend was reversed over the years for plant harvest loss. It appeared that modification in the design caused the ground loss to decrease but this cannot be viewed in isolation. It is possible also that weather conditions, branch moisture content and plant characteristics all combine to influence the quantity of ground loss.

The trach contents of the various harvests with the machine are shown in **Table 2**. Stick content was high at low branch moisture content. It was possible that weathering factor may have affected the stick content of the trash. Delayed harvest makes the branch moisture content to decrease causing the branch, leaves and bract materials to become brittle and easily removable. Perhaps, also, the plant characteristics may have contributed to the quantity of stick in the harvests. Where the branches of the cotton plant were interposed between the picking fingers and the cotton boll it became difficult for the fingers to reach the bolls. Further manipulation caused the fingers to break off brittle dry branches which were raked into the chamber thereby increasing the trash content.

Trash contents in the seed cotton from the machine-harvest compared to hand-harvest in the variety trial are summarized in **Table 3**. Seed cotton harvested with the machine averaged 5% burs and 13% sticks (pieces of branches and stalks) as compared with less than 2% for hand-harvest.

Summary and Conclusion

A hand-operated mechanical cotton harvester for extracting and conveying cotton from the plant with a raking type action was developed and tested. This method harvested over 90% of the cotton and conveyed about 98% into the sack. Tests carried out showed that the prototype harvester performed satisfactorily having a harvesting efficiency of over 90% with a 2% delivery loss and average capacity of 18 kg/h compared to less than 15 kg/h for hand-harvest. While the device is more particularly designed for hand operation, it is obvious that the novel details of its endless chain finger mechanism can be applied to any machinery for harvesting cotton or picking paper trash or the like.

Table 1. Harvest Losses for the Trials*

Year	Branch moisture content (%w.b)	Ground harvest loss (%)	Plant harvest loss (%)
1989	13	3.9^{c}	15^{c}
1991	18	2.8^{b}	8^{a}
1993	15	2.0^{a}	10^{b}

* Values in a column not having a common subscript are significantly different at $p \leqslant 0.05$.

Table 2. Trash Content in Harvested Cotton*

Year	Branch moisture content (%w.b)	Burs (%)	Sticks (%)
1989	13	6.3^{c}	15.4^{c}
1991	18	4.1^{a}	10.3^{a}
1993	15	5.5^{b}	13.8^{b}

* Values in a column not having a common subscript are significantly different at $p \leqslant 0.05$.

Table 3. Trash Content for Machine- and Hand-harvest

Branch moisture content (% w.b.)	Machine harvest Stick (%)	Machine harvest Burs (%)	Hand harvest Stick (%)	Hand harvest Burs (%)
13	15.4	6.3	1.2	0.52
18	10.5	4.1	1.1	0.31
15	13.8	5.5	0.9	0.33
Average	13.2	5.3	1.1	0.39

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Friction Coefficient of Pigeonpea Grain with Abrasive Surfaces Used for Dehulling

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Abstract

The static and dynamic coefficients of friction were measured for different combinations of pigeonpea grain moisture content with different abrasive surfaces and rotor speed used for dehulling. With the increase in grain moisture content and surface roughness (Emery grade, Nos.20, 30 and 40) the coefficients linearly increased. Surface roughness showed more pronounced effect than moisture content. The dynamic coefficient of friction within the range of peripheral speed (12-14 m/s) was reduced with the increase in speed and decrease in surface roughness (Nos. 20-40), the former being more effective than the latter. So far as the cultivars are concerned, no significant differences were observed for C11 and No. 148 cultivars of pigeonpea under similar conditions of parameters under study.

Introduction

The frictional properties of grain are very important considerations in designing processing and handling machinery. Literature in the field reveals that the limited data on static and dynamic coefficients of friction of grain have been available for many years (Chung and Verma, 1989). The methods used by various investigators to determine friction coefficients were designed to suit the particular conditions of the material (Mohsenin, 1986). Kuppuswamy and Wratten (1970) reported that the dynamic coefficient of friction of rough rice is less than the static value. The static coefficient of friction of the grains on steel, plywood and concrete were reported to increase with the increase in moisture content (Brubarkar and Pos, 1965). The static and dynamic coefficients of friction of barley and corn on plywood and steel were studied by Bickert and Buelow (1966). They found that the friction begins to increase when certain levels of moisture content were exceeded. They concluded that conditioning of the surface by grain and vapor environment were very important factors in the friction test.

In determining the coefficient of friction of shelled corn on sheet metal, Buelow (1961) found that the effect of load on the coefficient of friction was not statistically significant. Synder et al. (1967) reported that normal pressure and velocity have little effect on coefficient of kinetic friction of wheat on steel. Mohsenin (1986) reported similarly that for the organic materials, the coefficient of friction is independent of the normal pressure. Chung et al. (1984) determined the dynamic coefficient of friction of soybeans and shelled corn on circular plates of plywood, galvanized sheet and rubber. They reported that dynamic coefficient of friction increases at higher moisture levels, but lower surface velocity has little effect. Chung and Verma (1989) reported similar results for beans and peanuts using a rotating disc on which desired surface could be fitted with the computerized data acquisition system for friction force measurement.

However, the limited published data so far on static and dynamic coefficient of friction are on the surfaces used for handling and storage of grain. The literature is scanty for the data on abrasive surfaces used for the dehulling of grain. Hence, the study was undertaken with the objectives of determining the effect of pigeonpea grain moisture content on static and dynamic coefficient of friciton with various abrasive surfaces commonly used for dehulling and at different velocities.

Equipment and Procedure

Two cultivars of pigeonpea, viz., C11 and No. 148 which were relatively difficult to dehull and commonly grown in India were selected for the experiments as raw material. The grain moisture content was held constant at 8.2 ± 0.1% wb throughout the tests, except for the study of the effect of moisture content on coefficient of friction. This level was achieved by equilibriating the grain samples over the saturated solution of magnesium chloride (Oomah et al. 1981) at an ambient temperature of 27 ± 1°C and about 50% relative humidity.

The static coefficient of friciton was determined by using a tilting platform on which the abrasive surface was changeable. An aluminum box of size 150 mm × 100 mm area of contact between grain and abrasive surface with 40 mm height was filled completely with grain. The platform was raised gradually till the box just started sliding down the slope. This angle of tilt was measured on an arc in degrees and the static coefficient of friction was expressed as the tangent of this angle. The test was repeated for different surfaces.

The response surface method was used to evaluate the static coefficient of friction with the combinations of two tractors, viz., emery grade (EG: x_1) and grain moisture content (MC: x_2) as the independent variables as these factors have linear relationship with the static coefficient of friction (Chung et al. 1984 and Chung and Verma, 1989). The first order central rotatable design was used for the experimentation with two replications (Khuri and Cornell, 1987). The experimental design with coded and uncoded factors are shown in **Tables 1** and **2**, respectively. The following mathematical model was fitted in the observed experimental data.

$$y = b_0 + b_1x_1 + b_2x_2 \qquad ...(1)$$

The dynamic coefficient of friction was determined by using the apparatus described by Chung and Verma (1989). However, the rotating plate of Tangential Abrasive Dehulling Device (TADD) described by Reichert et al. (1986) was used after the removal of the upper cover holding the cups. It was replaced by a plate with a large hole at the centre for supporting the concentric cylinder of PVC pipes (**Fig. 1**). They were suspended on central vertical shaft fitted in two bearings so that there was neither any play in the shaft nor any bending moment was caused. The inner (R_I) and outer (R_0) radius of the annular space were 155 and 200 mm, respectively. The two cylinders were fixed by four radial partitions so that the whole unit was free to rotate with suspension shaft. A clearance of 0.5 mm was provided to prevent escaping of grain out of the annular space. The grain was covered with four plates, one in each segment.

Piezo-electric crystal cell was used for the measurement of resistance force. The cell was enclosed in a barrel (**Fig. 2**) and the transducer was mounted as shown in **Fig. 1**. The output of the transducer was fed to the recorder (IFICOS Strip Chart Recorder No. 3250) through the amplifier (IFICOS Piezo-electric Pressure Measuring Unit). The tail piece of angle iron was fixed on the outer cylinder to exert the resistance force completely on the transducer. The system was calibrated

Table 1. Factors and Their Levels for Measurement of Coefficient of Friction

Factor	Symbol		Levels	
	Coded	Uncoded	Coded	Uncoded
Emery grade. No.	x_1*	EG	−1	20
			0	30
			+1	40
Grain moisture content, % (wb)	x_2	MC	−1	6
			0	9
			+1	12
Rotor speed, m/s	x_2	RS	−1	12
			0	14
			+1	16

* x_1 (EG) was used with x_2 (MC) and x_2 (RS) in two separate experiments.

Table. 2 Static and Dynamic Coefficient of Friction for Different Combinations of Emery Grade and Grain Moisture Content

Expt No.	x_1 (EG)	x_2 (MC)	Static coefficient of friction		Dynamic coefficient of friction at 0.11 m/s	
			C11	No. 148	C11	No. 148
1	−1 (20)	−1 (6)	0.6008	0.6068	0.4683	0.4614
2	−1 (20)	−1 (6)	0.5949	0.6128	0.4717	0.4578
3	+1 (40)	−1 (6)	0.4986	0.4986	0.3816	0.3820
4	+1 (40)	−1 (6)	0.4877	0.5040	0.3740	0.3894
5	−1 (20)	+1 (12)	0.6432	0.6370	0.5207	0.5227
6	−1 (20)	+1 (12)	0.6494	0.6494	0.5318	0.5330
7	+1 (40)	+1 (12)	0.5373	0.5486	0.4368	0.4307
8	+1 (40)	+1 (12)	0.5486	0.5372	0.4290	0.4388
Regression coefficients						
b_0	—	—	0.570	0.574	0.452	0.452
b_1	—	—	−0.052	−0.052	−0.046	−0.042
b_2	—	—	0.024	0.019	0.028	0.029

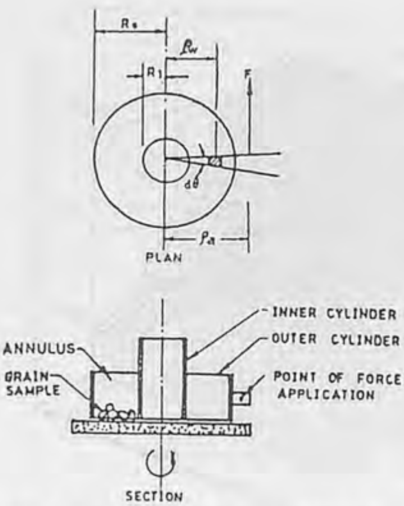


Fig. 1 Device for determination of dynamic coefficient of friction.

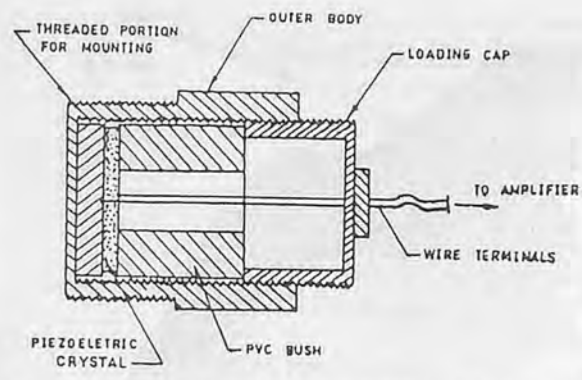


Fig. 2 Arrangement of piezo-electric crystal inside the transducer.

by using standard weights and adjusting the sensitivity of the recorder.

Similar experimental plan as shown in **Table 1** viz., emery grade and grain moisture content were used to determine the dynamic coefficient of friction in two replicates. The average torque arm between the cylinders was used as a weighted radius (R_w) to determine the average linear velocity between the disc and sample (Chung et al. 1984) which was calculated as.

$$R_w = \frac{2}{3}\left(\frac{R_o^3 - R_i^3}{R_o^2 - R_i^2}\right) \qquad ...(2)$$

and found to be 80.66 mm for R_o = 200 mm and R_i = 115 mm.

Each annular segment was filled with 100 g grain and covered with a plate weighing 100 g, the total normal load (P) on rotor disc being 800 g. The disc was rotated by an electric motor (187 W) at a speed of 110 mm/s (Chung and Verma, 1989). The resistance force (F) was measured at radius Rf from the centre of the suspension shaft (121 mm). Then the dynamic coefficient of friction (Gd) is given by

$$u_d = F.R_f/P.R_w \qquad ...(3)$$

Results and Discussion

The experimental results are presented in **Tables 3** and **4** and **Figs. 3** through **5**. The first order multiple regression model was fitted for these data.

Effect of Emery Grade and Grain Moisture Content

Table 2 shows the observed data for the different combinations of emery grade and grain moisture content along with the partial regression coefficients for each model. It could be observed that the coefficient for emery grade (b_1) was negative for all the four models of static and dynamic coefficients of friction. This indicates that friction coefficient is inversely proportional to the emery grade, i.e., with the decrease in number (decrease in roughness), the coefficient was found to increase. The coefficient of grain moisture content (b_2) was positive indicating that the prediction of magnitude of response increased with the increase in grain moisture content. This was due to the increased adhesion between the grain and emery surfaces at higher moisture values (Chung and Verma, 1989).

Table 3. Dynamic Coefficient of Friction for Different Combinations of Emery Grade and Rotor Speed

Expt No.	x_1 (EG)	x_2 (RS)	Dynamic coefficient of friction C11	No. 148
1	−1 (20)	−1 (12)	0.4028	0.4048
2	−1 (20)	−1 (12)	0.4060	0.4982
3	+1 (40)	−1 (12)	0.3703	0.3640
4	+1 (40)	−1 (12)	0.3641	0.3716
5	−1 (20)	+1 (16)	0.3874	0.3825
6	−1 (20)	+1 (16)	0.3830	0.3861
7	+1 (40)	+1 (16)	0.3506	0.3461
8	+1 (40)	+1 (16)	0.3475	0.3493
Regression coefficients				
b_0	—	—	0.376	0.376
b_1	—	—	−0.016	−0.019
b_2	—	—	−0.009	−0.010

Table 4. Analysis of Variance for Coefficient of Friction for Different Combinations of Emery Grade, Grain Moisture Content and Rotor Speed

Source	df	Coefficient of friction for EG and MC combinations: Static	At 0.11 m/s	Dynamic coefficient of friction for EG and RS combinations
C11				
Regression (model)	2	0.026420*	0.023200*	0.003318**
Residual	5	0.000389	0.000357	0.000070
Lack of fit	1	0.000230	0.000229	0.000070
Pure error	4	0.000159	0.000128	0.000039
F-ratio (LOF)		5.78	7.15	7.18
Variability explained		98.54	98.48	96.82
No. 148				
Regression (model)	2	0.024520*	0.020840*	0.003688**
Residual	5	0.000292	0.000288	0.000127
Lack of fit	1	0.000119	0.000169	0.000099
Pure error	4	0.000173	0.000119	0.000028
F-ratio (LOF)		2.75	5.68	3.82
Variability explained, %		98.82	98.63	97.62

Significant at *0.10% and **1.0% levels.

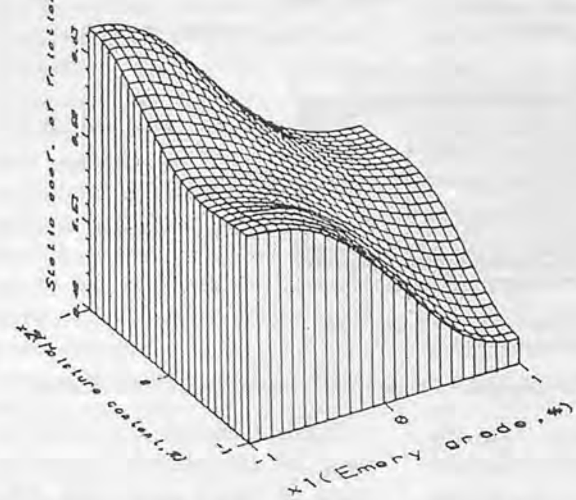


Fig. 3 Response surface of static coefficient of friction as influenced by emery grade and moisture content of pigeonpea grain (C11).

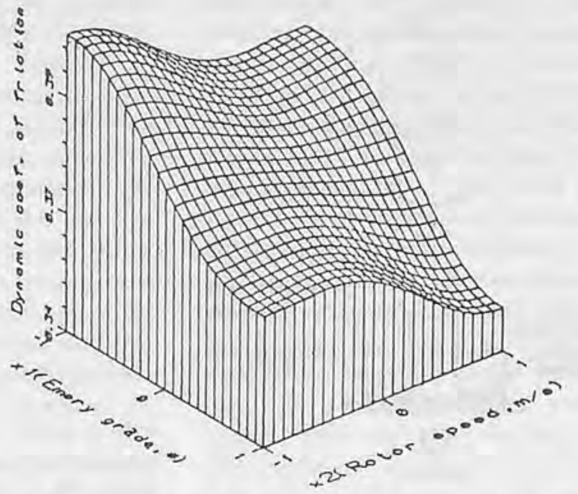


Fig. 4 Response surface of dynamic coefficient of friction (at 8% mc wb) as influenced by emery grade and rotor speed for C11.

The higher magnitude of b_1 than that of b_2 shows that emery grade had pronounced effect on coefficient of friction than grain moisture content.

Under the dynamic condition, the magnitude of coefficient for emery grade (b_1) was found to be reduced and for grain moisture content, it was increased. It is obvious from the response surfaces that the static coefficient of friction (**Fig. 3** was higher by about 40% to 75% as compared to the dynamic coefficient of friction, mainly due to the intercept, i.e., b_o in both conditions (**Table 2**). This could be due to the fact that under dynamic condition, the sliding friction was converted into rolling friction since both the surfaces in contact roll on each other. However, the variation of static and dynamic coefficients between the two cultivars was not considerable exhibiting the same pattern.

Effect of Emery Grade and Rotor Speed

The observed data for different combinations of emery grade and rotor speed are presented in **Table 3** for the two cultivars of pigeonpea under study. Both the partial regression coefficients, i.e., b_1 and b_2, were negative in this case, indicating that the dynamic coefficient of friction increased with the decrease in emery grade number (increase in roughness) but decreased with the increase in rotor speed (sliding velocity between the grain and abrasive surfaces) (**Fig. 4**. Thus, the dehulling operation at higher rotor speed and finer emery grade may require less power, which was clear in the lower resistance force recorded.

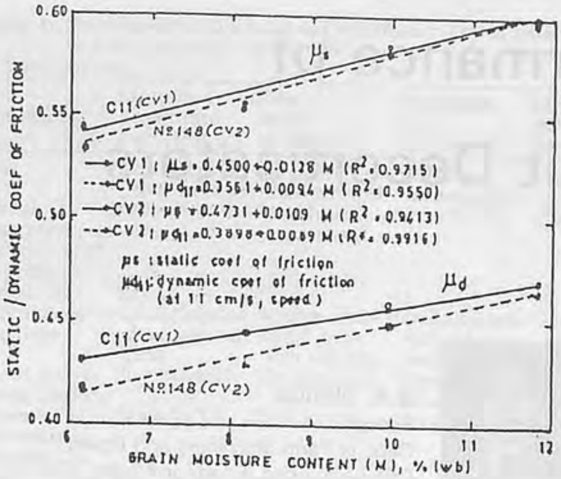
Statistical Analysis

Table 4 presents also the analysis of variance for all the above six models developed. The regression of the four models was highly significant ($P<0.001$) for the moisture content study and two models for rotor speed were significant at $P<0.01$. The lack of fit was non-significant for all the models with the statisfactory percentage variability explained (R^2 values). Hence, the models can be considered quite adequate for prediction of the respective coefficient of friction within the experimental region.

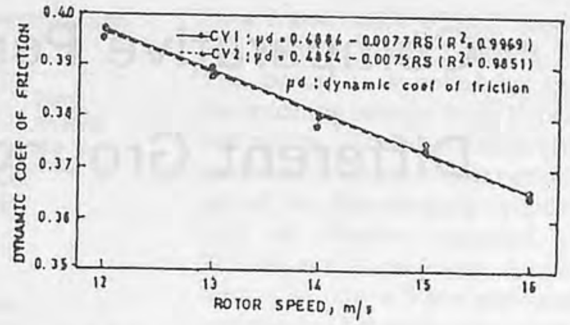
Variation of Coefficient of Friction with Sieve

The results are presented in **Fig. 5** for the effect of grain moisture content and rotor speed on static and dynamic coefficient of friction in contact of the sieve (1.5 mm diameter holes). It was observed that the coefficient of static and dynamic friction increased linearly when grain moisture content was increased from 6.15% to 11.76% (wb). The regression equations are presented in **Fig. 5a** with the variability explained between 94.13% to 99.16%. It was also observed that the static coefficient of friction was slightly less for C11 than that of No. 148 whereas the dynamic coefficient of friction at 0.11 m/s, the differences were slightly higher at higher grain moisture content level.

It could be seen from **Fig. 5b** that the dynamic coefficient of friction at higher rotor speed (12-16 m/s) also shows a linear relationship. The magnitude of dynamic coefficient of friction decreased with the increase in rotor speed. Although, little variation was observed for the data for C11 and No. 148 cultivars, both exhibited the regression coefficients of prediction equations very close to each other with



(a) Effect of grain moisture content



(b) Effect of rotor speed

Fig. 5 Variation of static and dynamic coefficient of friction with effect of grain moisture content (a) and rotor speed (b) with sieve (1.5 mm ϕ).

99.69% and 98.51% variability explained, respectively.

Conclusions

1. The static and dynamic coefficients of friction linearly increased as the grain moisture content and surface roughness increased. The latter showed more pronounced effect than the former.
2. The dynamic coefficient of friction (at 0.11 m/s) was reduced by about 40% to 75% than the static coefficient of friction under similar conditions of grain moisture content and surface roughness.
3. The dynamic coefficient of friction at higher speed (12-14 m/s) was further decreased with an increase in speed and decrease in surface roughness — the speed being more effective than the latter.
4. No significant differences were observed in static and dynamic coefficients of friction for C11 and No. 148 cultivars under similar conditions of grain, and type and speed of an abrasive surface.
5. Similar results were observed on the emery grades for the sieve surface (1.5 mm diameter hole) for both cultivars.

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Comparative Performance of Different Groundnut Decorticators

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Abstract

Many types of efficient groundnut decorticators are now available requiring low initial investment. The present study was undertaken to compare the cost of groundnut decortication and cleaning by various decorticating methods. The 1-hp electric motor operated groundnut decorticator cum cleaner has a minimum cost of Rs. 0.05/kg as compared to Rs. 0.24 Rs. 0.17, Rs. 0.13 and Rs. 4.00 by rotary hand-operated, oscillating hand-operated, oscillating pedal-operated and by conventional method of hand picking, respectively. Farmers can select a suitable groundnut decorticator as per their requirement and investment capabilities.

Introduction

In Orissa, groundnut is one of the most important oil seed crops. It is grown on an area of about 396.7 × 10^3 ha with a productivity of 1 493.1 kg/ha. The total production of groundnut in Orissa has rapidly increased over the last 10 years and is expected to increase further in the coming decade. The necessity for large quantities of costly groundnut seed delays the sowing time, if decortication is done by the conventional method of hand picking. In the conventional method of hand-picking, the groundnut pods are shelled or decorticated by pressing the pods by hand, by which the output is 5 kg/day. By this method, it is difficult to meet the demand at the rate of 120 kg of seeds/ha in the sowing season. Therefore, a labour saving machine, i.e., a mechanical decorticator helps a farmer towards timely decortication and minimization of labor involved in manual decortication.

Important factors affecting the mechanical decortication are pod moisture content, type and peripheral speed of decorticating elements. The comparative performance on the basis of cost economy were evaluated for the most commonly grown groundnut varieties in Orissa, i.e., AK-12-24 and ICGS-44.

Objectives

The objectives of the study were:

1. To compare the performance of four groundnut decorticators with that of the conventional method of hand-picking.
2. To recommend a suitable decorticator for the local farmers based on good performance and low cost of operation.
3. To evaluate the break-even use of the machines.

Materials and Methods

The construction feature of the groundnut decorticating machines compared in this present study is described below and in **Table 1**.

The motorized groundnut decorticator cum cleaner has a decorticating cylinder, a sieve, a blower, a separating chute and a hopper, all of which are mounted

Table 1. Specification and Results of Performance Test of Different Groundnut Decorticators (as per IS: 11473-1985).

I. Specification

Particulars	Motorized decorticator cum cleaner	Rotary hand-operated	Oscillating hand-operated	Oscillating pedal-operated	Hand-picking
Type	1 hp electric motor-operated	Manually hand-operated	Manually hand-operated	Manually pedal-operated	
Over-all dimensions, cm (length × width × height.)	80.5 × 51 × 115	39 × 42 × 73.5	50.5 × 30 × 95	120 × 33 × 138	
Weight, kg	62	15	13	36	
Decorticating unit type	C.I. Cylindrical drum with peg teeth	Wooden Cylindrical drum with rasp bars	Oscillating arm with C.I. peg teeth shoes	Oscillating arm with C.I. peg teeth shoes	
No. of decorticating elements	Six, detachable	Six, integrated	Three, detachable	Three, detachable	
Dimension of decorticating elements, cm	21 × 4.5 × 2.4	7 × 0.7 × 0.9	21 × 4.5 × 2.4	21 × 4.5 × 2.4	
Height of pegs, cm	0.9	—	0.9	0.9	
Sieve (concave) grate size, cm	4.5 × 0.9	2.0 × 0.9	4.5 × 0.9	4.5 × 0.9	
Sieve (concave) clearance, cm	1.9-3.0	1.9-2.8	2.2-3.4	2.2-3.4	
Cylinder dia, cm	22.5	20	—	—	
Rocking arm radius, cm	—	—	26	26	
Power transmission system	'V' Belt and pulley	—	—	—	—
Provision of blower	Provided (2850 rpm)	—	—	—	—
Variety of crops used in test	ICGS-44	AK-12-24	ICGS-44	ICGS-44	ICGS-44
Optimum speed	215 rpm	45 rpm	45 cycles	65 cycles	—
Optimum sieve clearance, cm	1.9	2.6	1.9	2.6	
Optimum capacity, kg/h	170	30	55	75	0.63
Decorticating efficiency, %	99	96	96.5	96.5	100
Breakage, %	1.5	4.6	4.2	6.3	Negligible

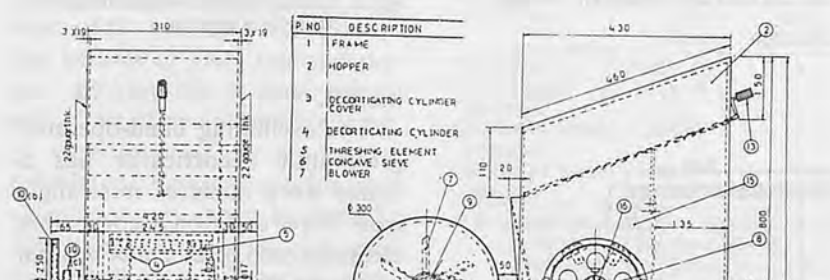

Fig. 1 Design details of low cost motorized groundnut decorticator.

on a frame made of m.s. angle iron as shown in **Fig. 1** and **Fig. 2**. The power is supplied to the decorticating cylinder by a 'V' belt and pullies. Six decorticating elements are arranged on the periphery of the decorticating cylinder with an effective diameter of 22.5 cm and 21 cm length. A m.s. sieve of 45 mm × 9 mm grate size is supported below the decorticating cylinder with a minimum clearance of 19 mm, which can also be increased. A blower is provided to blow the stream of air under the sieve for separation of kernels and hulls. The dry, broken shells are blown upwards along with the air stream and the seeds are collected at the bottom of the chute. A m.s. handle is provided to rotate the decorticating cylinder manually for decortication, if required; when power is not available.

The manually-operated rotary groundnut decorticator has a wooden frame and raspbar type wooden decorticating cylinder of 20 cm diameter and 15 cm width as shown **(Figs. 3 and 4)**. A m.s. sieve with grate size of 20 mm × 9 mm is fixed below the cylinder at a minimum clearance of 19 mm, which can be increased up to 28 mm. The cylinder shaft is rotated by hand with a m.s.

Fig. 2 Motorized groundnut decorticator in operation.

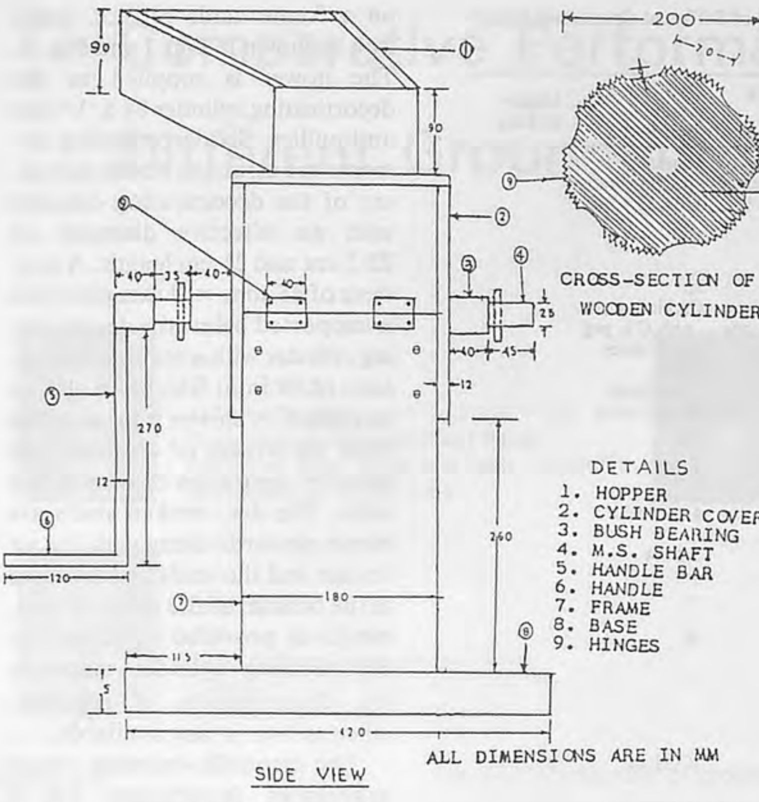

Fig. 3 Rotary hand-operated decorticator.

Fig. 4 Rotary hand-operated decorticator in operation.

Fig. 5 Oscillating hand-operated decorticator in operation.

Fig. 7 Oscillating pedal-operated decorticator in operation.

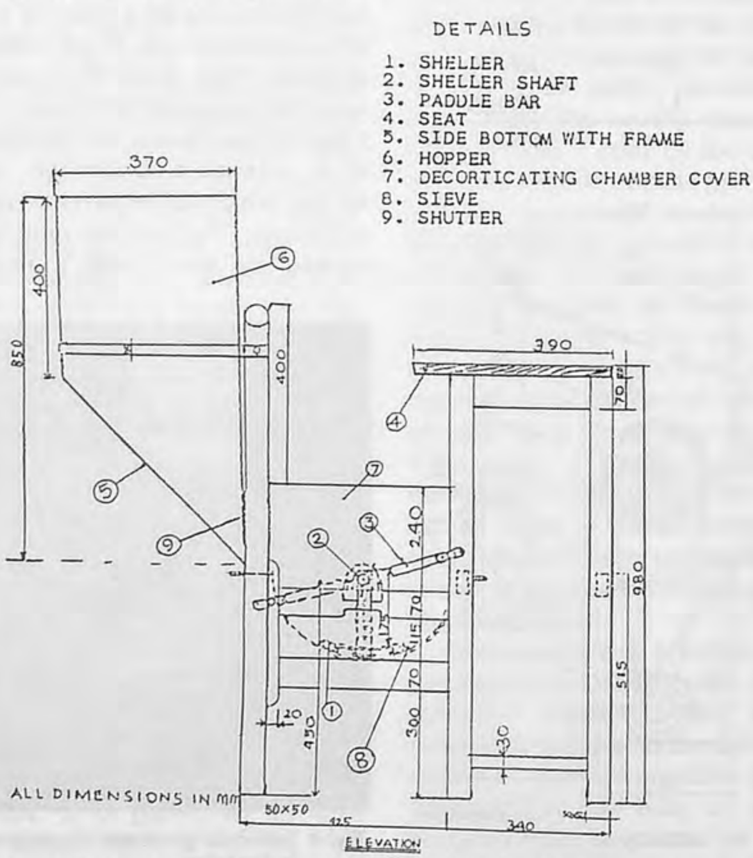

Fig. 6 Design details of pedal-operated groundnut decorticator.

handle.

The oscillating hand-operated groundnut decorticator has a frame work made of m.s. angle iron. Three cast iron decorticating elements each of 21 cm × 4.5 cm with pegs of 0.9 cm height are attached to the bottom end of 79 cm long oscillating handle **(Fig. 5)**. Bush bearings are provided to support the handle. A concave sieve with 45 mm × 9 mm holes is fixed below the decorticating elements at a minimum clearance of 19 mm, which can be increased.

The oscillating pedal-operated groundnut decorticator has a G.I. hopper to store 20 kg of groundnut. A pair of pedals to oscillate the decorticating elements are attached to a 26 cm long arm. A sieve and a wooden frame work with a seat for the operator are its major components (**Figs. 6** and **7**). The decorticating elements and sieve are of similar number and specification as the oscillating hand-operated groundnut decorticator. The hopper has a shutter to control the feed rate. The concave sieve is placed at a clearance adjustable from 22 mm to 34 mm.

Treatments

The following treatments were evaluated and their specification are given in **Table 1**.

i. Motorized groundnut decorticator cum cleaner.
ii. Rotary hand-operated groundnut decorticator.
iii. Oscillating hand-operated groundnut decorticator.
iv. Oscillating pedal-operated groundnut decorticator.
v. Conventional method of hand-picking.

Evaluation

For each groundnut decorticator, the output capacity, i.e., the quantity of pods decorticated per hour, decorticating efficiency, i.e., ratio of the amount of decorticating kernels to total kernel input per unit time, % of broken kernels. Cost of operation (fixed cost + variable cost) per hour were calculated. Assumptions for cost calculations are shown in **Table 2**. The break-even use in operating hours and its equivalent in quintals were calculated and presented in **Table 3**. The comparative performance is shown in **Fig. 8**.

Table 2. Cost Calculation for Decortication and Cleaning by Different Groundnut Decorticators (as per IS: 9164-1979).

Particulars	Motorized decorticator	1 hp motor	Rotary hand-operated	Oscillating hand-operated	Oscillating pedal-operated	Hand-picking
Useful life (L) Yrs (h)	8 (2 500)	15 (15 000)	5 (1 000)	5 (1 000)	5 (1 000)	
Annual use, h	312.5	1 000	200	200	200	
Cost of the machine, (P) Rs.	2 000.00	3 500.00	440.00	700.00	1 000.00	
Salvage value Rs. (S) (5%P)	100.00	175.00	22.00	35.00	50.00	
Annual Depreciation, Rs. (D = p-s)/L	237.50	221.66	83.60	133.00	190.00	
Annual interest, Rs (18% p + s)/2	189.00	330.75	41.58	66.15	94.5	
Annual Housing, Rs (H = 1.5% (p + s)/2	15.75	27.56	3.46	5.51	7.87	
Total annual fixed cost (F.C = D + I + H)	442.25	579.97	128.64	204.66	292.37	
Fixed cost, Rs/h	1.41	0.58	0.64	1.02	1.46	
Repair and maintenance cost (R = 12%P)/Rs/h, A	0.77	0.42	0.26	0.42	0.60	
Labour required	1 male @Rs25/8h		2 female @Rs.20/8h	2 males @Rs.25/8h	2 males @Rs.25/8h	1 female @Rs.20/8h
Labour charge, Rs./h	3.13	—	5.00	6.25	6.25	
Power cost @Rs.0.95/kWh	—	0.70	—	—	—	—
Total variable cost (V.C) Rs/h	3.90	1.12	5.26	6.67	6.85	
Total cost, Rs/h (F.C + V.C)	5.31	1.70	5.90	7.69	8.31	
Over head charges (20%), Rs/h	1.06	3.34	1.18	1.54	1.66	
Total cost, Rs/h	6.37 + = 8.41	2.04	7.08	9.23	9.97	
Optimum capacity kg/h	170		30	55	75	5kg/day
Cost, Rs/kg	0.05		0.24	0.17	0.13	4.00

Table 3. Break Even Use (B.E.U) of Groundnut Decorticators

Type of groundnut decorticator	Annual ownership cost (Rs.)	Custom rate (Rs./h.)	Operating cost of machine (Rs/h.)	B.E.U. (h.)	Optimum capacity (kg/h.)	B.E.U. (Qntls.)
Motorized	1 022.22	8.41	5.72	381.4	170	648.4
Rotary hand-operated	128.64	7.08	5.26	70.7	30	21.2
Oscillating hand-operated	204.66	9.23	6.67	80.00	55	44.0
Oscillating pedal-operated	292.37	9.97	6.85	93.7	75	70.30

N.B.: Break even use (h) = [Annual ownership cost (Rs.)]/[Custom rate (Rs./h) − Operating cost of machine (Rs/h)].
Break even use (Quintal) = B.E.U. (h) × Optimum capacity (qntl/h).

Results and Discussion

The performance of the motorized, rotary hand-operated, oscillating hand-operated and oscillating pedal-operated groundnut decorticators were evaluated and compared with the conventional method of hand-picking (**Tables 1** and **2** and **Fig. 8**).

The motorized groundnut decorticator cum cleaner has an output capacity of 170 kg/h with a decorticating efficiency of 99% and a breakage of 1.5%. The decorticating cost per kg with this machine was Rs. 0.05 with separation of shells and kernels.

The rotary hand-operated decorticator has an output capacity of 30 kg/h with 96% decorticating efficiency and 4.6% breakage. The decortication and cleaning cost per kg with this machine is Rs. 0.24.

The oscillating hand-operated decorticator has an output capacity of 55 kg/h with 96.5% decorticating efficiency and 4.2% breakage. The decortication and cleaning cost per kg with this machine is Rs. 0.17.

The oscillating pedal-operated decorticator has an output capac-

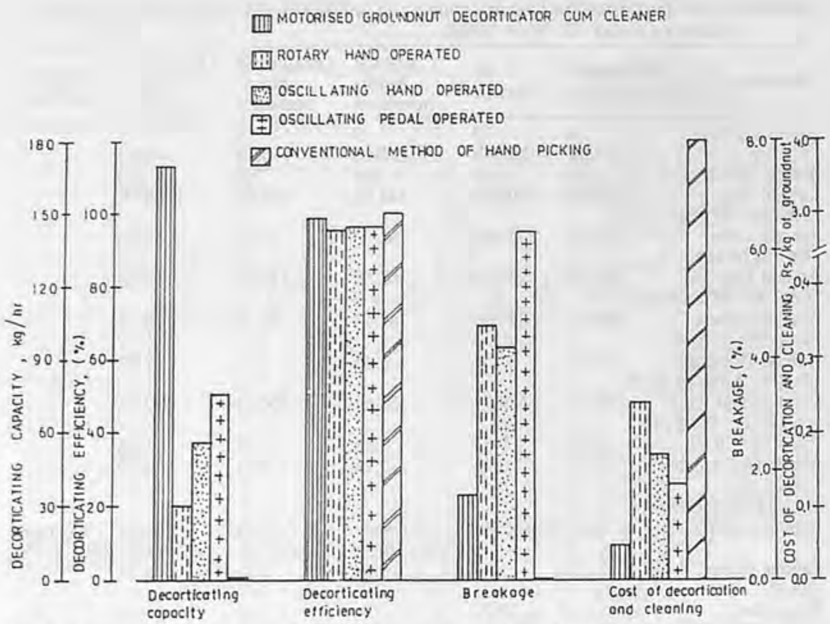

Fig. 8 Comparative performance of different groundnut decorticators.

ity of 75 kg/h with 96.5% decorticating efficiency and 6.3% breakage. The cost of decortication and cleaning per kg with this machine is Rs. 0.13.

In conventional hand-picking method, the decorticating capacity/h by a female worker is 0.63 kg and the cost of decortication and cleaning per kg is Rs. 4.00 having negligible breakage and almost 100% decorticating efficiency.

Conclusion

The cost of groundnut decortication, including cleaning of all the four different decorticators were evaluated and presented in (**Tables 2** and **3** and **Fig. 8**).

The cost of decortication, including cleaning by hand-picking method is Rs. 4.00/kg, which is highest among all the methods.

The cost of decortication, including cleaning per kg for the rotary hand-operated decorticator is Rs. 0.24 and the break-even use is 70.7 h equivalent to 21.2 quintals of groundnut decortication.

The cost of decortication, including cleaning per kg of groundnut for the oscillating hand-operated decorticator is Rs. 0.17 with a break even use of 80 h equivalent to 44 quintals.

The cost of decortication, including cleaning per kg for the oscillating pedal-operated decorticator, which is a modified version of the oscillating hand-operated decorticator is Rs. 0.13 with a break even use of 93.7 h equivalent to 70.3 quintals.

The cost of decortication including per kg for the motorized groundnut decorticator is Rs. 0.05, which is lowest among all the four machines with a break even use of 381.4 h equivalent to 648.4 quintals.

All the decorticators evaluated have distinct advantages over the conventional method because of their much lower decortication and cleaning cost per kg and they were developed chronologically with a reduction in cost of decortication and cleaning per kg of groundnut.

The motorized groundnut decorticator can be used where electricity is available. Even during power failure, it can be used as a hand-operated decorticator. The machine can be purchased for use by medium and large farmers, which also can be used by the unemployed rural youth for custom hiring purpose, because of its lowest cost of decortication, including cleaning. Rotary hand-operated and oscillating hand-operated decorticators are suitable for use by small farmers, where electricity is not available. The pedal-operated groundnut decorticator can be used by small and medium farmers as well as for custom hiring purpose due to its highest capacity among all the manually operated groundnut decorticators.

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Simulation of Engine-wasted Heat Stationary Deep-bed Rough Rice Dryer

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Abstract

Research into minimizing post harvest losses and the effective utilization of energy resources for crop drying has led to the idea to use wasted heat from the internal combustion engine. An attempt was made to measure the availability of engine waste heat that can be harvested for grain drying at different speeds of the engine. A static deep-bed rough rice dryer is proposed on the basis of measured available engine-wasted heat. With the measured drying airflow characteristics, a theoretical analysis on the proposed dryer performance is made with the help of partial differential equation (DE) model of deep-bed drying of grain by considering different depth of rough rice bulks in the dryer. Some information on the calculation of engine-wasted heat for crop drying, fan requirement and specification for design of flat-bed dryer are also presented.

Results have shown promise for this type of grain drying unit, especially in major rice growing regions where the same engines which are used for pumping irrigation water and rice milling purposes can also be used for grain drying.

Introduction

The most common and pressing problem in handling wet grain in most non-industrialized countries is delay in sun drying caused by unfavorable weather conditions. Food grains in non-industrialized countries are dried by the traditional method of sun drying. Usually rural farmers spread the grains on beaten earth or woven mats and the grains are submitted to adverse effects caused by dust, rain, wind, insect and rodent attack which result in the quality of the dried grains being low. Considerable losses ranging from 10% to 25% can often occur (Excell, 1980). Climatic conditions uch as the temperature and relative humidity have great influence on sun drying. In most tropical countries, the climate is characterized by hot and humid air during harvesting of paddy. Such air would be limited use for drying rough rice. The harsh climatic condition dictates the need for a more effective method of drying grain. Because of the difficulties of sun drying mostly in the wet season, this method has no future for complete drying of grains in humid tropical countries.

An attempt was made for the utilization of engine-wasted heat for grain drying as indicated by Someone et al. (1973) with a small dryer bed area and grain depth. They concluded that the energy requirement for grain drying can be minimized with the use of large bed area, low air temperature and low air velocity.

Abe et al. (1992) reported the utilization of engine-wasted heat for grain drying with a dryer capacity of 140 kg of rough rice. They found less kernel breakage than rice dried in the sun or dried too rapidly with highly heated air. They used a separate electric motor as a power source to drive the dryer fan in order to harvest engine-wasted heat, which was a serious drawback of their work.

A review of current literature revealed that very little work has been done on engine-wasted heat dryers. Data are not available on the performance of a dryer, dependent on engine-wasted heat as their source, upon which design decisions could be made. Engine-wasted heat drying systems must be properly designed in order to meet particular drying requirements of specific crop and to give satisfactory performance with respect to energy requirements. The designer should investigate the basic parameters, namely; dimensions, temperature, relative humidity, air flow rate, drying capacity and energy requirement on the basis of available engine-wasted heat. However, full scale experimentation for different products, drying seasons and system configuration is sometimes costly and not possible. It is an established fact that, in general, the partial differential equation (DE) model of deep-bed drying of grain is capable of predicting the performance of stationary deep-bed dryer, over a wide range of conditions, with reasonable accuracy. So the development of simulation model, on the basis of available engine-wasted heat for drying, is a valuable tool for predicting the performance of engine waste heated dryer. For this purpose partial DE model of deep bed drying of grain as indicated by Sharp (1982), Bala (1990), Brooker et al (1992), based on the laws of heat and mass transfer, is used. The objectives of the studies are:

(i) to find out the availability of engine-wasted heat that can be harvested for grain drying; and
(ii) to simulate the engine-wasted heat dryer performance on the basis of available engine-wasted heat.

Materials and Methods

Engine-fan Combination and Dryer Duct System

Figure 1 is a schematic diagram of the vertical section of the engine-fan combination dryer duct system and plenum chamber constructed for this study and the proposed flat-bed dryer chamber. For this study an air-cooled gasoline engine working on four-stroke cycle having an engine displacement of 105 cc, PS of 1.95 at 3 000 rpm and maximum PS of 2.5 at 4 200 rpm, fuel consumption of 0.25 kg/(PS. h) and a cooling efficiency of 30% is used. A 60-watt back ward curve centrifugal fan having a maximum air flow rate of 17 m^3/min and static pressure of 27 mmAq at a maximum speed of 2 000 rpm was used. Fan is directly coupled with the engine camshaft. The wasted heat from the engine is used to heat up the air being forced through the duct system to the proposed dryer. The proposed dryer is considered an open ended plywood box 90 × 90 cm in cross section and 50 cm deep. A PVC pipe, diameter 125 mm and length 1 225 mm, was used to connect the fan housing with the lower part of the gradually expanding plenum chamber via a 90-degree elbow (**Fig. 1**). The upper part of the plenum chamber, 90 × 90 cm in cross section, is considered to be connected to the lower part of the proposed dryer in order to facilitate the uniform air flow throughout the dryer base area. Proper insulation was given thoughout the duct system, including fan housing so that no heat can be lost by conduction. As the engine is 4-stroke cycle, the rpm of fan is half of the engine crankshaft rpm. To monitor the drying air temperatures at different air flow rates, copper constantan themocouple probes were connected at (A), the end of the straight duct and also at the entrance (B), (C) and (D) of the proposed dryer base (**Fig. 1**). Two thermocouples were connected to the engine fin surfaces in order to know the temperature of the engine fin at different air flow rates. Two thermocouples were also used to record the ambient dry-bulb and wet-bulb temperatures. The thermocouple probes were connected through an interface of AD converter (Green kit 77A model) then to personal computer for data collection. The temperature readings from the thermocouple probes were recorded every minute. Air flow rate was measured indirectly by connecting manometers at (A), the end of the straight duct.

Engine-wasted Heat

The current gasoline fed internal combustion engine has about 30% effective output from the

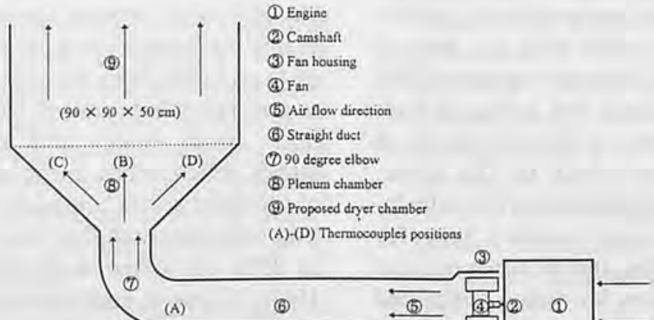

Fig. 1 Schematic diagram of the vertical section of the engine-fan combination dryer duct system and plenum chamber used in this study and the proposed engine-waste heated dryer chamber.

energy derived from burnt fuel, another 30% is lost through the exhaust, radiation and friction accounts for 10%, and the remaining 30% is lost through cooling system. The heat energy required to heat up the drying air is mostly derived from the engine cooling system heat loss. The cooling load of the engine at any given rpm can be determined by the following equation:

$$WHE = \eta_c \times \beta \times c_f \times \eta_f \times bp \quad (1)$$

where,

WHE = the waste heat energy released from the engine cooling system to the air, MJ/h

η_c = the cooling system efficiency, decimal

β = the fuel consumption rate, kg/(PS.h)

c_f = the calorific value of fuel, MJ/kg

η_f = the fuel efficiency, decimal

bp = the amount of break power of the engine at any rpm, PS.

Fuel efficiency is considered 78% and the average calorific value of gasoline is 44 MJ/kg. Thus, from Eqn. (1), the wasted heat released to the outside air by the cooling system at any given rpm of the engine is

$$WHE = (0.3 \times 0.25 \times 44.0 \times 0.78) \times bp = 2.574 \times bp \text{ MJ/h} \quad (2)$$

The total heat energy to be utilized to heat the drying air is calculated from the following equation:

$$HEA = 0.06 \times Q_a \times c_a \times p_a \times (T_a - T_{am}) \quad (3)$$

where,

HEA = the amount of heat energy absorbed by the drying air, MJ/h

Q_a = the volume flow rate of drying air, m^3/min.

c_a = the specific heat capacity of drying air, kJ/(kg°C)

p_a = the density of drying air, kg/m^3

T_a = the dryig air temperature, °C

T_{am} = the ambient temperature, °C

0.06 = the units conversion factor.

Generally, for design purpose an air volume of 1 m^3/(PS.min) is considered appropriate for an air-cooled engine. However, when the fan is directly coupled to the engine camshaft the air volume is increased during operation of the engine. The temperature of the discharge air is lowered as the air volume is increased.

Partial Differential Equation (DE) Model

The following simplified assumptions were made for deriving the DE model for deep-bed drying of rough rice by engine-wasted heat:

(a) Air flow is one dimensional;
(b) Conduction heat loss within the bed is negligible;
(c) Specific heats of dry grain, moisture and air are constant;
(d) The bin walls are adiabatic, with negligible heat capacity;
(e) Latent heat of vaporization is dependent on the moisture content;
(f) The temperature gradients within the individual kernels are negligible;
(g) The volume shrinkage of rough rice is negligible during the drying process;
(h) The single-kernel drying equation and the moisture equilibrium equation are accurate; and
(i) Air flow rate, drying air temperature and its relative humidity is considered constant through the drying period.

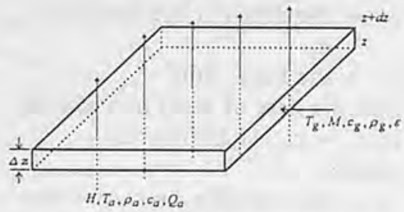

Fig. 2 Element of a rough rice bed in the proposed engine-waste heated dryer.

From knowledge of the physical characteristics of convective heat and mass transfer, and accounting for heat and mass balance in the drying process, a set of 4 partial defferential equations can be derived (Sharp, 1982; Bala, 1990; Brooker et al. 1992). Let us consider the mass and heat exchanges in an element of rough rice bed of a unit cross-sectional area and depth (z, z + dz), m over a time interval dt, min as shown in **Fig. 2**. There are four unknowns in this model: M, the average rough rice kernel moisture content (decimal dry basis); H, the air humidity ratio; T_a, the drying air temperature (°C); and T_g, the rough rice kernel temperature (°C). Thus two conservation equations and two rate equations are made resulting in four equations.

For the element bed of unit cross-sectional area:

Mass of rough rice grain in depth (z, z + dz) is p_g dz kg

where,

p_g = the bulk density of rough rice, kg/m^3

dz = the depth increment, m.

Mass of air within rough rice layer is p_a ϵ dz kg

where,

ϵ = the void ratio, decimal.

Mass of air passing through the layer in time dt is G_a dt kg

where,

G_a = the mass flow rate of drying air, kg/(min.m^2)

Let $T_a = T_a(z,t)$, $T_g = T_g(z,t)$ and $H = H(z,t)$

where,

z = the depth in bed from inlet, m

t = the time, min.

Specific heat of air at humidity H is $c_a + c_{wv}$ H kJ/(kg°C)

where,

c_{wv} = the specific heat of water vapor, kJ/(kg°C)

Specific heat of rough rice grain at moisture content M is $c_g + c_{lw}$ M kJ/(kg°C)

where,

c_g = the specific heat of rough rice grain, kJ/(kg°C)

c_{lw} = the specific heat of liquid water, kJ(kg°C).

Drying Rate Equation

The moisture content of a thin layer of rough rice is expressed by an appropriate thin layer equation. The modified empirical "thin-layer" equation. that describes the drying of several grains well, including rough rice, is

$$\frac{M - M_e}{Mi - M_e} = e^{(-kt^u)} \quad (4)$$

where,

u = the constant

k = the drying constant, l/time

M_e = the equilibrium moisture content of rough rice, decimal (d.b.)

M_t = the initial moisture content of rough rice, decimal (d.b.)

The Eqn. (4) is referred to as the Page equation. (Page, 1949) and in finite difference form can be expressed as

$$\Delta M = -(M - M_e)ku(t_{eq} + \Delta t)^{u-1}\Delta t \quad (5)$$

where,

$$t_{eq} = [-\ln(\frac{M - M_e}{M_i - M_e}) / k]^{l/u}$$

= the equivalent time, min.

ΔM = the change in moisture content, decimal (d.b.).

Mass (Moisture) Balance Equation

For a moisture balance in the elment bed (z, z + dz) over a interval dt: the change in moisture in air across the element of rough rice grain bed is equal to the difference between the moisture leaving the grain and the change in moisture in air within element. We can write

$$G_a dt[H(z+dz,t) - H(z,t)] = -\frac{\partial M}{\partial t}\partial t p_g \partial z - \epsilon p_a dz[H(z,t+dt) - H(z,t)] \quad (6)$$

Applying the Taylor series expansion and neglecting the higher order terms, Eqn. (6) becomes

$$G_a\frac{\partial H}{\partial z} = -p_g\frac{\partial M}{\partial t} - p_a\epsilon\frac{\partial H}{\partial t} \quad (7)$$

Form Eqn. (7), the change in air humidity, in finite difference form, can be written as

$$\Delta H = \frac{(-p_g/G_a)(\Delta M/\Delta t)}{(1/\Delta z) + (\epsilon p_a)/(G_a \Delta t)} \quad (8)$$

Heat Balance (Energy Balance of Air) Equation

The change in enthalpy of air across the element bed is equal to the change in enthalpy of rough rice within the element. We can write

$$G_a[c_a + c_{wv}H(z + dz) \times T_a(z + dz) + L_v H(z + dz)] - G_a[c_a + c_{wv}H(z)Ta(z) + L_v H(z)] = p_g[c_g + c_{lw}M(t + dt) \times T_g(t + dt)]dz - p_g[c_g + c_{lw}M(t)T_g(t)]dz \quad (9)$$

where,

L_v = the latent heat of vaporization of free water, kJ/kg

Applying the Taylor series expansion and neglecting the higher order terms and rearranging it, the air temperature change in finite difference form is

$$\Delta T_a = [\frac{p_g \Delta z \Delta M}{G_a \Delta t}\{S(c_{wv}T_a + L_v) - c_{lw}T_g\} - \frac{p_g \Delta z \Delta T_g}{G_a \Delta t}\{c_g + c_{lw}(M + \Delta M)\}] / \{c_a + c_{wv}(H - \frac{p_g S \Delta M \Delta z}{G_a \Delta t})\} \quad (10)$$

where,

$$S = \frac{1}{(1/\Delta z) + (\epsilon p_a) / (G_a \Delta t)}$$

Let

$$K = \frac{p_g \Delta z}{G_a \Delta t},$$

$E = c_a + c_{wv}(H - K \times S \times \Delta M)$,

$F = Sc_{wv}T_a + SL_v - c_{lw}T_g$,

$G = c_g + c_{lw}(M + \Delta M)$

Thus, from Eqn. (10), we can write

$$\Delta T_a = \frac{K}{E}(F \times \Delta M - \Delta Tg \times G) \quad (11)$$

where, S, K, E, F and G are operators.

Heat Transfer Rate Equation (Energy Balance of Grain)

The heat energy transferred by convection from the air to the rough rice grain in the bed element is equal to the sum of enthalpies required for heating the rough rice grain kernels, for evaporating water from the rough rice kernels, and for heating the water vapor evaporated from the rough rice kernels. We can write

$$h_{cv}(T_a - T_g)dt\,dz = p_g[\{c_g + c_{lw}M(t + dt)\}T_g(t + dt)dz] - p_g[\{c_g + c_{lw}M(t)\}T_g(t)dz] + \{p_g(L_g + c_{wv}T_g)(-\frac{\partial M}{\partial t})\}dz\,dt \quad (12)$$

where,

h_{cv} = the volumetric heat transfer co-efficient, kJ/(m^3 min. °C)

L_g = the latent heat of vaporization of moisture from grain, kJ/kg.

Applying the Taylor series expansion, rearranging it and neglecting the higher order terms, the rough rice grain temperature change in finite difference form is

$$\Delta T_g = [A + p_g(\Delta M/\Delta t)\{(2Y)/(h_{cv}) + (\Delta z F)/(G_a E)\}] / [1 + (p_g/\Delta t)\{(2B)/(h_{cv})$$

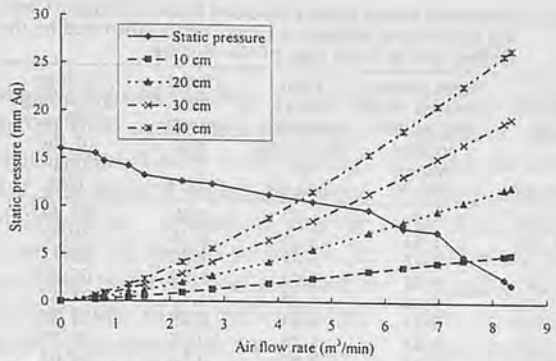

Fig. 3 Measured static pressure versus airflow curve and the system curves of the fan for 10, 20, 30 and 40 cm depths of rough rice considered in the proposed dryer at 2 500 rpm of the engine.

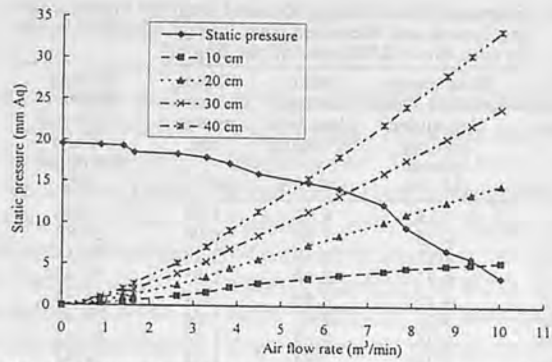

Fig. 4 Measured static pressure versus airflow curve and the system curves of the fan for 10, 20, 30 and 40 cm depths of rough rice considered in the proposed dryer at 3 000 rpm of the engine.

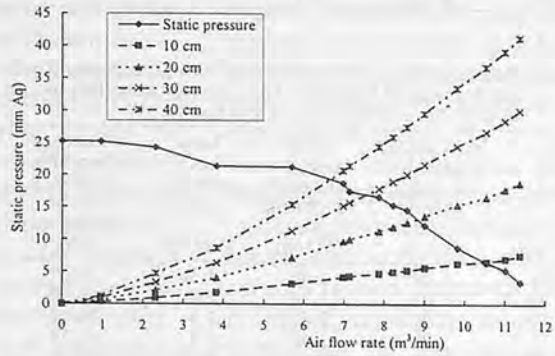

Fig. 5 Measured static pressure versus air flow curve and the system curves of the fan for 10, 20, 30 and 40 cm depths of rough rice considered in the proposed dryer at 3 500 rpm of the engine.

$$+ (\Delta M/h_{cv})(c_{lw} - c_{wv}) + (\Delta z/G_a E)(B + c_{lw}\Delta M)\}] \quad (13)$$

where,
$Y = c_{wv}T_g + L_g - c_{lw}T_g$
$A = 2(T_a - T_g)$
$B = c_g + c_{lw}M;$
Y, A and B are operators.

Method of Solution

Considering the drying of a thin layer of rough rice: if it is sufficiently thin, the properties of the rough rice can be regarded as constant within the layer. Considering also a time interval which is sufficiently short for the properties of the air to be constant with respect to time at the inlet to and at the outlet from the layer.

Drying constant k, and equilibrium moisture content, M_e are determined from the inlet air temperature and relative humidity. The change in moisture content, over the time interval Δt, in the first layer is determined using the thin layer drying Eqn. (5). The change in air humidity ratio of the air after it has been passing through the layer for a time Δt is calculated from Eqn. (8). The change in the temperatures of the drying air and of the rough rice are calculated from Eqn. (11) and Eqn. (13), respectively. The process is repeated layer by layer until the top of the bed is reached.

If the relative humidity higher than the equilibrium value corresponding to the moisture content and temperature of the rough rice is reached, no further drying is allowed to be calculated, but the heat transfer and heat balance equations continue to operate. If the air reaches a high relative humidity and then passes to further layers at lower temperature, its relative humidity will rise as it is cooled at constant absolute humidity. When rthe relative humidity exceeds 99%, the condensation routine deposits back the moisture from the over-saturated air. Air and grain temperatures are adjusted for this condensation so that the upper layers of the deep bed can gain water in the earlier stages of drying. The same procedure is repeated throughout the drying period. The calculation can be stopped when any desired condition, for instance average grain bed moisture content or total drying time, is reached. The model was programmed in BASIC suitable for microcomputers.

Result and Discussions

Performance of the Engine-fan Combination

The static pressure versus airflow curve for the experimental backward curve centrifugal fan at 2 500, 3 000 and 3 500 rpm of the engine are shown in **Figs. 3**, **4** and **5**, respectively. Also, the static pressure versus airflow curve and the system curves by considering the 10, 20, 30 and 40 cm depth of grain beds in the proposed dryer, respectively, are plotted on the same graph as shown in **Figs. 3-5**. Each system curve includes the losses in the straight duct, 90 degree elbow, gradual expansion of the plenum chamber and depth of rough rice bed considered. All these losses are calculated according to the procedure described by

Table 1. Estimated Waste Energy Released from the Engine Cooling System and Measured Heat Energy Absorbed by the Drying Air at 2 500 rpm of the Engine

Air flow rate (m³/min)	Heat energy released from the engine cooling system (MJ/h)	Heat energy absorbed by drying air (MJ/h)	Total heat energy loss (MJ/h)	Energy absorbing efficiency of the drying air (%)
0.64	4.25	1.02	3.23	24.00
0.84	4.25	1.27	2.98	29.88
1.29	4.25	1.97	2.28	46.35
1.57	4.25	2.40	1.85	56.47
2.05	4.25	2.48	1.77	58.35
2.88	4.25	2.61	1.64	61.41
3.96	4.25	2.82	1.43	66.35
4.81	4.25	2.84	1.41	66.82
5.89	4.25	3.24	1.01	76.24
6.55	4.25	3.42	0.83	80.47
7.21	4.25	3.23	1.02	76.00
7.71	4.25	3.24	1.01	76.24
8.48	4.25	2.99	1.26	70.35
8.63	4.25	2.92	1.33	68.71

Table 3. Estimated Waste Energy Released from the Engine Cooling System and Measured Heat Energy Absorbed by the Drying Air at 3 500 rpm of the Engine

Air flow rate (m³/min)	Heat energy released from the engine cooling system (MJ/h)	Heat energy absorbed by drying air (MJ/h)	Total heat energy loss (MJ/h)	Energy absorbing efficiency of the drying air (%)
1.00	5.41	1.72	3.69	31.79
1.29	5.41	2.16	3.25	39.93
2.57	5.41	3.61	1.80	66.73
3.96	5.41	3.52	1.89	65.06
5.00	5.41	4.24	1.17	78.37
5.89	5.41	4.43	0.98	81.89
7.38	5.41	4.76	0.65	87.99
8.51	5.41	4.85	0.56	89.65
8.87	5.41	4.81	0.60	88.91
9.32	5.41	4.80	0.61	88.72
10.17	5.41	4.76	0.65	87.99
10.95	5.41	4.60	0.81	85.03
11.39	5.41	4.48	0.93	82.81
11.77	5.41	4.39	1.02	81.15

Table 2. Estimated Waste Energy Released from the Engine Cooling System and Measured Heat Energy Absorbed by the Drying Air at 3 000 rpm of the Engine

Air flow rate (m³/min)	Heat energy released from the engine cooling system (MJ/h)	Heat energy absorbed by drying air (MJ/h)	Total heat energy loss (MJ/h)	Energy absorbing efficiency of the drying air (%)
0.91	5.02	1.66	3.36	33.07
1.44	5.02	2.54	2.48	50.60
1.57	5.02	2.61	2.41	51.99
1.71	5.02	2.65	2.37	52.79
2.73	5.02	3.02	2.00	60.16
3.40	5.02	3.69	1.33	73.51
3.96	5.02	4.03	0.99	80.28
4.63	5.02	4.22	0.90	82.07
5.82	5.02	4.64	0.38	92.43
6.55	5.02	4.65	0.37	92.63
7.61	5.02	4.64	0.38	92.43
8.13	5.02	4.63	0.39	92.23
9.09	5.02	4.56	0.46	90.48
9.67	5.02	4.46	0.56	88.84
10.35	5.02	4.14	0.88	82.47

Brooker et al (1992) in their book. A curve plotted as pressure drop versus air flow is called a system curve. The static pressure and the air flow at the point of intersection are those at which the system will operate. From the stability point of view, it seems to be more appropriate to operate the engine at 3 000 rpm at which the fan rpm is 1 500. Due to the higher torque at 2 500 rpm of the engine, the whole system becomes more noisy and vibration is also higher at this speed. The whole system also seems to be unstable at 3 500 rpm of the engine. From **Tables 1-3**, it can be observed that the energy absorbing capacity of the drying air from the engine waste is highest at 3 000 rpm of the engine for the measured drying air flow rates by considering 10, 20, 30 and 40 cm depth of rough rice beds, respectively. On the basis of these it is concluded that 3 000 rpm of the engine is best rpm for this fan and engine combination.

Possible air flow rates for 10, 20, 30 and 40 cm depth of grain beds in the proposed dryer are presented in **Figs. 3-5** for 2 500, 3 000 and 3 500 rpm of the engine, respectively. From **Fig. 4**, it can be observed that the air flow rates are 5.60, 6.55, 7.70 and 9.45 m³/min for the assumed 40, 30, 20 and 10 cm depth of rough rice grain beds in the dryer, respectively, at 3 000 rpm of the engine. **Figs. 6** and **7** show the measured drying air temperatures and relative humidities of the drying air at different drying air flow rates and at

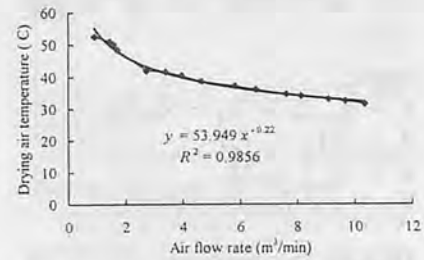

Fig. 6 Variations of drying air temperature with air flow rates at 3 000 rpm of the engine (average ambient temperature is 25.6°C).

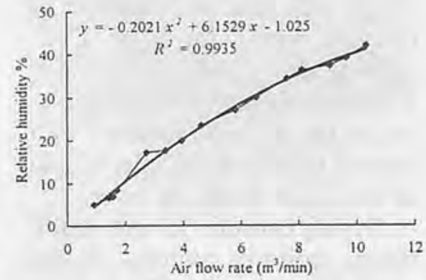

Fig. 7 Variations of drying air relative humidities with air flow rates at 3 000 rpm of the engine (averae ambient air relative humidity is 78.0%).

3 000 rpm of the engine, respectively. The drying air temperature and the air flow rate best follow the power regression equation with $R^2 = 0.99$ (**Fig. 6**) at an average ambient temperature 25.6°C. The drying air relative humidity and the air flow rate follow the second degree polynomial regression equation with $R^2 = 0.99$ (**Fig. 7**) at an average ambient air relative humidity 78%. Relative humidity is calculated from the measured dry-bulb and wet-bulb temperatures of the drying air. It can be

observed from the **Fig. 6** that as the air flow rate is increasing drying air temperature is decreasing. But the reverse effect has been observed in the case of relative humidity, as shown in **Fig. 7**. It has been observed that engine fin temperature varies with the ambient temperature at the same air flow rate and at the same engine speed. Very little variation has been observed between the temperatures measured at the end (A) of straight duct and at the entrance (B), (C) and (D) of the proposed dryer, respectively (**Fig. 1**). So the average of the temperatures measured at (A) for a particular air flow rate is considered as the drying air temperature in the deep drying model and in other estimations.

The estimated energy released as wasted heat from the engine cooling system, measured heat energy absorbed by the drying air, energy losses from the system and waste energy absorbing efficiency of the drying air at different measured air flow rates and at different speed of the engine are presented in **Tables 1-3**. Wasted heat energy released from the engine cooling system is calculated from Eqn. (2). Heat energy absorbed by the drying air is calculated from Eqn. (3). Heat energy losses, which are due to various causes, are estimated to be the difference between the estimated heat energy released by the engine cooling system at a given rpm and the heat energy, estimated from the measured air flow rate and drying temperature rise, absorbed by the drying air. The average energy losses are for comparison purposes only and do not imply that the loss rate will remain constant over the period of the drying experiment. Since instrument is not available to directly measure the system energy losses, one can only speculate about the source of these losses. Although the fan housing, duct system and plenum chamber are properly covered with insulating material, it is probably safe to assume that a portion of the heat will be absorbed by the air in heating the fan housing, duct and plenum chamber. It is also possible that some heat energy is lost by conduction and radiation in the system.

As much as possible the wasted heat from the engine must be absorbed by the drying air as it is drawn over the engine and to the fan. This heat is sufficient to increase the drying air temperature some 5 to 20°C depending on the air flow rate (**Fig. 6**). It can be observed from **Table 2** that about 93% of the wasted heat from the engine is absorbed by drying air at 3 000 rpm of the engine. Efficiency seems to be very high because part of the exhaust energy is also absorbed by the drying air when the fan is in operation. This is not directly due to harvesting of exhaust gasses, rather exhaust wasted heat energy is abosrbed and carried by the drying air from the surface of the exhaust passage. So it will not be harmful to the grain, rather it will give an extra benefit to the system. The fan also helps in the proper cooling of the engine. It has been observed that at the same ambient temperature, and engine speed, the engine fin temperature becomes 30°C higher than the engine running with a fan coupled at the camshaft. This will help to increase the operating life of the engine.

In principle a low temperature dryer by using the moisture absorbing capacity slightly over the ambient air may require less energy than the latent heat of vaporization of water (~2.5 MJ/kg) to remove moisture from the grain. Tests were conducted by the NIAE (1956) on radial bin, floor ventilated bin and on-floor low temperature dryers. For the radial bin dryer, monitored for a single season's drying, the mean energy required was 3.52 MJ/kg water removed, at an average relative humidity of 79%. Single tests, with results adjusted to an average relative humidity of 80%, on the floor ventilated bin and on-floor dryers gave an energy consumption of 3.65 and 3.72 MJ/kg of water removed, respectively. Using these results as a guide, the amount of wasted heat from the engine absorbed by the air is enough to vaporize 1.2 kg to 1.3 kg of water per hour from wet grain on a day having a relative humidity of 80%.

Simulated Moisture Content

The main (basic) input data for the simulation model are summarized in **Table 4**. Standard tabulated values are used for other input parameters in the partial DE model regarding the drying air, the rough rice and the water properties. On the basis of air flow characteristics as mentioned in **Table 4**, simulation is performed by considering 10, 20, 30 and 40 cm deep grain beds in the proposed dryer. The principal simulated results are shown in **Figs. 8-12** which plot moisture content % (w.b.) against drying time. Moisture content of all rice is considered 22% (w.b.) at the beginning of drying and is dried to an overall average of approximately 14.5% m.c. (w.b.). Bulk density of the medium-grain rough rice at 22% m.c. (w.b.) is considered 700 kg/m^3 (Brooker et al., 1992). On the basis of this bulk density, the initial weight of moist grain in 10, 20, 30 and 40 cm deep grain beds in the proposed dryer are calculated, as shown in **Table 5**.

Simulations show that the time required to bring the average approximate moisture content 14.5% (w.b.) from the initial 22% m.c. (w.b.) are 8, 9, 11 and 14 h

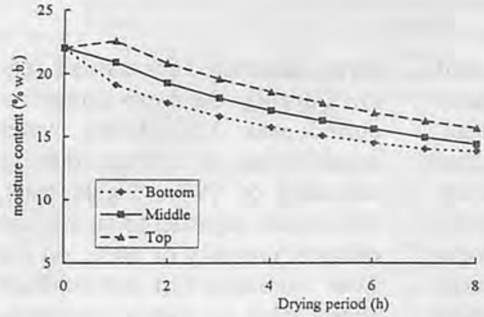

Fig. 8 Simulated moisture content at bottom, middle and top layers of a grain bed of depth 10 cm with drying periods (average m.c. at the end of drying is 14.48% w.b.).

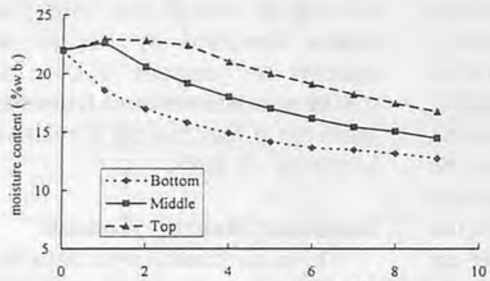

Fig. 9 Simulated moisture content at bottom, middle and top layers of a grain bed of depth 20 cm with drying periods (average m.c. at the end of drying is 14.46% w.b.).

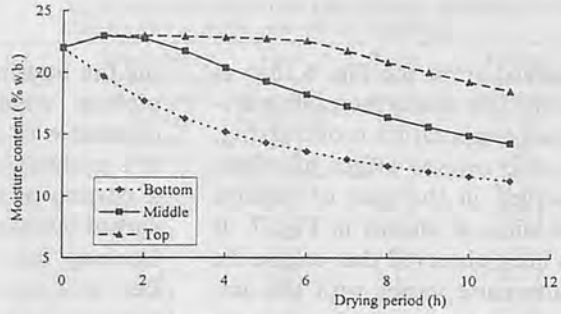

Fig. 10 Simulated moisture content at bottom, middle and top layers of a grain bed of depth 30 cm with drying periods (average m.c. at the end of drying is 14.56% w.b.).

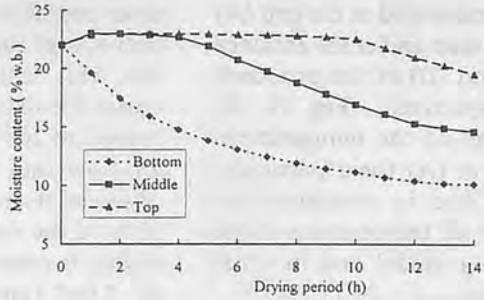

Fig. 11 Simulated moisture content at bottom, middle and top layers of a grain bed of depth 40 cm with drying periods (average m.c. at the end of drying is 14.5% w.b.).

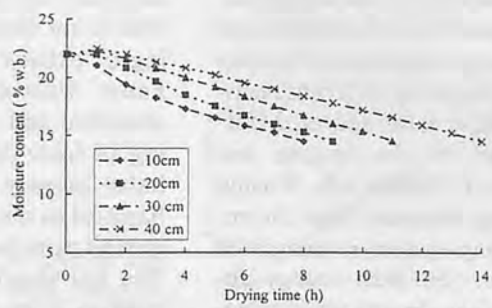

Fig. 12 Average simulated moisture content with drying periods in 10, 20, 30 and 40 cm depth of grain beds considered in the dryer in this study (average m.c. at the end of drying is approximately 14.5% w.b.).

while the depth of grain bed are considered 10, 20, 30 and 40 cm, respectively. The moisture gradient between the top layer and bottom layer are found to be 1.8, 4.0, 7.25, and 9.41% (w.b.) of 10, 20, 30 and 40 cm depth of grain beds, respectively, at the end of drying (**Table 5**). These results show that the moisture gradient between the top layer and bottom layer is a problem in higher depth of grain bed. As the depth of grain bed is increased the moisture gradient also increased. The moisture gradient is quite high in 40 cm depth of grain bed which is not desirable. Undisturbed, continuous air flow, fixed bed drying has an inherent problem of over-drying where the drying air is introduced, particularly with the higher air temperatures that tend to lower the moisture content below the desired 13% w.b. as reported by Angladette (1963). But in an engine-waste heated dryer, drying air temperatures are not so high. It is 32.3-37.3°C depending upon the depth of grain bed considered in the proposed dryer at an average ambient temperature of 25.6°C.

Though the simulation results show that over-drying takes place at the bottom layer and the grain remains under-drying at the top layer, when the depth of the grain bed is considered 40 cm, in actual drying it may not be so because of low drying air temperature (37.3°C). Fourteen hours are required in one batch drying, when the depth of the grain bed is considered 40 cm. So if the drying starts in the morning, the drying temperature will be low at the beginning because of low ambient temperature in the morning. As the ambient temperature gradually increases till noon, the drying air temperature will also gradually increase. Again, from afternoon, the ambient temperature will gradually fall, and the drying air temperature will fall, accordingly. This indicates that the drying air temperature and its relative humidity will vary thorughout the drying period while drying a particular depth of grain bed depend-

Table 4. Drying Air Flow Characteristics at 3 000 rpm of the Engine for Different Depth of Grain Beds

Depth of grain bed	Air flow rate	Static pressure	Drying air temperature	Relative humidity of drying air
(cm)	(m^3/min.)	(mmAq)	(°C)	(%)
10	9.45	5.50	32.25	42.60
20	7.70	10.50	34.25	35.65
30	6.55	13.80	35.80	31.00
40	5.60	15.00	37.30	26.90

Ambient temperature 25.6°C, relative humidity 78%.

Table 5. Effect of Depth of Grain Bed on Energy Requirement for Drying and Moisture Gradient at 3 000 rpm of the Engine

Depth of grain bed	Initial weight of grain	Average initial m.c.	Average final m.c.	Bottom layer m.c.	Top layer m.c.	Moisture gradient	Total drying period	Energy required
(cm)	(kg)	(% w.b.)	(%W.b.)	(% w.b.)	(% w.b.)	(% w.b.)	(h)	(MJ/h)
10	56.70	22.0	14.50	13.93	15.73	1.80	8	7.10
20	113.40	22.0	14.50	12.70	16.70	4.00	9	5.20
30	170.10	22.0	14.50	11.28	18.53	7.25	11	4.50
40	226.80	22.0	14.50	10.10	19.51	9.41	14	3.13

m.c. = moisture content, w.b. = wet basis.

ing upon the ambient temperature. So in practice, under-drying in the top layer and over-drying in the bottom layer will not be a problem while drying a 40 cm deep of rough rice bed. In simulation it has been found that only up to 5 cm height of grain bed from the bottom layer is subjected to over-drying and up to 5 cm depth from the surface layer remains under-drying at the end of drying, when the depth of the grain bed is considered 40 cm. If the grain is left in the dryer for a few hours after drying is over, it will come in moisture equilibrium with the ambient air. Even, if over-drying happens in the bottom layer, as the simulation shows, then kernel breakage and checks will not be a problem due to long exposure of the bottom layer to low drying air temperature.

The partial differential equation models of deep-bed drying were principally constructed for studying the physics of drying and modeling of high temperature drying (+60°C) and the lack of accuracy of the thin layer drying and equilibrium moisture content equations in the deep bed drying model. These might have been caused by over-drying in the bottom layer, while grain depth is considered 40 cm. On the basis of these, it is concluded that a 40 cm depth of rough rice is optimum for an engine-waste heated flat bed dryer if the drying period remains fixed at 14 hours at an average ambient temperature 25.0°C.

With the measured energy absorbed by the drying air, energy required to remove per kg of water from the grain to bring the final average moisture content to approximately 14.5% (w.b.) is calculated for the 10, 20, 30 and 40 cm depth of grain beds, respectively (**Table 5**). Results show that as the depth of grain bed is increased the energy required to remove per kg of moisture from the moist grain decreased. The maximum energy requirement is 7.1 MJ/kg of water removed from the moist grain while the depth of grain bed is considered 10 cm (**Table 5**). The minimum energy requirement is 3.13 MJ/kg of water removed which is lower than the average energy requirement of a highly efficient mechanical dryer 3.6 MJ/kg (Sharp, 1982) while the depth of grain bed is considered 40 cm (**Table 5**). This indicates that there is little chance to increase the dryer base area with this engine waste heat if the total drying period remains fixed at 14 h. On the basis of the minimum energy requirement (3.13 MJ/kg) and available engine waste for drying, an estimation is made to find the amount of grain that can be dried by the engine waste heat. The wasted heat from an engine of a power range of 1.9-10 PS, with a dryer base area of 0.81-4.5 m^2 and an air flow rate of 5.6-30.8 m^3/min, can dry about 0.2-1.25 metric ton of rough rice from 22% m.c. (w.b.) to 14.5% m.c. (w.b.) in 14 h at an average ambient temperature 25°C and relative humidity 80%.

Conclusion

On the basis of measured available engine waste heat energy that can be directly forced through the grain bed, a theoretical study on engine waste heated dryer performance was made. Results have shown promise for this type of grain drying unit, especially in the major rice growing regions where the same engine which is used for pumping irrigation watter and rice milling purposes can also be used for grain drying. Although an exact prediction of engine-wasted heat stationary deep-bed rough rice dryer performance may be impossible, comparative performance studies of different dryer designs can be made with confidence. These simulations need experimental verification.

The 37°C drying air temperature can be attained under most tropical conditions with waste engine heat, thus no additional capital investment nor operating cost is necessary for the supplemental heating of drying air. Even the provision can be easily made to use the same engine simultaneously for duel purposes like drying and milling or irrigation as the power requirement by the fan is very low compared with the engine power. The main advantage of the engine-wasted heat dryer is that it can be easily fabricated by local technicians using locally available materials and less kernel breakage and stress checks will occur than when rice is improperly dried in the sun or too rapidly dried with highly heated air. The use of this static flat-bed dryer in rural areas where electricity is not available, should then encourage the harvesting of improved rice varieties with field moisture contents as high as 23% m.c. (w.b.) to minimize harvest shatter losses.

(Continued on page 48)

Testing of Bullock-operated Variable Width Sugarcane Ridger

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Abstract

A bullock-operated variable width sugarcane ridger was developed and tested. Field test was carried out on three different soils (light, medium, heavy). The comparative performance of the developed ridger and traditional ridger was studied in terms of draft requirement. The width obtained on developed ridger varied from 60 to 100 cm; whereas on traditional ridger, the width obtained was from 65 to 80 cm. The draft requirement (60 to 100 cm) varied from 80 to 123 kgf for the developed ridger whereas 99 to 141 kgf for traditional ridger. The average percent decreased draft in newly developed ridger with respect to the traditional ridger was observed to be about 17% for all widths in all types of soil.

Introduction

In India land holding is small in size. The cultivation of such holding is based mainly on small implements drawn by small power. Therefore, Indian farmers need small machineries preferably animal-drawn which are within their economic range and will serve their purposes. Animals are the main source of draught power on most of the small farm holding in India. It is, therefore, essential that India's small farmer's must acquire modern bullock-drawn implements.

Sugarcane is an important cash crop in India. Sugarcane crops require a well prepared, well structured and well aerated soil profile to a depth of about 25 cm without hardpan in the subsoil for higher production. Presently, farmers use different types of implements for land preparation in sugarcane cultivation (mainly for making ridges and furrows). Therefore, it is desirable to use ridgers.

The ridger is an implement used to form ridges and furrows for sown- or row-planted crops in well tilled soil. It cuts and throws soil in two opposite directions simultaneously for forming ridges. Most of the farmers use bund former, mouldboard ploughs and few use blade harrow to form the ridges. The use of these implements require more draft as well as their use is time consuming and uneconomical. Sugarcane crop is planted at 60 to 90 cm row-to-row distance depending upon the type of soil and variety of crop. By considering all, these factors the ridger was designed. Thus the project was undertaken with a view to testing the developed variable width sugarcane ridger and compare it with the traditional ridger.

Objectives

To evaluate the performance of a newly developed ridger and the traditional ridger in terms of draft requirement.

i. On three different soils, viz., a. light, b. medium and c. heavy.
ii. For three different spacings of ridges and furrows i.e., 65, 75 and 95 cm.

Methodology

By considering drawbacks in the traditional ridger, the main modifications done to the new ridger is an arrangement for varying the width of operation to suit different sugarcane varieties. The different spacings (60, 75 and 95 cm) of ridges and furrows or achieved by providing a slot on the mouldboard and with the help of nuts and bolts **(Fig. 1)**. Thus the width of formation was varied by sliding the wingboards up and down whereas, in the traditional ridger, the width is varied by expansion of wingboards having rod arrangement behind the wingboards **(Fig. 2)**.

The comparative performance of the newly developed ridger and

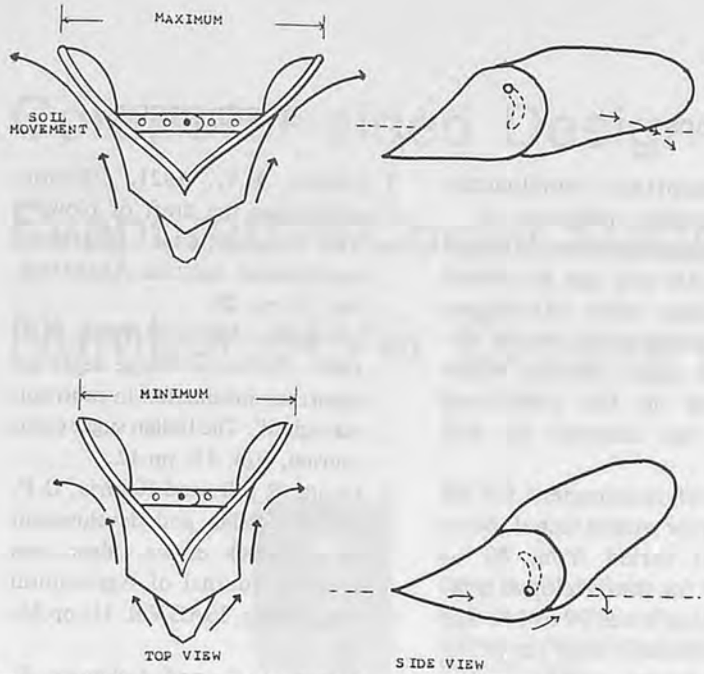

Fig. 1 Top and side view of wingboards showing maximum and minimum of width, in the developed ridger.

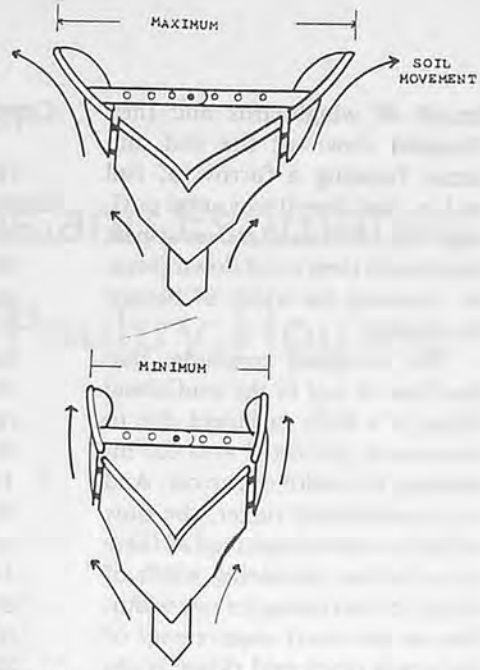

Fig. 2 Top view of wingboards showing maximum and minimum of width, in the traditional ridger.

traditional ridger was studied in terms of draft requirement in the field for three different spacings of ridges and furrows (65, 75 and 95 cm) generally adopted for sugarcane cultivation and on three different types of soil (light, medium and heavy).

Result and Discussion

The average moisture content at the time of testing was 6.07% in light soil, 6.2% in medium soil, ad 6.07% in heavy soil.

The percentage of soil constituents for light, medium and heavy soils are presented in **Table 1**.

Measurement of Draft

The draft requirement varied from 79 to 123 kgf for the developed ridger in all three types of soils, i.e., light, medium and heavy. The draft required for traditional ridger was between 99 and 141 kgf in all three types of soil. The draft requirement for all widths in each soil is shown in **Table 2**.

Table 1. Texture of Soil

Soil type	Clay (%)	Silt (%)	Sand (%)
Light	19.8	28.2	51.9
Medium	21.2	61.6	17.2
Heavy	43.7	26.3	30

The average percent decreased in newly developed ridger with respect to traditional ridger was about 17% in all the trials for all widths in all types of soil. The draft requirement was contributed by the pattern of soil movement. In the traditional ridger the expansion of wingboards increased the width, hence the total projected area against the soil movement increased so that the total resistance offered by the wingboard to the movement of soil increased. Because of this a small soil turbulence was created at the interaction of wingboards and soil thereby increasing the width.

Whereas in the developed ridger the width of furrow formation was varied by sliding the wingboards up and down, hence the distance between wingboards was slightly changed. The width of furrow formation was mainly changed by the movement of wingboards up and down so that the path (trajectory) of soil movement on the wingboards was increased or decreased thereby changing the furrow width.

When the wingboards were at full down position **(Fig. 1)** the soil movement was throughout the full

Table 2. Draft Requirement for Developed and Traditional Ridger

Soil type	Width obtained (cm)	Traditional ridger: Depth obtained (cm)	Traditional ridger: Draft requirement (kgf)	Developed ridger: Depth obtained (cm)	Developed ridger: Draft requirement (kgf)	Developed ridger: % decrease draft w.r.t. traditional ridger
Light	65	20	99	24	79.2	20
	75	25	118.8	25	98.5	17
	95	—	—	22	98.5	—
Medium	65	22	118.8	22	98.5	17
	75	24	118.8	24	98.5	17
	95	—	—	22	117	—
Heavy	65	21	141.0	21	117	17
	75	24	141.0	24	117	17
	95	—	—	21	123	—

length of wingboards and then dropped down at the end and hence forming a furrow of full width. And then it was at up position (**Fig. 1**) the soil moves at part length and then it fell down thereby reducing the width of furrow formation.

The foregoing concludes that the flow of soil in the traditional ridger is a little turbulent due to increase in projected area for increasing the width of furrow. And in the developed ridger, the flow of soil is a bit streamlined as there is no further increasing width of ridger for increasing furrow width. Due to this draft requirement of the newly developed ridger is observed less in comparison to the traditional ridger.

Test for Field Capacity

The field capacity was between 0.12 and 0.14 ha/h for both ridgers. The field capacity for both ridgers was at par. But it may vary on large scale operation as draft requirement affects the speed of bullocks, i.e., as draft requirement increases the speed of bullocks on average is reduced in mass operation.

Conclusions

The important conclusions drawn from this study are:

1. The spacing between the ridges (60 to 100 cm) can be varied by making width adjustment of the wingboards on the developed ridger whereas width obtained on the traditional ridger was between 65 and 80 cm.
2. The draft requirement for all the furrow widths tested (60 to 100 cm) varied from 80 to 123 kgf for the developed ridger whereas it was 99 to 141 kgf for traditional ridger for (65 to 80 cm) furrow widths.
3. The furrow formed has trapezoidal shape and depth varied from 20 to 26 cm for the 60 to 90 cm spacing for developed ridger and 19 to 24 cm for traditional ridger in all three types of soil, i.e., light, medium and heavy.
4. The percent draft decreased in newly developed ridger w.r.t. traditional ridger is 17% on an average for all widths in all types of soil.

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(Continued from page 45)

Simulation of Engine-wasted Heat Stationary Deep-bed Rough Rice Dryer

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Computer-aided Design for Evaporative Cooler Systems and Estimating Number of Air Coolers in Poultry Houses

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Abstract

High levels of temperature, humidity and radiation adversely effect the productivity of poultry. The drop in production is very large when temperature goes beyond 25-30°C. Birds increase evaporative heat loss as ambient temperature increases to levels at which they cannot readily dissipate the metabolic body heat by the sensible transfer means of convection, conduction and radiation. The sensible heat transfer depends on temperature difference. Latent heat transfer depends on vapor pressure difference and air movement.

Hot weather environmental control for livestock and poultry housing has not received attention as has cold weather housing. During hot weather there are fewer alternatives. The workable means for coping with hot weather conditions are evaporative cooling. Evaporative cooling is successful under low humidity climatic conditions. Air movement is a means of maximizing sensible heat loss as long as the ambient air temperature is less than the animal body temperature.

Science and technology are basic in dealing with environmental factors as the real need for feed component has been studied. The poultry housing and ventilation equipment are the real key to reaching maximum poultry production.

Thus correct and suitable design which deals with the weather conditions, environmental factors, building location, water quality, insulation, and number of birds could be the best design of evaporative system design which will adequately cover the needs of poultry buildings to attain economical production.

Literature Review

Stanley E. Curtis (1980) presented in his book "Environmental management in animal agriculture" a simple way to calculate minimum ventilation rate for summer. He suggested the use of evaporative cooling as a solution to lower inside temperature when regular ventilation does not change temperature.

M. Esmay (1986) suggested several alternatives to drop inside temperature in hot climate. One such important way is evaporative cooling when outside temperature is higher than inside temperature.

Munters, the designer company for evaporative cooling systems suggested several ways in design and selection of system component.

Objectives

1. Construct simulation model to design an evaporative cooling system (pads) for poultry housing.
2. Determine the component of the evaporative system so it could be ordered from any company.
3. Use of simulation model to evaluate several alternatives in evaporative design.
4. Estimate of number of evaporative coolers if pad system is not used.

Methods

To calculate minimum ventilation rate for summer (MVRS), we must know all the physical factors which effect this variable. And to make these calculation accurate it is necessary to know the declination angle s which is usually presented in tables. Howell et al. (1982) suggested the following approximate equation —

$\sigma = 23.45 \sin\{360[284 + n/365]\}$

where
n = the day of the year with highest temperature

The solar altitude α at noon is

$$\alpha_{noon} = 90 - (L - \sigma)$$

where
L = north latitude of location in degrees

To calculate the incident angle θ for both sides of the sloped roof (gable) when the direction of the building is East-West. For both sides north and south —

$$\Theta_n = 90 + \beta - \alpha$$
$$\Theta_s = 90 - \beta - \alpha$$

where
Θ_n = incident angle for the north slope
Θ_s = incident angle for south slope
β = roof slope

To estimate the incident radiation K we should know the cosine of incident angle θ and the direct normal radiation.

$$K = \cos \Theta I_d$$

where
I_d = the incident solar radiation W/m^2

The sol-air temperature t_e may by calculated for the north and south roofs and walls from this equation.

$$t_e = t_o + (\alpha I/f_c)$$

where
t_o = outside temperature °C
α = solar absorptivity of the outside roof surface
I = the combined incident solar radiation W/m^2
f_c = the convective film coefficient of outside roof surface $W/m^2.C$

To calculate heat load q_{sr} (gain) in W for the north and south sloop and walls.

$$q_{sr} = A * (t_e - MDT)/R$$

and for other walls which effected by outside temperature only is calculated by

$$q_{to} = A * (t_o - MDT)/R$$

where
A = area m^2
R = overall resistance $m^2.C/W$
MDT = maximum design temperature °C

Sensible heat from birds q_{sb} is calculated based on M. Esmay by 30% sensible heat (when inside temperature is above 29°C) and 70% latent heat, so the equation will be —

$$q_{sb} = h_s * W * N_b * 0.3$$

where
h_s = sensible heat produced by birds in W/kg.bird
for broilers = 4
for hens = 3.2
W = bird's weight kg
N_b = number of birds

To calculate heat gain from manure and water vapor q_{hw} produced by the birds in W, assuming 50% of the water vapor will stay in the system and affect it and the rest will leave the system with the ventilation.

$$q_{hw} = MP * W * N_b * 0.7 * 0.5 * 2407$$

where
MP = moisture produced by the birds $kg_{H20}/kg.h$

The total heat gain q_{tot} affecting the building will be sum of all heat gains. Building heat gain q_{bl}, heat produced by birds q_{sb}, and heat gain from manure and water vapor q_{hw}.

$$q_{bl} = q_{sr} + q_{to}$$
$$q_{tot} = q_{bl} + q_{sb} + q_{hw}$$

To calculate MVRS we should know the temperature rise limit TRL. Knowledge of the animal's thermal requirements makes it possible to choose a maximum point above which temperature inside the house - the animal's microenvironmental temperature - should not be allowed to rise. Of course, as this point is chosen it should be remembered that maximum temperature would occur for only a few hours each day, and that if acclimatized many animals can withstand some stress each afternoon without suffering a loss of production. The maximum temperature is higher than the inside design temperature (SDT) used in cold conditions. It is most economical to operate at the lower end of this range during cold weather, and at the upper end during hot weather. In this case —
MDT = 29°C
SDT = 26°C

$$TRL = MDT - SDT$$

where
SDT = Summer design temperature

Stanley E. Curtis used a simple equation to calculate MVRS

$$MVRS = (0.827 * q_{tot})/TRL$$

To calculate the number of air exchanges for any building which is a number between 6-60/h.

$$AXR = (MVRS * 60)/V_{ib}$$

where
V_{ib} = building volume m^3

To design an evaporative cooling system we should estimate the main effective variables like water distribution, water flow, water quantity and quality, bleed-off flow and pipe system.

The adequate wetting of the pads is the main element in evaporative cooling process. This factor depends on the pump capacity for the system. To select this pump we should know the total head, suction head, discharge head, elevation head, and friction head.

To calculate pressure P_{sr} in the system in kPa we should know number, size of the pipes and the fittings are their friction.

$$P_{sr} = H_t / 0.10255$$

where

H_t = total head m

To estimate the pump power requirement Mwt in W the following equation could be used —

$$M_{wt} = (H_t * F_t * 9.9467)/\mu_m * \mu_p$$

where

F_t = pump capacity or peak demand l/s

μ_m = motor efficiency %

μ_p = pump efficiency %

To prevent or eliminate salt sediment (accumulation) on the pads because of water evaporation, bleed-off ratio (B/E) must be calculated, which is the ratio of water leaving the system B (Bleed-off) to the water evaporated from the pads E. To calculate that pH value for the water we should know calcium concentration and bicarbonate concentration in ppm.

To calculate dry bulb temperature after the pads, we should identify pad efficiency and air speed. The following equations used to calculate the efficiency for two types of pads.

$$\mu_{pd1} = \mathrm{Exp}(\log(88.07409) - 0.11072 * A_{vi}$$

$$\mu_{pd2} = \mathrm{Exp}(\log(97.47760) - 0.04200 * A_{vi})$$

where

μ_{pd1} = pad efficiency for 100 mm pad depth

μ_{pd2} = pad efficiency for 200 mm pad depth

The expected dry bulb temperature after the pad T_{in} is calculated according to this equation —

$$T_{in} = T_{db} - (T_{db} - T_{wb}) * \mu_{pd}/100$$

where

T_{db} = outside dry bulb temperature °C

T_{wb} = outside wet bulb temperature °C

To determine the amount of evaporated water from the pad E in kg/h the following equation is used —

$$E = MVRS * (W_2 - W_1) * 1.2$$

where

W_1 = humidity ratio for air before entering the pads $kg_{H20}/kg_{d.a}$

W_2 = humidity ratio for air after entering the pads $kg_{H20}/kg_{d.a}$

The pump capacity F_{tl} is the sum of the Wp bleed-off flow B_f, and the minimum water flow W_p for sufficient wetting which is estimated according to Munters by 1 $l/s.m^2$.

$$F_{tl} = B_f + W_p$$

To calculate the number of pads or number of evaporative air coolers (if needed) the following equations were used —

$$P_{da} = MVRS/A_{vi}$$
$$P_{dl} = P_{da}/P_{dh}$$
$$P_{no} = P_{dl}/P_{dw}$$
$$N_{co} = MVRS/V_{cl}$$

where

P_{da} = total pad area m^2

P_{dl} = total pad length m

P_{dh} = pad hight m

P_{dw} = pad width (0.6) m

N_{co} = number of air coolers

V_{cl} = air flow of the evaporative air cooler m^3/min

Results and Discussion

The model was used to calculate all needs for an evaporative cooling system design (pad) for a poultry house near Baghdad (See appendix A for more detail).

The objective of this model was not only getting all the needs to built a system adequate to weather conditions but to be economical and environmentally adequate for birds. All variables were fixed to accept air velocity through the pads which was 1-1.5 m/s.

Two types of pads were used in this run 100 mm and 200 mm.

Figure 1 shows the expected changes with the use of 100 mm pad depth, and air velocity 1 m/s. The expected dry bulb temperature after the pad was 26°C which is equal to summer design temperature (SDT). This temperature is not the final inside temperature in the house, which will be higher by 1.5°C the average difference inside the house. Then the final temperature will be 27.5°C. This temperature falls in the upper critical temperature UCT. Pad numbers calculated from the model for this situation was 52 with length of 1 m, the ventilation rate was 31.58 m^3/s, relative humidity 63%. **Table 1** shows results for several runs. The best air velocity was 1.5 m/s. Selecting air velocity less than 1.5 m/s means same inside temperature and more pads. The number of pads with air velocity of 1.5 was 35 pads, which is from economic side better and without effecting the efficiency of the system.

The relative humidity was lower than 58% which is more desirable: to get rid off the moisture produced by the birds and the manure and other sources. The ventilation rate is a little higher than the previous one 32.35 $^3/s$.

Different types of pad used in the second run 200 mm depth. **Fig. 2** shows the results. The air velocity was first used 1 m/s. The expected inside temperature after the pad was 23°C, the final

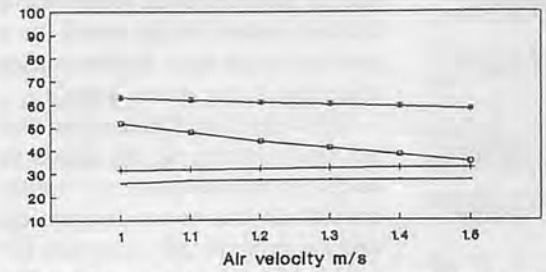

Fig. 1 Effect of air velocity on 100 mm pad depth pad.

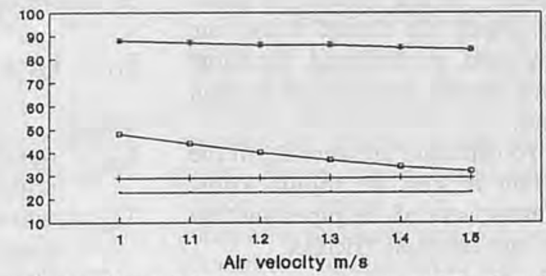

Fig. 2 Effect of air velocity on 200 mm pad depth pad.

Appendix Table

Building Information

Length	70.00 m
Width	11.00 m
Height	3.00 m
Latitude	33.14 deg
Roof angle	18.00 deg
R-value for walls	1.25 m^2.C/W
R-value for ceiling	1.20 m^2.C/W
Orientation East — West	

Environmental Information

Incident Solar Radiation	1 000.0 W/m^2
Outside temperature	45.0 °C
Outside relative hum.	10.0 %
Expected inside temp.	23.1 °C
Expected inside rel. hum	84.2 %
Summer design temp.	26.0 °C
Maximum design temp.	29.0 °C
Summer Ventilation rate	1 758.8 m^3 /min

Management Information

Number of birds	7 000
Kind of birds	Broilers

Evaporative Cooling System Specification Design

Pad length (total) (L)	19.00 m
Pad width (W)	0.60 m
Pad height (H)	1.00 m
Pad depth	0.20 m
Pad efficiency	91.77 %
Number of pads	32.00
Air velocity	1.50 m/s
No. of gutter supports	7.00
Bleed-off ratio (B/E)	1.65
Bleed-off flow	31.55 L/min
Wetting flow rate	234.51 L/min
Pump capacity-total flow	266.05 L/min
Pump power	180.18 W
Water calcium concen.	200.00 ppm
Water carbonate concen.	150.00 ppm
Water pH value	7.40
Suction head	0.91 m
Discharge head	2.45 m
Total head	3.36 m
No. of elbow 90 degree	4.00
No. of tee 90 degree	1.00
No. of coupling used	2.00
No. of angle valves	0.00
No. of globe valves	0.00
No. of gate valves	2.00
Minimum tank volume	0.98 m^3
Pipe size	38.10 mm
System pressure	32.91 kpa
Air exchange rate	45.00 l/h
4 500 cfm air cooler	13.00
8 000 cfm air cooler	7.00
10 000 cfm air cooler	6.00

Table 1. Results of Changes for Pad Depth 100 and 200 mm

Run No.	Vent. m^3/s	Inside Temp. °C	Rel. Hum. %	Air Vel. m/s	No. of pads
1	29.00	23	88	1.0	48
2	29.07	23	87	1.0	44
3	29.13	23	86	1.2	40
4	29.18	23	86	1.3	37
5	29.25	23	85	1.4	34
6	29.31	23	84	1.5	32
			pad depth 100 mm		
1	31.58	26	63	1.0	52
2	31.57	27	62	1.1	48
3	31.90	27	61	1.2	44
4	32.05	27	60	1.3	41
5	32.20	27	59	1.4	38
6	32.35	27	58	1.5	35

Note: Pad length was 1 m and roof slope was 18 degrees.

temperature with the average difference of 1.5°C was 24.5°C which still in the optimal range of production for birds. But there was an increase in inside relative humidity of 88% and this could be accepted for a short time.

When an air velocity was 1.5 m/s we can see the inside condition were changed for the better. The inside dry bulb temperature was the same 23°C. The relative humidity dropped to 84%, the pad number also decreased from 48 to 32, this means a savings in initial cost of the system. There was very small increase in ventilation rate from 29 to 29.32 m^3/s. The expected quantity of water (pump capacity) was 379.83 l/min when the air velocity was 1 m/s, then this amount decreased to 266 l/min when the air velocity increased to 1.5 m/s.

Comparing **Figs. 1** and **2** we can recognize the logical behavior of the system when affected by variable air velocity. This means that the model is a trough representative of the system and could be used to design any evaporative cooling system.

From the above we can deduce that the best selection is using pad with depth of 200 mm, with length of 1 m. Then the number of pads will be 32. The model also used to determine if there is any changes in the system if the fixed variable will be changed like roof angle. The roof angle was changed from 12 to 30 degree.

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Optimum Land Size Determination for Sugar Beet Mechanization in Turkiye

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Abstract

The purpose of this research was to determine the optimum land parcel size in order to maximize the use of total task time and minimize total area loss. It was found that there were a lot of factors affecting optimum parcel size, such as technical specifications of the farm machineries, land irrigation status, soil type, and farm techniques applied to farm mechanization. A computer program written in FORTRAN, was developed to make a precise calculation about area loss as well as the total time for completing farm operation from the soil tillage and seed bed preparation time to harvesting time of the sugar beet mechanization in Turkiye. Measured data about farm machineries and the soil were used in the computer model. When the area loss and the total task time were numerically equal, the total area of the land was suggested as an optimum parcel size. For sugar beet mechanization, the optimum parcel size calculated between 1.6-2.4 ha depending on soil type, machine parameters, land irrigation status, farm techniques and the number of workers involved for completing agricultural operations on time.

Introduction

In most developing countries, farm lands are always divided into small parts from one generatoin to another because of inheritance laws. Unfortunately, this situation brings up big mechanization problems such as small and irregular shaped lands for farming. If the dimension and the shape of the farm land is not suitable, agricultural operations such as soil tillage, seed bed preparation, seeding, planting and harvesting with machineries becomes ineffective and expensive. Therefore, the quantity of agricultural products per unit area becomes low, which means low national income. For this reason, those countries with national income dependent on exporting farm products should start to plan land use as soon as possible.

Turkiye is one of the developing countries which has significant improvement in planning the use of farm lands in economic sense. In recent years, a lot of land consolidation projects were completed not only for purpose of planning farm mechanization, but also for effective irrigation planning and landscape work. The main criterion for consolidating farm lands was to determine the optimum size of the parcel that was considered in the project.

In literature, there are similar studies that determine the optimum farm sizes for use of farm machineries in efforts to take advantage of economy of scale. For example, the optimum land size in Europe was determined between 0.4 ha and 6.65 ha [1]; and in Japan, it was 0.3 ha [2], or between 0.5-1 ha for intensive farming [3].

Materials and Methods

The optimum land size for mechanized sugar beet production was obtained as an optimum area from the task time and area loss calculations on rectangular shape parcels when total task time and the unused area of the land were numerically equal. Soil tillage with a plow, sowing sugar beet with a seeding machine, cultivating, pesticide and insecticide control with a sprayer, and harvesting of the sugar beet with a harvester were considered as important agricultural operations for sugar beet mechanization.

The optimization model was based on calculating task time and total area loss of the rectangular shaped parcels whose width/length ratio were changed 1/10 to 10/1. In this case the ra-

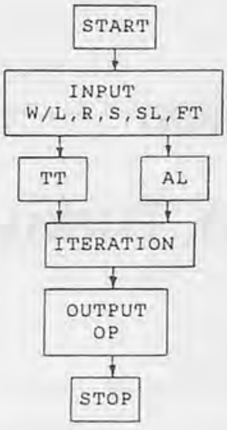

Fig. 1 Flowchart for calculation the optimum parcel size in sugar beet mechanization.

tio 1/1 defines square. The width of the parcel was obtained experimentally to be equal to the maximum flow length of the water for irrigation [4, 5]. The task time model includes the effective working time on the land, machine preparation time for the work, the machine turning time at the end of the parcel, unexpected lost time for machine breakdown, and the time spent on the way. The time lost on the way depended on machine speed on the way and the distance of the land from the main buildings. Similarly, the effective working time was calculated using the machine width and working speed on the land.

The effective working time was the total time about the actual work with farm machineries starting form soil tillage, seeding, cultivating, plant protecting and harvesting on the land. Machine width, working speed and width/length ratio of the land were important parameters for calculating effective working time. Machine turning time at the end of the parcel was calculated by considering normal and spiral farm techniques. Farm machineries were allowed to make U-turn and then started with the second row at the each end of the land in normal farm technique. On the other hand, machineries were allowed to make one quarter turn each time and then started with the farthest row in the second technique. The machine preparation time depended on ability of the worker, the type of the work and machine. However, in this case, the total preparation time was assumed as approximately 5% effective working time [6]. Unexpected lost time in the case of the machine breakdown was difficult to determine. In general, it was also assumed as 2% of the effective working time [6].

The second part of the model was to estimate the unused area in the field as area loss. The unused area consists of the machine turning area at each end of the parcel and the area loss at the parcel edges. Approximately, 50 cm width strip along the edges of the parcel was considered as edge loss [6].

The flowchart for the model is given in **Fig. 1**.

The first step for the calculation of the optimum parcel size for sugar beet mechanization, as shown in **Fig. 1**, was to enter necessary parameters for the system such as width length ratio (W/L) of parcel and its average slope (SL), soil type (S), type of the road (R) used from the main building to the field and the farm techniques (FT) applied. The computer program then calculated the task time (TT) and area loss (AL) by searching the optimum point with numerical iteration. The optimum parcel size (OP) for the sugar beet mechanization was given as final output having the total task time and the total area loss were numerically equal.

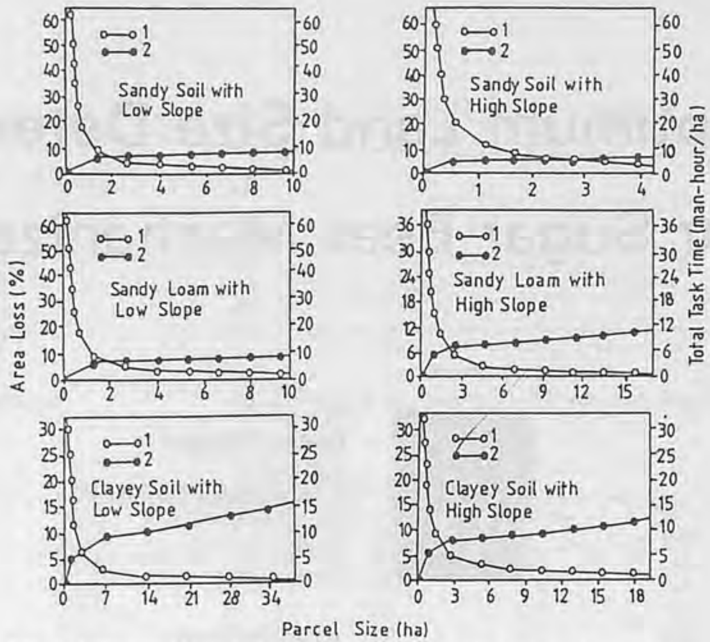

Fig. 2 Area loss (1) and the task time (2) of sugar beet mechanization.

Result and Discussion

For sugar beet mechanization in Turkiye, the optimum parcel size were calculated between 1.6-2.4 ha depending on soil type, machine parameters, land irrigation status, farm techniques and the number of workers involved for completing agricultural operations on time. In **Fig. 2**, the optimum parcel size of the task time and the area loss of the sugar beet mechanization for several conditions are given.

The calculation of the optimum parcel sizes for the sugar beet mechanization was performed for the distance of land to be 1 km from the main buildings. Although the model was able to account for all conditions an asphalt road was chosen as the road type for simplification of the task time calculation. The results of the analysis showed that the

Table 1. Calculated Optimum Parcel Sizes for Sugar Beet Mechanization in Turkiye

Farm Techniques	Soil Type	Parcel Slope (%)		
		0.25	0.5	1.5
Normal	Sandy soil	1.94	1.90	2.18
	Loamy sand	1.59	1.66	1.82
	Clayey soil	1.62	1.59	1.81
Spiral	Sandy soil	2.04	2.04	2.36
	Loamy sand	1.62	1.68	1.86
	Clayey soil	1.64	1.62	1.84

optimum parcel size increased with 14% and 26% when the parcel slope increased from 0.5% and 1.5%. Similarly, it was found that the spiral farm technique required more time to complete farm work which resulted generally at 6% higher in task time per unit area. In **Table 1**, the calculated optimum parcel sizes for sugar beet mechanization are given as final report.

Conclusion

Developing countries should prevent their agricultural lands for dividing into small parcels where agricultural production is ineffective and expensive. Small-sized farms should be consolidated to increase agricultural productivity as the result of the enforcement of new legislation, if necessary. The optimum parcel size for the crop should be determined precisely to be basis for the legislation.

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A Motorized Boat-mounted Aquatic Weeder

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Abstract

An engine-operated aquatic weeder with its various units mounted on a motorized boat structure was designed, fabricated and tested in actual lake conditions. The shape, size, stability and balancing conditions of a catamaran type of boat structure were theoretically analyzed using a computer program developed for the purpose. The dimensions of each hull and the overall dimensions of the catamaran type of boat structure were decided to suit to the requirement of the aquatic weeder.

The machine was tested for 120 h for its balancing, stability and cutting habits in actual lake conditions. The results indicated that the balancing and the stability of the aquatic weeder with its operator of weight 60.0 kg were excellent. The submerged standing weeds were successfully cut by the cutting unit operating 24 cm below the waterline and at a speed of about 1 200 strokes/min. The speed of travel was 3.25 km/h.

Introduction

Excessive weed growth is one of the major problems ailing most of the aquatic bodies in India, e.g., areas of prolonged water logging, irrigation canals, ponds, lakes and reservoirs. The varied nature of weeds, their prolific growth and mixed communities of weeds in varied intensities make the problem of their removal much more serious and difficult.

For the use of aquatic bodies and reclamation of water logged areas, removal of weeds has emerged as important necessity as the problem is on the rise. In this context, the aquatic bodies which need continuous attention to keep them free from weeds are like aquacultural ponds, lakes, irrigation canals, water bodies for recreation and navigation.

Grinwald (1968) and Brayant (1970) suggested that mechanical removal of weeds was economical and effective.

Koegel (1975) studied two commercial harvesters and concluded that the productivity was affected by the width of cut, forward velocity, maneuverability, magnitude of wind and waves, water depth and bottom obstructions, operator's skill, matching of accessory equipment to the harvester and overall conditions.

Velu (1976) designed a machine for eradication of submerged and floating weeds. It consisted of barge and an endless rake. The rake had an inner forked and an outer forked rake designed to uproot the submerged weeds.

Price (1981) while reviewing the different methods of weed control in a drainage channel pointed out that such machines mainly consisted of reciprocating cutter bar with cutter bar speed between 50 and 600 strokes/min, floating barge and propulsion unit.

Sankarnarayanan et al. (1985) developed a harvesting machine for harvesting of *Salvinia molesta* locally known as African Payal.

Singh (1986) designed and developed a water hyacinth harvester. It was equipped with a power transfer system and a transporting trolley of size 3 m length and 2 m width. Propellers were provided at each end of the propelling shaft projecting at the rear end of the barge.

Some efficient mechanical weed harvesting machine units were developed by the English and French companies viz. John Wilder (Engineering) Limited, Wallingford (England) and Simplex Bateaux Faucardeure (France).

A machine developed for eradication of the aquatic weeds should be economical, robust and small in size to suit the socio-economic environment of the Indian farmers and a large number of small as well as large aquatic bodies in India. Keeping this point in view, the present approach is to develop a low cost technology for mechanical weed eradication which could be effective as well as adoptable for the Indian farmers for irrigation, navigation, aquacultural and domestic purpose.

Theoretical Design

General Terminology

The general terminology used in the design of a boat or a ship

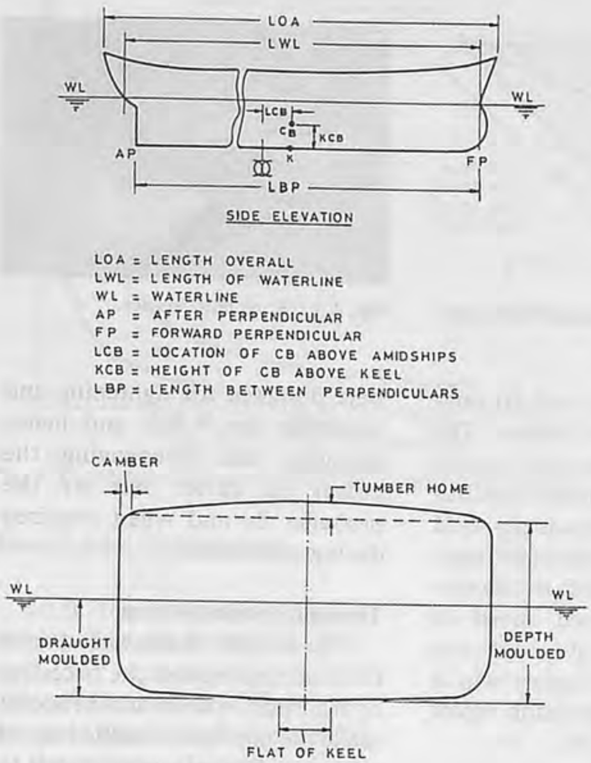

Fig. 1 General terminology for boat in water.

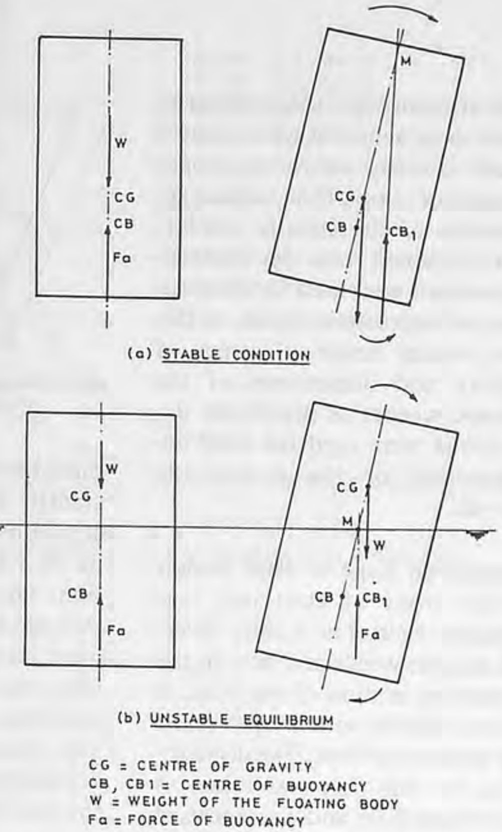

Fig. 2 Stability conditions.

has been adopted (Smith, 1975). Some of these are shown in **Fig. 1**. As shown in **Fig. 2** for the stability conditions, centre of buoyancy (CB) is the centroid of the underwater volume of the boat through which total force of buoyancy is assumed to act. The vertical line through the centre of gravity of the boat (CG) and the centre of buoyancy (CB) in static condition and the vertical line through these points when the boat is subjected to a small angle of inclination interested at a point which is termed as the metacentre (M). In the case of trim it is called longitudinal metacentre.

The distance between the metacentre and CG, metacentre and CB are called metacentric height and the metacentric radius, respectively. The transverse and the longitudinal (metacentric heights and radii) prefixes are associated with the corresponding transverse and longitudinal metacentres.

Floating Principle

A floating body remains stable as long as its metacentre is above its centre of gravity. It may overturn to get a more stable state in case an unbalance force acts on it to change the position of the metacentre or any modification in loading pattern resulting to increase the height of the centre of gravity such that metacentre height becomes negative. Thus, metacentric heights are the measure of the stability of a floating body. The CG of aquatic machine is determined and the transverse and longitudinal metacentric heights are determined corresponding to the maximum possible range of displacement. At each stage, the two values are compared for positive values of metacentric height.

To cope with the various conditions and for checking the stability of the machine in water bodies, it is necessary to have values of the following parameters:

i) displacement;
ii) waterline areas;
iii) position of centre of loatation;
iv) position of centre of gravity of the displacement;
v) position of centre of gravity of the aquatic weeder·
vi) transverse moment of inertia of water planes;
vii) longitudinal moment of inertia of wterplanes;
viii) height of transverse metacentre above the base line.;
ix) longitudinal metacentre above the base line.
x) moment to change trim; and
xi) displacement in kg for one centimetre additional immersion at various draught values.

Design Software

A computer programme in the language Fortran 77 was written and a software was developed at Cyber system available at I.I.T., Kharagpur for deciding the shape, size and the dimensions of the boat-structure for design displacement to be achieved with suitable factor of safety and for the proper accommodation of the various accessories of the aquatic weeder. Data obtained from this theoretical analysis were used for development of hydrostatic curves. In this case, actual height of centre of gravity and dimensions of the aquatic weeder as practically determined were used for final development of the hydrostatic curves.

Catamaran Type of Hull Design

The front of each hull was designed such that it may divert the aquatic weeds and help in the streamline motion of the boat. It is conic section with its apex at the top level of the boat. The displacement by this part was taken as additional float and was not used during theoretical analysis. Based on the slant height along the drum, dimensions of this frontal part of each hull were decided. As shown in **Fig. 3**, D = 58 cm, λ = 38.94, ϕ_1 = 51.06 and ϕ_2 = 19.47 where λ is the angle made by the bottom most slant height with the keel level in the horizontal plane.

Propelling Unit

A three-bladed screw propeller rotating inside water generates backward thrust for the forward travel of the aquatic weeder. The rotational speed of the screw propeller was selected so as to have sufficient backward thrust avoiding the phenomenon of cavitation and the splashing of water.

Cutting Unit

An one-metre reciprocating type of cutter bar was selected. Power to the cutter bar was provided through a cam and a connecting rod arrangement. A suitable angle iron frame was fabricated to support the cutter bar. The cam shaft was mounted at the bottom of this frame using

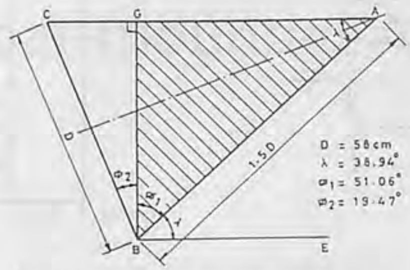

Fig. 3 Wedge-shaped conical frontal part of a hull (▨).

bush-bearings as it had to constantly work under water. The stroke of the cutting unit was to be 76.2 mm with proper registration. The cam shaft was powered through a sprocket-chain arrangement from main shaft of the cutting unit. The rated speed of operation of the cutter bar was kept around 1 000 strokes/min at a fixed depth of operation below the waterline **(Fig. 4)**.

Steering and Navigation System

A rudder is provided at a distance of 45 cm in front of the screw propeller in the middle line plane. The turning moment to the machine was provided by a rectangular aluminum plate of size 34.5 × 33.5 × 0.4 cm acting as rudder. This helps to take the thrust generated by the screw propeller either on the left or on the right side of the middle line plane. This was actuated by the steering wheel for taking turn.

Power Transfer System

Mitsubishi-AD8 engine with rated power of 8 hp and maximum power of 10 hp was used as power source during operation. A pulley rated rpm and the maximum rpm of the engine are 1 500 and 1 600, respectively. The power from the driving V-pulley (double-grooved) mounted on the engine crank shaft to the driven V-pulley was led through Idler V-pulley in the bracket attached to the lever arm by V-belt **(Fig. 4)**. The spring loaded key and slot arrangements

Fig. 4 I.I.T. aquatic weeder.

were designed for tightening and loosening the V-belt and hence engaging and disengaging the power to cutter bar or the propeller as and when required during operation.

Transportation System

The weight, shape and size of the machine created the necessity of transport wheels and hence to make the machine towable behind a tractor. For transport wheels in the front and in the rear were fixed to the main frame of the machine, which wheels are removed after lowering the machine inside the aquatic body and are fitted again to the main frame before taking the machine out of the aquatic body for transport.

Operator's Seat

The sitting arrangement for the operator and for other two persons if required were provided. While designing the seat for the operator ergonomic considerations were taken into account such as approach to the controls and back rest with seven degrees backward tilt from vertical. Water acts as a natural shock absorber during the operation of the machine. At the same time, nagligible bumping action is expected inside water so only a cushion was provided while no seat suspension system is required. This reduces the overall cost of the machine.

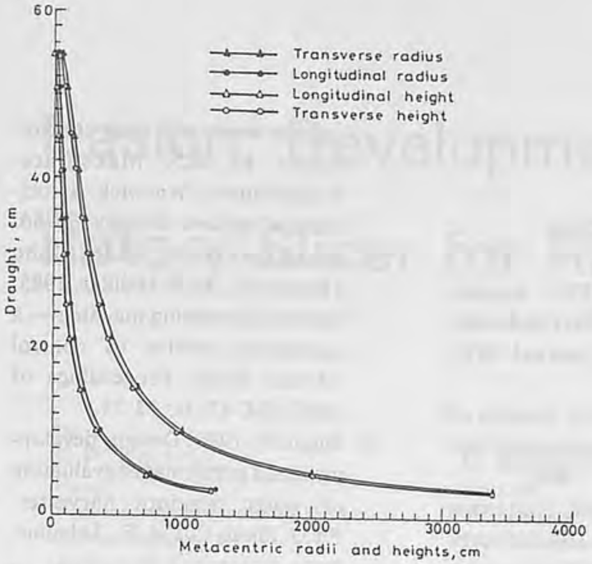

Fig. 5 Variation of metacentric radii and heights with draught.

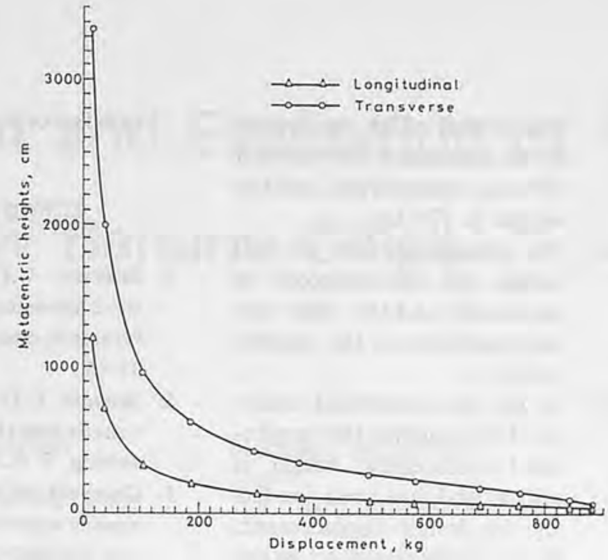

Fig. 6 Variation of metacentric heights with displacement.

Result and Discussion

Using the computer software developed values determined were used for generation of the hydrostatic curves. These are shown in **Figs. 5** to 7. The height of the centre of gravity (CG) above the keel was 42 cm above the keel level. The comparison of hydrostatic curves for the transverse and longitudinal metacentric heights is critical. It is still 57.5 cm for a corresponding displacement of 600 kg. This height is sufficient for safe operation of the machine with the design displacement of 600 kg. The corresponding draught level is 36.5 cm.

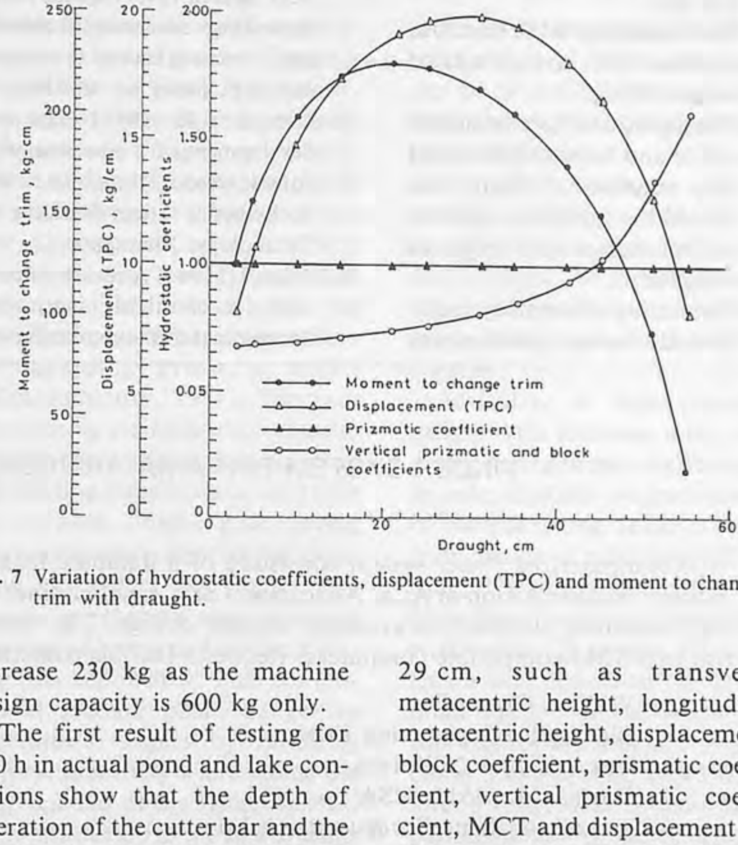

Fig. 7 Variation of hydrostatic coefficients, displacement (TPC) and moment to change trim with draught.

At this design the value of draught the height of centre of buoyancy and transverse metacentric height are 20.5 cm and 201 cm, respectively. The corresponding value of block coefficient, vertical prismatic coefficient, and prismatic coefficient are 0.87, 0.87 and 1.0, respectively. The moment to cause trim (MCT) and displacement per unit draught at the design draught of 36.5 cm are 195 kg-m and 19.15 kg/cm, respectively. The maximum value of MCT obtained was 225 kg-m as apparent from the analysis. Since the weight of the machine is 370 kg, any extra loading on the machine should not increase 230 kg as the machine design capacity is 600 kg only.

The first result of testing for 120 h in actual pond and lake conditions show that the depth of operation of the cutter bar and the screw propeller were 24 cm and 45 cm below the waterline. The balancing and stability of the machine were excellent with an operator of 60 kg at a draught of 29 cm. At this value of draught, transverse metacentric height was 283.9 cm and the longitudinal metacentric height was 86.2 cm. the value of other parameters corresponding to this of draught of 29 cm, such as transverse metacentric height, longitudinal metacentric height, displacement, block coefficient, prismatic coefficient, vertical prismatic coefficient, MCT and displacement per cm draught are 11.13 cm, 283.9 cm, 86.2 cm, 456.5 kg, 073, 1.0, 0.73, 217 kg-m and 19.9 kg/cm, respectively.

Conclusions

The following conclusions were drawn from present level of investigation:

i) The overall width and length of the machine is 234 cm and 247 cm, respectively, and the weight is 370 kg.
ii) The catamaran type of hull design was advantageous in movement and the clear cutting operation in the aquatic bodies.
iii) As per the theoretical analysis of the machine the longitudinal metacentric height is critical and was used for fixing the design displacement. Hence, additional load on the machine should not exceed 230 kg.
iv) The balancing of the boat was excellent with an operator of weight 60 kg.
v) The aquatic weeder was quite stable and balanced in actual lake conditions which confirmed the sufficient value of critical metacentric height as designed.
vi) The cutting of standing aquatic weeds during operation was satisfactory.

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Design, Development and Evaluation of Rotary Slicer for Raw Banana Chips

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Abstract

An electrically-operated rotary slicer for raw banana (*Musa paradisiaca*) was designed, fabricated and evaluated. It has a capacity of slicing 61 kg peeled banana per hour with an efficiency of 90%, which is two times higher than the capacity achieved by conventional method. It produces chips of uniform thickness with roundness index of 0.85. The machine costs Rs. 1 700 and the cost of operation for producing one kg of wet chips is Rs. 0.15.

Introduction

Banana wafers are very popular snack food in South India because of its crispness. The wafers are made throughout the year since salt fried banana wafers have very good demand in both internal and external markets. Four major unit operations, viz; peeling of fruit, cutting of fruit into slices, frying and packaging are involved in the present practice of making wafers. Each unit operation is carried out manually and household tool knife is used for peeling and cutting of the fruit. A few entrepreneurs use platform type manual slicer. More than 200 banana chips makers are engaged in this business at small and cottage level in and around Coimbatore district of Tamil Nadu alone. Banana wafers are produced from fully matured unripe fruit of 'Nendran' variety. The simplest mode of processing is that banana fruits are hand-peeled using stainless steel scouring knife and then steeped in 2% salt solution for 10-15 min to maintain the colour by avoiding browning effect (Krishnankutty, 1981). Slicing is done using stainless steel adjustable wooden platform hand slicer by holding three bananas at a time in between fingers and moving across the sharp edge of the slicer. The cut chips, due to moving force, are thrown away through the slot adjacent to knife in a frying pan kept below. This conventional method poses danger to operator's fingers by inflicting injury, especially while slicing the fag/tail end of the fruit. Besides, non-uniformity in the size of chips results in poor end quality of wafers after frying. The output of the conventional method with one person peeling and another person slicing and frying has been found to be around 50-70 kg of fried wafers/day. Therefore, operating cost works out to be about Rs. 0.30* per kg of wet slice.

The consumer preference for banana wafers is based on colour, crispness, aroma and taste (Krishnankutty et al. 1981). Crispness can be controlled by maintaining uniformity of chip thickness and proper frying.

To avoid drudgery and any injury to workers, enhance the capacity and to maintain quality and hygiene, the present research was undertaken in the inferest of farm level and small-scale industries.

Initially, a hand-operated gadget with stainless steel string slicing arrangement was developed in order to study the performance of banana slicing. It had a hollow frame made of mild steel 250 mm long, 130 mm wide and 70 mm high (**Fig. 1**). About 126 stainless steel strings (28 gauge), 7.5 mm in length were tied across the longitudinal side at 5 mm above the bottom edge of the frame. The frame could move up and down by the lever arrangement. During the experimentation, peeled banana was kept on a 10 mm raised platform and the frame with strings moved down manually so as the strings pierced into fruit pulp to cut slices. Due to the blunt edge of string, a force of about 28 N was required to pierce a single string into the banana (Kachru et al. 1994). When so many strings acted over one fruit at a time, the force requirement for strings to pierce

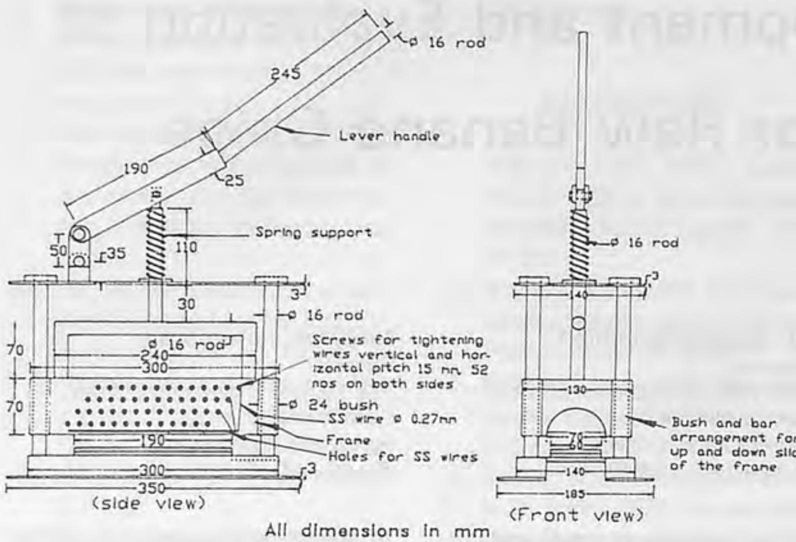

Fig. 1 Multiple stiring banana slicer.

into fruit was very high (about 3.5 kN), causing compression of fruit from the bottom and in the case of softer fruit, distortion of the pulp. This restricted the strings to reach to the bottom of the fruit and complete the cut. Moreover, due to stickiness of the pulp, removal of slices after cut was difficult.

To overcome the drawbacks of the string slicer, a motorized rotary banana slicer was designed and developed.

Design of Machine

The peeled raw banana has varying radius of curvature (Kachru et al. 1994) and the diameter of pulp varies lengthwise as well as fruit to fruit. The chipping is carried out directly over frying pan. Therefore, the feeding mechanism, cutting disc and outlet were the three important parts of the machine to be designed.

The design of the slicing disc was based on cutting velocity, shear angle, bevel angle and number of blades. The diameter of cutting disc on which blades were to be mounted was 22 cm, i.e., approximately three times the maximum effective width of banana.

An experiment was conducted to determine the design parameters of blades using a pendulum type impact testing machine. Blades of different bevel angles, viz; 15°, 20°, 25° and 30° were mounted on the pendulum and the stationary fruit sample, kept normal to the direction of pendulum movement, was cut. The energy consumed in cutting the fruit and the velocity of cut by different blades were calculated using the following expression (Visvanathan et al. 1990):

$$E = W_p R_c(\cos\theta_c - \cos\theta_o) \quad \ldots(1)$$

and

$$V = L.\frac{2E}{I} \quad \ldots(2)$$

The minimum value of energy requirement was 7.36 kg-cm for 15° bevel angle of blade and the velocity of cut for this blade was 179.76 cm/s.

The torque available on chipping disc was calculated as follows:

$$v = \frac{\pi DN}{60} \quad \ldots(3)$$

Substituting the values for D as 0.22 m, v as 1.80 m/s and considering factor of safety as 1.6, we get:

$$N = 250 \text{ rpm}$$

For calculating the power given to cutting disc, the following expression was used (Balasubramanian et al. 1993)

$$P = \frac{2\pi NT}{4\,500\eta} \quad \ldots(4)$$

Substituting the values for P as 0.1 hp, N as 250 and η as 0.9, we get:

$$T = 25.77 \text{ kg-cm}$$

This torque was available to perform slicing of the fruit pulp. The possible number of slices which could be cut in one rotation of disc was

$$n = \frac{T}{E} \quad \ldots(5)$$

Substituting the values for T as 25.77 kg-cm and E as 7.36 kg-cm:

$$n = 3.50$$

The proposed slicer disc would have 3 blades with 15° bevel angle.

The total energy stored in the rotating disc was given by Balasubramanian et al. (1993) as,

$$E_d = Iw^2C_s \quad \ldots(6)$$

where,

$$w = \frac{2\pi N}{60} \quad \ldots(7)$$

$$I = \frac{W_d R^2}{g} \quad \ldots(8)$$

$$W_d = \pi R^2 t\sigma \quad \ldots(9)$$

Substituting the values for E_d as 25.77 kg-cm, g as 981 cm/s^2, R as 11 cm, N as 250 rpm, σ as 2 300 kg/m^3 and C_s as 0.64 kg;

$$t = 5.50 \approx 6 \text{ mm}$$

Hence, the disc thickness would be 6 mm.

Development of Machine

The electrically-operated rotary banana slicer is a horizontal feed type unit (**Fig. 2**). It consists of a frame assembly, slicer disc with blades, disc cover, feeding chute and the primemover.

Frame

The slicer had overall dimensions of 400 × 240 × 230 mm and fabricated of ISA 25 × 25 × 3 MS section. On this frame were mounted the bearings, driven shaft, disc cover and primemover.

Slicer Disc

A 6-mm thick chrome plated mild steel disc of diameter 22 cm provided with three slots of 65 mm × 17 mm separated by 120° was fabricated. Each of these slots carried a blade of 60 × 50 mm with a bevel angle of 15° for slicing. the blades were mounted at an angle of 25° to the surface of disc and protruded out on cutting face of disc and could be adjusted for choosing the desired thickness of slice.

A 465-mm long mild steel shaft with a diameter of 23 mm was chosen to drive the slicing disc and provide ample space to keep the frying pan directly below the slice outlet.

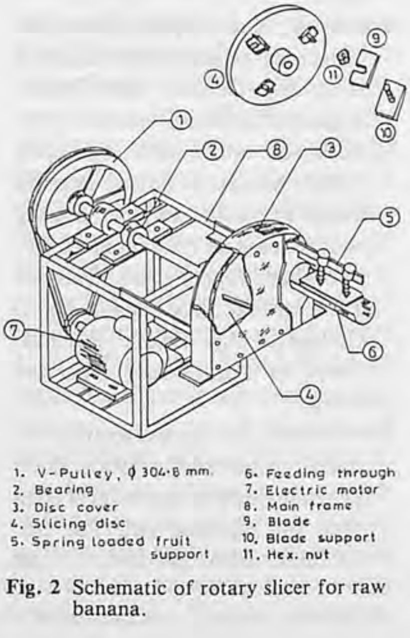

Fig. 2 Schematic of rotary slicer for raw banana.

Feeding Chute and Disc Cover

Peeled banana is reported to have a maximum effective width of 66.5 mm and minimum effective length of 137 mm (Kachru et al. 1994). Therefore, a stainless steel semi-circular feeding chute, with a diameter of 70 mm and 130 mm long was fabricated. The chute was fixed to the cover of disc at the 3 o'clock position, offset from the center by 75 mm, in order that the chips are directly discharged into the pan by centrifugal action (**Fig. 2**). In this way, the problem of chips adhering to the disc cover was also avoided. The tip of the feeding chute entering the disc cover had a clearance of 2 mm from the tip of the blades. The feeding chute had a spring loaded top support to dampen vibration and guide the peeled banana while slicing.

Primemover

A single phase, 230 VAC, 50 Hz, 1 440 rpm TEFC motor rated at 75 W serves as a primemover. The drive was transmitted through V-belt pulley with 6 times speed reduction, so as to produce 250 rpm at the disc. The motor was mounted at the base of the frame assembly on slotted plate so as to adjust the belt tension. The motor weight also gave stability to the machine.

Performance Evaluation

The operating capacity of the machine was determined by feeding peeled banana into the machine and finally weighing all slices irrespective of damage per unit time. The efficiency of slicing was determined by the following expression:

$$\alpha = [(\text{wt.of all slices} - \text{wt.of damaged slices}).100]/(\text{wt.of all slices}) \quad \ldots(10)$$

The effective capacity was determined using the following formula:

$$E.C. = O.C. \times \alpha/100 \quad \ldots(11)$$

Three randomly-drawn sample lots, each carrying 50 slices, were used for determination of thickness, uniformity of thickness and roundness of slices. For each slice, thickness was determined at four places around the periphery of slice using micrometer with a least count 0.01 mm.

The roundness of each slice was determined using the following expression (Mohsenin, 1981):

$$\text{Roundness} = \frac{A_p}{A_c} \quad \ldots(12)$$

The moisture content was determined by keeping 3.5-4.0 mm thick banana slices in single layer in an aluminum dish in an oven at 130±1°C for 1.5 h (Rameshbabu and Nayak, 1993).

Results and Discussion

The rotary slicer was evaluated for its performance on the basis of its capacity, efficiency, and quality parameters of chips for the 'Nendran' variety of banana.

The average values of slicing efficiency, operating capacity and effective capacity at 189.7% (db) moisture content of banana pulp were 89.95%, 60.66 kg/h and 54.57 kg/h, respectively.

The mean thickness of the cut slices was 1.55 mm with a standard deviation of 0.27 mm. The frequency analysis of the data showed that 95% of the chips had thickness ranging between 1.0 to 2.0 mm. The slicing was performed by increasing the protrusion of blades and in this case, the mean thickness of the slices was 2.42 mm with a standard deviation of 0.05 mm. The frequency analysis of the data showed that 100% of the chips had thickness ranging between 2.0 mm and 2.5 mm. Also, the Chi-square analysis of the data revealed that the slice thickness was significantly uniform at 95% confidence level.

The slices were cut with average roundness index of 0.852 with maximum, minimum and standard deviation of 0.996, 0.712 and 0.057, respectively.

The economic analysis of mechanized process of slicing is shown in **Table 1**. The analysis is carried out considering that the entrepreneur would buy the raw material (raw banana), slice and fry and sell the final product. The analysis shows that the cost of operation per kg of wet chips was reduced to 50% by mechanizing the process. This substantial reduction in operational cost is due to an increase in fried wafer production from 50-70 kg/day to approximately 200 kg/day. The investment on the machine is justified by the increase in the return on the investment.

Table 1. Economic Analysis* of Banana Slicing

Particulars	Conventional slicing	Motorized rotary banana slicer
Cost of machine, Rs.	50	1 700
Working capital, Rs./day	1 960	6 700
Cost of slicing, Rs./h	5	9
Cost of slicing, Re./kg of wet chips	0.30	0.15
Break-even-point, kg/year	N.A.	767
Return on investment, %	22.4	34.2
Pay-back period, days	N.A.	15
Gross profits, Rs./month	7 000	38 000

*100% sale basis.

Assumptions: In both cases, operation would continue for 8 h/day and 200 days/year. The raw banana is available for Rs. 8/kg. The fried chips produced by conventional method are sold for Rs. 40/kg and produced by motorized rotary banana slicer would fetch a higher price of Rs. 45/kg due to better quality. The operation involves one skilled and one unskilled labourers, paid Rs. 35 and Rs. 20/day, respectively.

Nomenclature

- α = Efficiency of slicing, %
- A_p = Projected area of a slice resting on most stable position, mm^2
- A_c = Area of smallest circumscribing circle to the slice, mm^2
- C_s = Fluctuation of speed, kg
- D = Diameter of cutting disc, cm
- E = Minimum energy required to cut a slice, kg-cm
- E_d = Energy stored in disc, kg-cm
- E.C.= Effective capacity of the machine, kg/h
- F_c = Centrifugal force, kg
- g = Acceleration due to gravity, 981 cm/s^2
- I = Moment of inertia, kg-m-s^2
- L = Length of pendulum, cm
- MS = mild steel
- N = Speed of rotation of disc, rpm
- N.A= Not applicable
- O.C= Operating capacity of the machine, kg/h
- P = Power given to cutting disc, hp
- p.a. = per annum
- R_c = position of centre of gravity of the pendulum from axis of rotation, cm
- Rs = Indian Rupees
- r = Radius of cutting disc, cm
- t = Disc thickness, mm
- T = Torque available on disc, kg-cm
- TEFC = Totally enclosed fan cooled
- v = linear velocity for cut, cm/s
- w = angular velocity
- W = Watt
- W_d = Weight of disc of diameter D, kg
- W_p = Weight of pendulum, kg
- W_s = Weight of slice, g
- η = Mechanical efficiency (assumed 90%)
- σ = Density of disc material (2 300 kg/m^3)
- θ_o,θ_c = Angles subscribed by the pendulum after equilibrium position when released from rest at any angle, in-absence of cutting and after cutting, respectively.

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Energy Utilization Pattern in the Manufacture of Black Tea

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Abstract

The consumption of direct energy from major sources in tea industry in Assam was studied. Two important sectors of the tea industry are garden and factory. Human labour is the major source of energy for different unit operations in the garden. A tea garden requires an estimated 18 408 MJ/ha of human energy in the first year. The demand decreases from 2nd year and remains at about 3 653 MJ/ha per year from 5th year onwards. Besides, 2 262 MJ/ha of energy is required initially for land preparation and about 86 MJ/ha annually for green leaf transportation in the form of diesel. Major commercial sources of energy in the factory are electricity, coal, oil and gas. The degree of variation in electricity consumption in producing one kg of made tea among the factories is less than the variation in the use of other fuels. Electricity consumption varies from 5.56 MJ/kg to 13.48 MJ/kg of made tea. Fuel energy requirement varies from 14.9 MJ/kg to 63.62 MJ/kg among the factories. Total energy consumption is greater for factories where coal is used as fuel. The factory using only oil is the most economical in energy consumption. Further investigation is needed to explore the possibility of reducing the variation in the rate of energy consumption in tea manufacturing.

Introduction

Tea occupies a prominant position in the industrial map of India, in general, and Assam in particular. It is one of the bigger export earners as well. From the production level of 758 million kg in 1993 to reach the production target of 1 000 million kg by the year 2 000 A.D., the Indian tea industry will need to achieve a fairly rapid growth rate in the remaining years of this century. To maintain the present level of export, the industry will need to substantially improve its productivity at the same time reducing the cost of production. In view of this, therefore, attempts should be made for higher efficiency of utilization of fuel and electricity, these being two major components of manufacturing cost.

Production of tea involves operations in both garden and factory. Human labour is the major source of energy for different unit operations in the garden. On the other hand, commercial energy dominates the factory operations.

The tea plant produces pluckable tea leaves from 1st year onwards although it attains its full potential from the age of 5 years. Besides performing normal cultivation practices, the tea plant is trained as a low spreading bush through treatments like centering, pruning and plucking.

In the factory fresh green tea leaves are subjected to various processes (Werkhoven) to arrive at the marketable product (**Fig. 1**). The first step in the manufacture of black tea is withering wherein the fresh tea leaves are partially dried in a current of air, mostly at

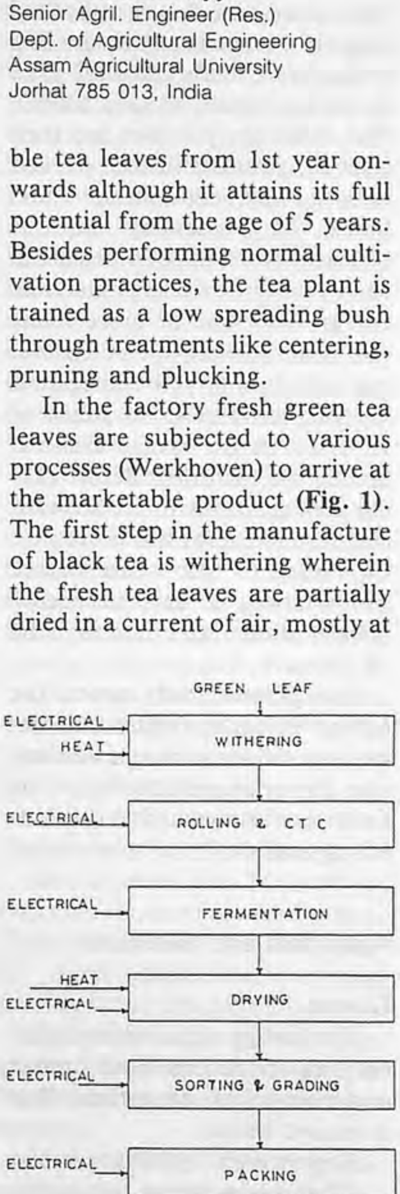

Fig. 1 Tea manufacturing process.

ambient conditions with some hot air utilization. The withered leaves are then rolled where the leaves are twisted, torn and crushed. In a related and subsequent process called CTC, more severe leaf damage occurs. The leaves then undergo a process of oxidation (fermentation) wherein enzymic and chemical reactions occur. During this process an air flow is provided through a humidifier. The fermented leaves are then dried in a current of hot air. The temperature levels generally used in drying remain around 100°C. The dried tea particles are then cleaned, graded and then packed into suitable containers. Thus, black tea processing requires primarily two types of commercial energy - electrical energy and heat energy from one or more fuels, viz, coal, furnace oil or natural gas. Preliminary investigation reveals that there exists plenty of variation in the energy demand among the factories. Before taking any measures to ensure efficient use of energy it is necessary, therefore, to know the degree of variation, if any, in energy consumption rates among the factories.

The present study covers the energy use pattern under different unit operations in the garden and also during manufacturing in the factories of some randomly selected gardens.

Materials and Methods

Garden

The energy required in different unit operations, both human and mechanical, are estimated as discussed below.

Requirement of human energy — The human energy required to perform different unit operations in the garden are estimated assuming an economic life span of 50 years for the tea plant. The man-days required to perform the unit operations per ha of land are obtained from standard task rates. These are then converted into equivalent energy by using suitable energy coefficient value (Datta *et al.*). Annual energy requirements during its life span for different unit operations and the total energy required per year per ha were also calculated.

Requirement of fuel energy — Fuel energy is required in the garden mostly during the operations of land preparation and green leaf transportation. Since land preparation is carried out only at the beginning, it is included in the first year only. One ploughing followed by two harrowing operations are assumed. Fuel consumption values (ℓ/ha) for these operations are converted into equivalent energy (MJ/ha) by using appropriate equivalent coefficient. A 26 kW tractor is assumed for the above.

For transporting green leaf from garden to factory, tractor trailers are normally used. Fuel consumption values (ℓ/km) for this operation are converted into equivalent energy (MJ/ha) by appropriate equivalent energy coefficient with the following assumptions:

Tractor size : 26 kW
Average transportation distance : 1 km
Trailer carrying capacity : 2 400 kg/trip
Annual made tea production : 2 000 kg/ha
Average recovery : 21%

Fuel consumption values in the above operations were taken from Test Report of a 26 kW tractor.

Energy consumption per kg of made tea — Annual energy consumption values calculated in MJ/ha are converted to MJ/kg of made tea.

Factory

For the purpose of this study, factories of 15 gardens located in Assam were selected at random and the following information was collected:

i) Annual made tea output, kg
ii) Annual electricity consumption, kWh
iii) Annual fuel consumption, kg of coal/ℓ of T.D. oil/m^3 of natural gas.

Energy consumption — Electricity consumption (kWh) of all the 15 factories were converted into equivalent energy (MJ) by using appropriate equivalent coefficient. Similarly, consumption values of other fuels were also converted into equivalent energy by their respective equivalent coefficients.

Energy consumption per kg of made tea — Energy required per kg of made tea was calculated from the annual production and annual energy consumption values for each garden. Similarly, the contributions made by electricity and different fuels used in producing one kg of made tea as percentages of the total energy consumption have also been determined.

Results and Discussion

Requirement of Human Energy in the Garden

The annual human energy requirements for different unit operations in tea garden are shown in **Table 1**. From the Table it is observed that maximum energy (18 408 MJ/ha) is required in the first year of tea cultivation. This is due to the fact that operations like cleaning and levelling of the land, preparation of land, application of organic matter, centering and pegging, etc., are to be performed only in the first year. Among the above operations, centering and pegging demand the highest human energy (2 038 MJ/ha) followed by clean-

Table 1. Annual Human Energy Consumption in Garden for Different Unit Operations

Unit Operation	Energy required (MJ/ha)				
	1st yr.	2nd yr.	3rd yr	4th yr	5th yr. onward
Cleaning & levelling of land	1 960	—	—	—	—
Land preparation	784	—	—	—	—
Drainage ditch digging & maintenance	1 176	157	157	157	157
Organic matter application	470	—	—	—	—
Planting	7 213	737	376	—	—
Shade tree planting/ maintenance	706	157	157	157	78
Mulching	2 352	784	—	—	—
Centering & pegging	2 038	—	—	—	—
Pruning	157	314	470	627	627
Tipping & plucking	235	314	627	941	941
Cultivation	941	1 882	1 411	1 411	1 411
Application of plant protection chemical	188	188	188	251	251
Application of fertilizer	188	188	188	188	188
Total	18 408	4 721	3 574	3 732	3 653

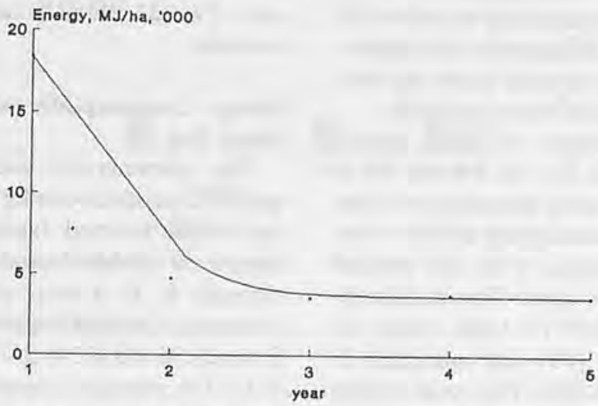

Fig. 2 Year wise human energy consumption.

ing and levelling of land (1 960 MJ/ha). However, depending on the initial condition of the land, energy requirement in cleaning and levelling may vary. Again, planting operation needs the highest energy (8 326 MJ/ha) among all operations and primarily it is only during the first year. In the first year it needs 7 213 MJ of energy to plant one ha of land; some amount of human energy is also needed in the second year (737 MJ/ha) and third year (376 MJ/ha) of cultivation. Mulching operation is done mainly in the first year demanding 2 352 MJ/ha of energy and 784 MJ/ha in the 2nd year.

Except for the six field operations mentioned above, generally, all other operations are performed during each year of the plantation's life span. But there exists a variation in their annual energy demands, particularly during the first five years of cultivation. An estimated 1 176 MJ/ha of energy is needed in the first year for digging the drainage ditches and for maintaining the ditches requiring some 156 MJ/ha of energy every year.

Similarly, for planting shade trees the major portion of energy (706 MJ/ha) is needed in the first year, 157 MJ/ha is needed from second to fourth year and a constant amount (78 MJ/ha) is necessary from fifth year onwards for maintenance. Contrary to the above two operations, annual energy demand for operations like pruning, tipping and plucking, and application of plant protection chemicals are low during the initial years. For the operations like pruning and tipping and plucking energy requirement increases from the first to fourth year and then remains constant for the remaining period. It is estimated that 941 MJ/ha of manual energy is needed for tipping and plucking and 627 MJ/ha for pruning from fourth year onwards. Tea plants take time to develop fully and this is the reason for the low level of energy requirement during initial growing period. An estimated 251MJ/ha of human energy is needed from fourth year onwards for the application of plant protection chemicals and 188 MJ/ha every year for fertilizer application. Much care is taken of the newly grown plants which is why the second year requirement maximum energy (1 882 MJ/ha) for cultivation operation. The annual demand for energy in cultivating one ha of cropped area from third year onward is 1 411 MJ.

From **Table 1** and **Fig. 2** it is also seen that total energy requirement from third year onwards remains more or less constant, the slight difference being accounted for mainly by the operations of pruning, tipping and plucking. It is estimated that the total annual human energy requirement in the garden will remain at 3 653 MJ/ha from the fifth year onwards.

Requirement for Fuel Energy

It is observed that 2 262 MJ of energy is required for land preparation per hectare. This is the energy from diesel fuel to run tractor-driven tillage tools.

To transport green leaf from garden to factory an estimated 86.47 MJ/ha of energy is needed annually.

Table 2 shows the energy requirement in the garden to produce one kg of made tea from different direct sources. Land preparation needs 1.13 MJ/kg in the 1st year only, whereas

Table 2. Energy Requirement in Garden per kg of Made Tea

Operation	Source	Energy (MJ/kg of made tea)
Land preparation	diesel	1.13
Manual operations in garden	human	1.978
Green leaf transportation	diesel	0.043

Table 3. Annual Output of Made Tea and Commercial Energy Consumption in 15 Different Factories

Factory	Made tea output '000 kg	Energy consumption, GJ				
		electrical	coal	oil	gas	total
F1	366	2588	23285	—	—	25873
F2	520	7010	—	—	16458	23468
F3	568	6651	27292	—	—	33943
F4	636	3841	—	—	17719	21560
F5	642	3569	—	—	18220	21789
F6	757	8054	30310	7646	—	46010
F7	793	7018	—	—	14781	21799
F8	827	7625	28531	—	—	36156
F9	1208	12998	55435	—	—	68433
F10	1249	8348	—	—	34447	42790
F11	1350	11488	53217	14404	—	79110
F12	1633	10874	—	—	45642	56518
F13	1694	12637	—	—	51785	64422
F14	1894	17008	—	28220	—	45228
F15	2272	15745	62979	21129	—	99854

green leaf transportation needs 0.043 MJ/kg every year, both in the form of diesel. Human energy requirement in the garden is estimated at 1.83 MJ/kg considering an annual consumption level of 3 653 MJ.

Energy Requirement in Factory

Annual made tea production and annual consumption of four different types of commercial energy are presented in **Table 3**. Electricity is used in all the factories with a variation in its annual consumption from 2 588 GJ (F1) to 17 008 GJ (F14). Electricity consumption is not proportional to annual output. Variation that exists in the type and amount of other sources of commercial energy used in different operations in the factory might account for this.

Except for three factories (F6, F11 and F15) all others use only one type of fuel. These three factories use both coal and T.D. oil. Total energy consumption of F6, F11 and F15 are 46 010, 79 110 and 99 854 GJ, respectively. Total energy consumption as well as its individual components vary directly with the annual made tea output for these three factories.

The source of heat energy in factories F1, F3, F8 and F9 is coal only. Both electrical and heat energy consumption of these factories increase with the annual made tea output. The maximum consumption of total energy is 68 433 GJ (F9) and minimum is 25 873 GJ (F1). The total energy consumption of factory F14, where T.D. oil only is used is 45 228 GJ.

Natural gas is used as source of heat energy in seven factories, viz, F2, F4, F5, F7, F10, F12 and F13. Excepting F2, the total energy consumption in the other six factories varies directly with the output. However, variation in total energy consumption in factories F4 (21 560 GJ), F5 (21 789 GJ) and F7 (21 799 GJ) are very nominal.

Energy Consumption per kg of Made Tea

The commercial energy required to produce one kg of made tea in the selected factories are shown in **Table 4** and **Figs. 3** through **8**. It is seen that total commercial energy required varies from 23.88 MJ (F14) to 70.69 MJ (F1). The average consumption is 44.36 MJ with a standard deviation of 13.92. The total fuel energy requirement varies from 14.9 MJ/kg (F14) to 63.62 MJ/kg (F1) with a mean and standard deviation of 35.78 MJ and 13.26, respectively **(Fig. 8)**. On the other hand, the requirement of electrical energy shows lesser variation with minimum and maximum of 5.56 MJ (F5) and 13.48 MJ (F2),

Table 4. Energy Consumption per kg of Made Tea

Factory	Energy consumption, MJ/kg made tea				
	electricity	coal	oil	gas	total
F1	7.07	63.62	—	—	70.69
F2	13.48	—	—	31.65	45.13
F3	11.71	48.05	—	—	59.76
F4	6.04	—	—	27.86	33.90
F5	5.56	—	—	28.38	33.94
F6	10.64	40.04	10.1	—	60.78
F7	8.85	—	—	18.64	27.49
F8	9.22	34.50	—	—	43.72
F9	10.76	45.89	—	—	56.65
F10	6.68	—	—	27.58	34.26
F11	8.51	39.42	10.67	—	58.60
F12	6.66	—	—	27.95	34.62
F13	7.46	—	—	30.57	38.03
F14	8.98	—	14.90	—	23.88
F15	6.93	27.72	9.30	—	43.95

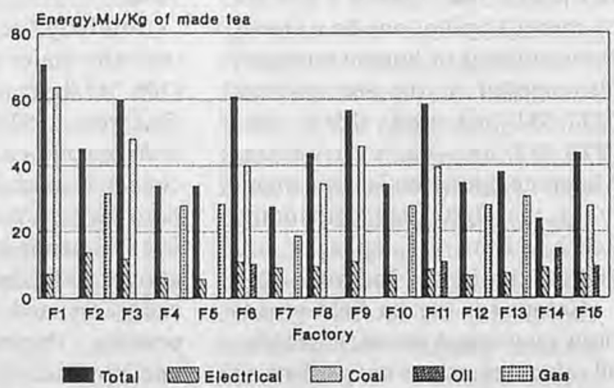

Fig. 3 Energy consumption pattern.

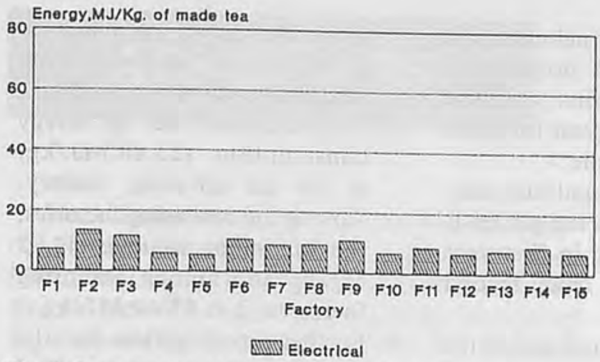

Fig. 4 Consumption pattern of electrical energy.

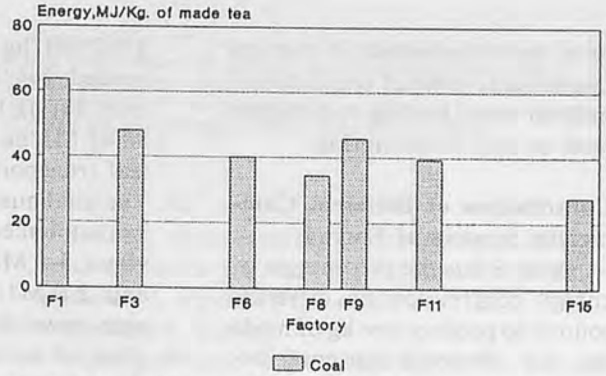

Fig. 5 Consumption pattern of coal energy.

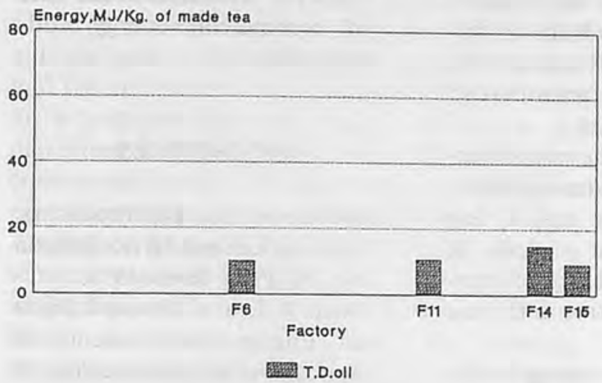

Fig. 6 Consumption pattern of oil energy.

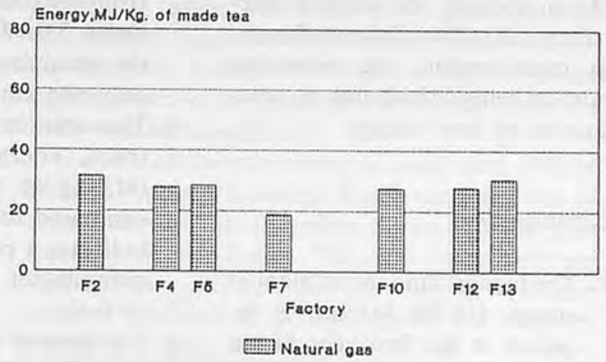

Fig. 7 Consumption pattern of energy of natural gas.

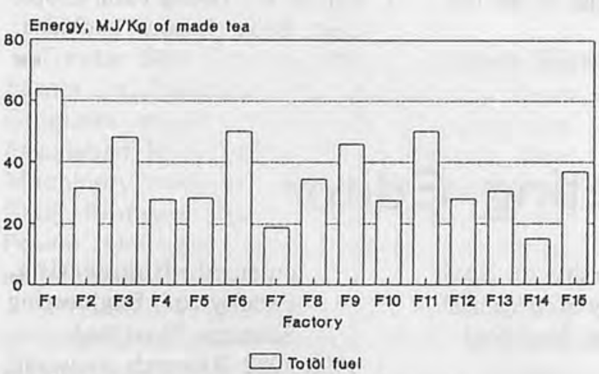

Fig. 8 Consumption pattern of total fuel energy.

Table 5. Contribution from Different Sources as Percentages of Total Energy

Factory	Contribution (percentage of total)			
	electricity	coal	oil	gas
F1	10.00	90.00	—	—
F2	29.87	—	—	70.13
F3	19.60	80.40	—	—
F4	17.82	—	—	82.18
F5	16.38	—	—	83.62
F6	17.51	65.88	16.61	—
F7	32.19	—	—	67.81
F8	21.09	78.91	—	—
F9	19.00	81.00	—	—
F10	19.50	—	—	80.50
F11	14.52	67.27	18.21	—
F12	19.24	—	—	80.76
F13	19.62	—	—	80.38
F14	37.60	—	62.40	—
F15	15.77	63.07	21.16	—

respectively (**Fig. 4**). The mean and standard deviation are 8.57 MJ and 2.27 for electricity consumption, respectively. Generally, consumption of electricity is better controlled compared to other sources resulting in a lesser degree of variation.

It is also evident from **Fig. 3** that the total energy consumption to produce one kg of made tea is high for factories where coal is used. This is because the coal burning air heaters are of indirect type whose overall thermal efficiencies are low (Hall).

On the other hand, F14 shows the lowest total energy consumption (23.88 MJ/kg) and also the lowest fuel energy consumption (14.9 MJ/kg). The garden uses T.D. oil as the only fuel. Oil burning air heaters are of difect type with higher overall thermal efficiency values. Moreover, oil being a relatively costly commodity, considerable control in its use showed be exercised leading to lower fuel consumption.

Fig. 7 shows the energy consumption values for factories using natural gas. It is seen that the values are on the higher side, ranging from 27.58 MJ/kg to 31.65 MJ/kg except for factory F7 where the value is only 18.64 MJ/kg. These factories have switched over from oil or coal to natural gas very recently. In most cases the old air heaters have been converted to run on natural gas, some of which are of indirect type, leading to a higher level of fuel consumption.

Contribution of Different Commercial Sources of Energy

Table 5 lists the percentages of energy contributed by different sources to produce one kg of made tea. It is observed that contribution of electrical energy varies from 10% (F1) to 37.60% (F14). On an average, the share of electricity is 20.65% of the total energy consumption, the remaining portion being contributed by other sources of heat energy.

Concusion

1. The highest amount of manual energy (18 408 MJ/ha) is required in the first year of tea cultivation and it decreases to approximately 3 653 MJ/ha from fifth year onwards.
2. An estimated amount of 2 262 MJ/ha of fuel energy is needed for land preparation only during the first year and 86.47 MJ/ha per year for green leaf transportation.
3. The total human and fuel energy requirement in the garden is about 3.1 MJ/kg in first year and 2.0 MJ/kg from second year onwards.
4. Coal, oil and natural gas are the three major sources of heat energy used in the factory apart from electricity. Of the 15 factories, 3 used both coal and oil, the remaining 12 factories used only one kind fuel.
5. The variation in the rate of electrical energy consumption (MJ/kg of made tea) is less compared to that of fuels. It indicates a possibility of exercising better control in the use of fuels.
6. The share of heat energy is substainally greater than the share of electrical energy in all the factories.
7. The total commercial energy requirement per kg of made tea is highest for factories using coal followed by gas and oil.
8. The minimum rate of energy consumption (23.88 MJ/kg) is for an oil-using factory. Among the coal using factories, the minimum value is 43.72 MJ/kg and among gas using factories, it is 27.49 MJ/kg.
9. Further investigation on the matter is necessary to find ways for reduction in the rate of commercial energy consumption.

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ABSTRACTS

The ABSTRACT 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.

439

Modelling of Pellet Feed Resistance to External Forces: Okursoy, Rasim, Instructor, Farm Machinery Dept.; Ibrahim AK, Assist. Professor, Animal Science Dept., respectively, Faculty of Agriculture, Uludag University, Bursa, Turkey.

The objective of this study was to develop an analytical model for the maximum breaking force of the pellet feeds which are used for feeding farm animals such as beef and dairy cattle, sheep, calf and fish in Turkey. The analytical model is based on a multiple regression model that relates the maximum breaking forces with pellet moisture content and density. The model data were obtained experimentally using a one-channel PCDL-770 Data Acquisition System with a PC interface. A pressure sensitive load cell was also used for sampling the analog signals that were produced by applied forces. Results of measurements showed higher breaking forces necessary for the beef and daily cattle pellet feeds than the sheep, calf and fish pellet feeds.

440

Tractor Size Selection Model for Multi-crop Farms in Pakistan: Rizwan-ul-Haq, Former Graduate student, University of Agriculture, Faisalabad; Munir Ahmad, Senior Engineer, Farm Machinery Institute, NARC, Islamabad; Shafi Sabir, Professor, Dept. of Farm Machinery and Power, University of Agriculture, Faisalabad, respectively, Pakistan.

Selection of proper size of tractor is imperative in reducing the cost of production and to increase the net profit of the growers. Various techniques have been developed, but these are not applicable in Pakistani's condition. Moreover, in Pakistan small range of tractors (33-56 kW) are available and farmers are supposed to select optimum size tractor from this range.

A computer aided model was developed to select proper size of tractor for multi-crop farms of Pakistan based on time constraint approach. It also enable the farmers to predict free time of tractor availavle for custom hiring.

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Evaluation of Groundnut Losses Under Mechanical and Manual Digging and for Mobile and Mechanical Threshing: Omar, Elnogomy A., Former Grad. Student; Omar A. Rahama, Assoc. Professor, respectively, Dept. of Agric. Eng., Univ. of Gezira P.O. Box 20 Wadmedani Sudan; Mamoun I. Dawelbeit, Senior Res. Sci., ARC, Wadmedani, Sudan.

An experiment was conducted at the Rahad Agricultural Corporation for the purpose of studying the effect of digging and threshing methods on groundnut losses. The study involved mechanical and manual digging. Mobile and stationaly threshing were also studied. The objective was to find the optimum period for digging and threshing under the local weather conditions.

Results of the experiment showed that 18% to 25% soil moisture content which occures 10 to 14 days from the last irrigation resulted in the least quantity of crop losses. The crop losses were not significantly affected by the digging method, whether mechanical or manual, in the range of moisture content specified.

Mobile threshing demonstracted the least percentage of losses (11.3%) when the net amount of crop collected in the machine bin was taken as the measure of performance efficiency. However, crop losses under stationary threshing was not entirely due to the method but rather the feeding technique by the workers which resulted in a sizable amount either left on the ground or missed on the way of the machine. Delaying groundnut threshing beyond 6 days after digging date resulted in significant losses.

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A Cone Seed Metering Device for Minimum Tillage Seed Drilling in a Rice-wheat Cropping System: Borlagdan, Paterno C., Research Assistant; Graeme Quick, Head, respectively, Agricultural Engineering Division, IRRI, P.O Box 933, 1099 Manila, Philippines.

A simple cone seed metering device for paddy and wheat was designed and developed by the Agricultur-

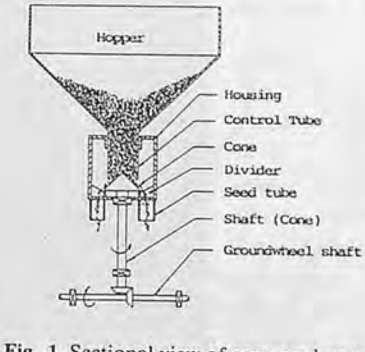

Fig. 1 Sectional view of cone seed metering device.

al Engineering Division of the International Rice Research Institute (IRRI) in Los Banos, Laguna, Phillipines.

Fig. 2 The rice-wheat minidrill prototype.

Laboratory test results showed satisfactory performance. Damage to seeds was negligible (0.09% in wheat and 0.16% in paddy). The coefficients of variation at various rotational speeds and clearances corresponding to various seeding rates were minimal.

The performance of metering device was affected by the clearance and cone rotational speed. Generally, the seed rate increased with clearance and cone rotational speed for both paddy and wheat.

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Design, Construction and Evaluation of a Wooden Multi-produce Silo for Strong Grains and Grain Legumes: Adesuyi, S.A., Senior Research Fellow, Post-harvest Technology Programme; A.M. Oguntuase, Senior Lecturer, Dept. of Civil Engineering, respectively, Federal University of Technology, Akure, Ondo State, Nigeria.

A multi-produce silo was constructed from low-cost wooden materials. It is capable of storing four different agricultural products simultaneously. It can be constructed by a local carpenter. Its useful life is about 10 years. Details of design, construction methods and choice of materials are presented. Precautions to be taken when using the silo are also outlines. A comparison of the multi-produce silo with the warehouse for storing maize and cowpeas for nine months showed that the silo was significantly better at 5% level than the warehouse in terms of safe moisture content, insect damage, viability, live insects and mouldiness.

Fig. 1 Photograph of motorized weeder prototype.

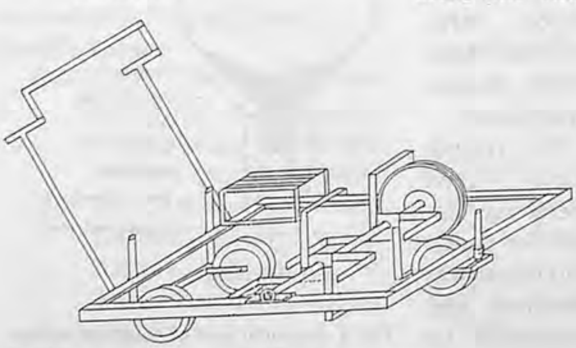

Fig. 2 Assembly drawing of motorized weeder.

463

Development of a Motorized Mechanical Weeder: Aderele, A.O., Lecturer, Samaru College of Agriculture, Ahmadu Bello University, P.M.B. 1058, Zaria, Nigeria.

The Mecanical method of weed control is accepted world wide. This method involves cultivation, bush burning, uprooting with hand, mowing and crushing, manual hoe weeding and use of mechanical weeders which may be manually operated or powered by internal combustion engines or tractors. Several machines have already been developed for weeding in Nigeria but much is yet to be achieved as most Nigerian farmers still depend on weedings hoes to control weeds on their farms. This condition has led to the development of a simple weeder.

This motorized mechanical weeder could weed up to maximum width of 550 mm and a depth of 50 mm. It is powered by a small internal combustion engine of 3.75 kW capacity. The actual average field capacity of the weeder was 0.0457 ha/h and weeding efficiency of about 80%. The only two factors considered for the performance of this weeder were depth of weeding and blade rotary speed which were found to have no statistical significant effect on its performance.

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Design and Development of a Single-row Rotary Reaper Windrower: Solanki, R.C., Technical Offcer (Agri. Engg.) W.T.C.; O.P. Singhal, Principal Scientist, Agril. Engg. Division, IARI, New Delhi-110012, India.

A single-row rotary reaper windrower was designed and developed for harvesting cereal crops like wheat, paddy and oats. It is a portable, engine-operated machine mounted on a small trolley. A front-mounted rotary serrated disc harvests the crops which are laid down on the ground in a neat row. The total weight of the machine is

Fig. 1 Single row rotary reaper windrower in operation.

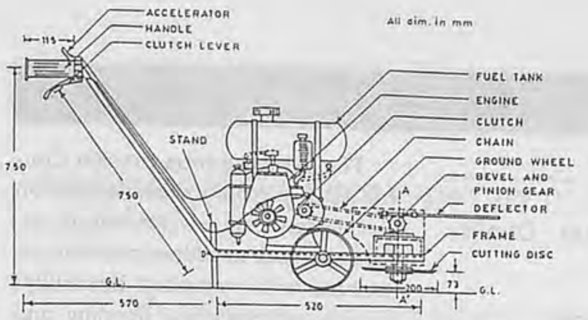

Fig. 2 Side view of a single row rotary reaper windrower.

about 30 kg. The harvesting performance of the machine has been quite encouraging.

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Design Development and Performance Evaluation of Food Pellets (Wadian) Extruder: Sharma, D.N., Professor and Head, Department of Farm Power and Machinery, College of Agricultural Engineering and Technology; Romika Sethi, M.Sc. Student, Department of Family Resource Management, College of Home Science; Sudesh Gandhi, Scientist, Department of Family Resource Management, College of Home Science, respectively, CCS Haryana Agricultural University, Hisar-125 004 (Haryana), India.

Women play an important role in the household economy. The present efforts to open up new avenues of self-employment for women will not only make them self-reliant and simultaneously but also raise their status and the family income. Food pellets (*wadian*) making is one of the important income generating activities for women. A simple prototype extruder was designed, developed and tested which is capable of producing 16 pellets at a time. The equipment utilizes locally available material and can be fabricated by local artisans. It operates on the principle of flow of semi-solid food materials through circular openings under pressure. The output of the machine is 4 to 5 times more than the traditional manual method. It will be v ery useful for running a part-time, small-scale industry for the homemakers.

Fig. 1 Food pellet extruder (Wadian maker).


Fig. 2 Extruder in operation.

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Field Performance Evaluation and Adaptability Study of IJO-BARI Seeder: Wohab, M.A., Scientific Officer, A.H. Talukder, Senior Scientific Officer, G. Moula, Senior Scientific Officer, respectively, Regional Agricultural Research Station, Jamalpur, Bangladesh; and Sultan Ahmmed, Scientific Officer; M.A. Satter, Principal Scientific Officer, respectively, Agricultural Engineering Division, BARI, Joydebpur, Gazipur, Bangladesh.

In order to test the field performance as well its adaption, the IJO-BARI seeder, an experiment was conducted at FSR site, Narikeli, Jamalpur during 1991-92. Jute and wheat seeds were sown to compare the performance of the seeder with that of broadcasting method. One person could operate the seeder with a sowing capacity of 0.12 ha/h. Seed rate for jute and wheat were 6.20 (cap.)/4.25 (Oli.) and 120 kg/ha, respectively. The time required to sow with the seeder and broadcast were 16.5 and 10.5 man-h/ha, respectively. The cost of sowing using the seeder was 122.0 Tk/ha and that of broadcast was 96.0 Tk/ha. Weeding time as well as cost of weeding in line sown jute field was about 50% less than that of broadcast field. Line sowing by seeder increased jute yield about 11% and wheat yield by, about 10%. The field performance of the seeder was quite satisfactory. ■■

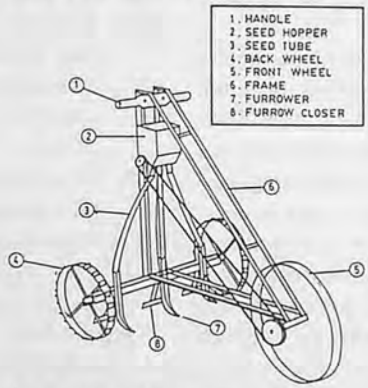

Fig. 1 IJO-BARI seeder.


Fig. 2 IJO-BARI seeder in operation.

Agricultural Equipment Technology Conference '97

February 10-12, 1997
Hyatt Regency Louisville, Louisville, Kentucky, USA

This second Agricutural Equipment Technology Conference (AETC '97), sponsored by the Power and Machinery Division of ASAE, will continue building on the highly successful first AETC held in November 1995 in Chicago. AETC '97 will bring together engineers, managers, researchers and other professionals in the agricultural equipment industry to exchange information, discuss opportunities and address challenges for production agriculture in the 21st century.

AETC '97 will focus on machinery and machinery systems for agricultural production. Invited speakers will address issues such as precision farming, turf & landscape, manufacturing, new equipment, electronics, and international business growth. A technology transfer session will feature exhibits of leading-edge products and systems and the latest in academic and applied research.

AETC '97 precedes the 32nd Annual National Farm Machinery Show on February 12 to 15, 1997. In 1996 over 280 000 people attended what has become known as America's premiere farm equipment exposition. The show features exhibitors offering a full range of agricultural products for all types of farming operations. Admission to the Farm Show is free.

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May 18-22, 1997
Manhattan, Kansas, USA

This conference will review and discuss state of the art information on the development and implementation of rapid end-use predictive technologies in the wheat marketing industry and new methods to preserve and market-based on wheat quality attributes.

Tentative Program: Each session is scheduled for a half day and co-chaired by USA and International scientists. Speakers will be invited.

I. Quality Prediction and Biochemical Determinants
II. Rapid Quality Prediction Systems
III. Breeding and Genetics of Quality
IV. New Industrial Uses of Wheat
V. Economics of End-Use Quality and International Competitiveness
VI. Preservation of Quality in Storage and Handling
VII. Prediction of Cleanliness, Soundness, Grade Determinants and Need for International Standards.

Organized by: Kansas State University, USDA/ARS Grain Marketing and Production Research Center, American Institute of Baking

Contact: Grain Industry Alliance Attn: Director's Office, Intl. Wheat Quality Conference, 200 Research Dr., Manhattan, KS66505, Fax: 913-539-5353, Inet: iwqcinfo@usgmrl.ksu.edu, Web Page: http://www.usgmrl.ksu.edu/iwqc/

Conference on Management of Landscapes Disturbed by Channel Incision: Stabilization, Rehabilitation, Restoration

May 20-22, 1997
Oxford, Mississippi, USA

The Demonstration Erosion Control (DEC) Project seeks to develop and demonstrate a watershed or systems approach to address problems associated with watershed instability: erosion, sedimentation, flooding, and environmental degradation. Initiated by the federal government in 1984, DEC demonstration activities are targeted at 15 watersheds comprising 6 000 square km within the Yazoo River Basin in the Lower Mississippi Valley with measured suspended sediment yields averaging about twice the national average. The DEC Project is conducted through cooperative efforts of several agencies and institutions. The U.S. Army Corps of Engineers (Corps), Vicksburg District; and U.S. Department of Agriculture (USDA) Natural Resources Conservation Service are responsible for planning, design and construction; while the USDA Agricultural Research Service National Sedimentation Laboratory, the Corps Waterways Experiment Station, the CCHE of the University of Mississippi, and the U.S. Geological Survey are responsible for research and monitoring.

The conference is to provide a forum for technology transfer among researchers, scholars, potential users, state engineers, environmental engineers, fisheries and wildlife personnel, general contractors, etc. in the field of upland soil erosion/control, sediment yield, channel stabilization, bank erosion, stream ecology, restoration, and environmental impact. At the conference, the latest technologies developed by scientists and engineers shall be reported to professionals interested in solving similar problems outside the DEC areas, and researchers and experts are invited to present their latest contributions to advance the state of technology.

Technical Areas: A) Hydrology and Research Methodology; B) Sediment Transport and Geomorphology; C) Hydraulics and Design, and D) Environment and Ecology.

Contact: Dr. Eddy Langendoen

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Please let us know your need. We shall promptly reply them. Inquire on any catalog listed in the advertisement in this issue. We shall try our best to serve you. We welcome articles of interest to agricultural mechanization.

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CCHE, School of Engineering, The University of Mississippi University, MS 38677. e-mail: DEC@hydra.cche.olemiss.edu, Voice: (601) 232-5083 Fax: (601) 232-7796

The Sherbrooke Conference

May 27-30, 1997
Sherbrooke, Quebec, Canada

The Canadian Society of Agricultural Engineering will hold its annual conference next year in Sherbrooke, Quebec. It will meet jointly with the Canadian Society for Civil Engineering under the theme: "Major Public Works-Key Technologies for the 21st Century."

You are invited to present a paper for the CSAE technical program. Sessions will be organized on subjects such as structures and environment, power and machinery, food engineering, energy and processing, soil and water, information and computer technologies. Other proposals thought to be of current interest will also be considered. There will be a number of joint sessions sponsored by both technical Societies.

Contact: Centre de recherche et de d'eveloppement sur les sols et les grandes cultures, Agriculture et Agroalimentaire Canada, 2560, boul. Hochelaga Sainte-Foy, Québec G1V2J3 Canada.

Elmia Wood 97—Forestry Trade Fair

June 4-7, 1997

In 1997, Elmia Wood will be 20 years old. As usual, it will be a venue on which forestry professionals from all over the world will be converging to learn from Scandinavian forestry expertise.

The Elmia fair has a number of unique features. First, there are no exhibition halls housing static showroom displays of well-polished, gleaming machines. The site is in the heart of the forest, where the machinery, vehicles and personnel will be seen at work as in the real world. Here the world's leading manufacturers will be demonstrating their innovative products.

In short, the latest and the best of what the suppliers have to offer will be seen in action.

Contact: Elmia Wood, Box 6066, S-550 06 Jönköping, Sweden. Tel. +46 36 152000; Fax. +46 36 164692.

EXPO 97—International Lawn, Garden & Power Equipment

July 26-28, 1997
Louisville, Kentucky, USA

EXPO '97 will take place from Saturday, July 26 through Monday, July 28, 1997 in Louisville, Kentucky at the Kentucky Exposition Center.

Alexandria, VA, August 20, 1996—Over 30 000 registrants signed up for the 13th annual International Lawn, Garden and Power Equipment EXPO surpassing even last year's record-breaking numbers. If that wasn't enough, a record-breaking 618 exhibitors weere on board to service the attendees — and out of this number, 196 were new to EXPO. Leading the way in increases over last year were: attendees from lawn garden centers, up by 30%; merchandise buyers, up by 16%; lawn and landscape professionals up by 11% and rental dealers up by 10%. And while the dealer/retailer category remained flat, there was a 1% increase in international visitors.

Contact: Sellers Expositions, 550 South Fourth Avenue Suite 200 Louisville, KY 40202-2504 USA. 1-800-558-8767, Tel: 502-562-1962, Fax: 502-562-1970.

3rd International Symposium on Sensors in Horticulture

August 17-21, 1997
Israel

The Workshop and Symposium on Sensors in Horticulture, which took place in The Netherlands (1991) and in Denmark (1994), were the start of a new, successful tradition. You are cordially invited to attend the "3rd International Symposium on Sensors in Horticulture". This symposium will be an opportunity to learn about current research and technology and to share new and improverd sensor applications and state-of-the-art techniques with peers from around the world.

Sponsors:

CIGR (International Commission of Agricultural Eng.)

ISHS (International Society for Horticultural Science)

IAAE (Israeli Association of Agricultural Engineering)

SR (Shanyray Technologies, Ltd.)

Topics of Papers: Papers may address a broad range of topics related to Sensors in Horticulture, for example:

- Sensor technology
- Sensors used to control/monitor the plant growth process ("Speaking plant concept")
- Climate, chemical, or physical property sensors
- Sensors for farming equipment
- Sensors used in post-harvest processes
- Sensors for quality assurance/quality control
- Sensors for non-destructive measurement
- Biosensors
- Multisensor fusion

Call for Papers: Participants interested in contributing oral or poster presentations should submit an abstract (approximately 200 words) to the symposium coordinator **before April 1, 1997**. Abstracts selected for presentation will be added to the symposium's home page.

Notification of acceptance will be sent to authors along with the ISHS guidelines for final papers. Final papers are limited to 6 pages, single spaced, including tables and figures, and must be submitted (preferably on diskette, Word Version 6 or higher), in camera-ready form, **before July 1, 1997**.

Contact: Dr. Itzhak Shmulevich, Technion - Israel Institute of Technology, Faculty of Agricultural Engineering, Technion City, Haifa 32000 Israel. Telephone: +972-4-8292626 Fax: +972-4-8221529 E-mail: agshmilo@tx.technion.ac.il

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XXVIIth International Congress on Work Science

August 25-27, 1997
Kaposvar, Hungary

Extending European Union. A fact that steadily raises questions about quality of different aspects of agricultural production such as labour, production and products. Main topics of the Congress are offering an excellent occasion to discuss all sides of these unavoidable issues.

Congress will be held in Kaposvár, Hungary at Faculty of Animal Science of Pannon Agricultural University on 25-27 August, 1997.

Scientific Programme:

1. Labour and Working Conditions
2. Integrated Quality Management
3. Work Organization and Farm Planning
4. Other CIOSTA Related Topics

Contact: Pannon Agricultural University, Faculty of Animal Science, Kaposvár, Institute of Economics and Organization. H-7400 Kaposvár, Guba S. u 40, Tel: (+36) 82 314-155 Fax: (+36) 82 320-175 email: mathe@atk.kaposvar.pate.hu.

'97 BISFPPT — 1997 Beijing International Symposium on Food Processing and Packaging Technology

October 27-31, 1997
Beijing, P.R. China

The subject matter of the symposium will center on the exchanging of food, farm and sideline product processing and packaging technology, including their new achievements, new development and new trends, going further into how to prepare ourselves in being confronted with these new development.

This symposium will be synchronized with the 1997 International Food Processing and Packaging Machinery Exhibition (CHINA FOODTECH '97). The programs of the symposium include:

to carry out the academic discussions;

to visit CHINA FOODTECH '97 and other related technological visits; and

to conduct the products and information releasing meeting (the CHINA FOODTECH '97 Forum) so to publicize our machinery and technology.

Call for Papers: Papers written for this symposium should be in line with the subject matter of this symposium. Anyone who intends to publish an article for the symposium should be at first write an abstract of less than 400 English words (1 page) and sent it to the Secretariat of '97 BISFPPT for examining by the Academic Committee of the Symposium Organizing Commission before Jan. 31 1997. The abstract should include: author's full name, address, working unit, purpose of the research, contents and the main theory from his research, conclusion and the importance of his research. If the paper has been selected for use after examining, the author of it will be notified by February 28, 1997, and the author should get the article prepared (not over 8 pages) and sent it to the Secretariat of '97 BISFPPT by May 31, 1997, along with a Pre-registration Fee of US$80/an article. If the author of the article who has paid the Pre-registration Fee, will not be able to present himself (or herself) in the symposium for some reason, he will be sure to receive a proceeding of this symposium after '97 BISFPPT.

Sponsors: Chinese Society for Agricultural Machinery Packing and Food Processing Engineering Institution, Chinese Mechanical Engineering Society

Contact: Secretariat of '97 BISFPPT Organizing Commission, No. 1 Beishatan Deshengmen Wai, Beijing 100083, P.R. China. Tel: 0086-10-62043686, 62017131-2233 Fax: 0086-10-62043686, 62017326.

ISAMA 97 — International Symposium on Agricultural Mechanization and Automation

November 17-21, 1997
National Taiwan University, Taipei, Taiwan

This is the first announcement of the International Symposium on Agricultural Mechanization and Automation in Taipei, Taiwan. This symposium is organized by the Chinese Institute of Agricultural Machinery, Taiwan, R.O.C. A platform will be provided to exchange ideas, scientific results and practical experience in the field of agricultural mechanization and automation.

Topics:

- Farm Machinery Manufacture
- Power and Machinery
- Information and Electrical Technologies
- Structure and Environment
- Food and Processing Engineering
- Waste Management
- Robotics and Control Engineering
- Systems Engineering
- Automation in Agriculture, Animal Husbandry, Fishery Production and Marketing

Call for Papers: Papers are invited on any of the listed topics and related areas. Authors are asked to submit an abstract of not more than 500 words in English by **January 31, 1997**. The abstract should stress the aim of the research and include a short description of the methods used, together with an overview of the results and conclusions. The organizing committee will select contributions for presentation. Authors will be notified of acceptance of their abstracts by March 31, 1997. Full manuscripts are due by June 30, 1997.

Contact: Professor Fu-Ming Lu, Dept. of Agricultural Machinery Engineering, National Taiwan University, 136 Chou-Shan Road, Taipei, Taiwan 106. Phone: +886 2 3637436, Fax: +886 2 3627620, E-mail: lufuming@ccms.ntu.edu.tw

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Joint International Conference on Agricultural Engineering and Technology Exhibition (Theme: Agricultural Engineering for Sustainable Development)

December 15-18, 1997
Hotel Sonar Gaon, Dhaka, Bangladesh

The objectives of this international meeting are as follows: 1) to share hand to hand experiences on the scientific and engineering developments in sustainable agriculture. 2) Establish a strong collaborative link between the professional societies and institutions of the developed and developing countries for professional development and integrity.

Call for Papers: The Organizing Committee is pleased to invite papers/posters on the following topics:

1. Power, Machinery and Farm Mechanization 2. Irrigation, Flood Control, Water Resources and Management 3. Soil and Water 4. Post-harvest Technology 5. Advances in Food Processing and Engineering 6. Fruits, Vegetables Production, Handling, Processing and Packaging 7. Physical and Chemical Properties of Bio-materials 8. Advances in Agricultural Waste Management Technologies 9. Green House Engineering and Agro-Chemical Technology 10. Electronic and Software Technology 11. Automation and Control Technology 12. Modeling with Finite Elements 13. Research Planning, Management, Evaluation and Technology Transfer 14. Sustainable Agricultural Systems and Modeling 15. Weather and Environment Technology 16. Appropriate Agricultural Technology 17. Bio, Solar, Wind, Renewable Energy and Technology 18. Poultry Housing and Environment, etc.

Anyone involve in agriculture and food (directly or indirectly) may submit a paper. Last date of submission of abstract (in English, 200-300 words): December 1, 1996. Notice of acceptance: February 29, 1997. Final paper ready: September 30, 1997.

Send abstract(s) by mail, fax or e-mail to one of the following:

1. Professor John Gerrish, 202 Agricultural Engineering Department. Michigan State University, East Lansing, MI 48824, USA. Fax: 517-353-8982, E-mail: gerrish@egr.msu.edu
2. Dr. M.A. Mazed. Director-General. Bangladesh Agricultural Research Institute, Joydevpur, Gazipur Bangladesh. Fax: 880-2-841678. E-mail: dg@bari.bdmail.net
3. Dr. Abdul Ghaly. Professor. Technological University of Nova Scotia. Po Box 1000. Halifax, Nova Scotia, Canada B3J 2X4. E-mail: aeghaly@tuns.ca

Inquiry about registration, sessions, extended date for abstract, award international program, accommodation, local transportation, air travel and visa, contact: Dr. Habib Chowdhury, Chairman (international activities). 9, Agricultural Engineering Department. Michigan State University, East Lansing, MI 49924, USA. Ph: 517-353-3883, Fax: 517-353-8982, E-mail: chowdhu2@pilot.msu.edu

Registration fee for developed countries is US$150.00; for all developing countries, US$75.00. Special package deals (Luxury class or First class) are available upon request. Luxury class package costs US$240.00 for 4 days and First class package costs US$120.00 for 4 days. These package deals include accommodation, meals, airport service, technical tours, and cultural show. Additional days can be arranged upon advance request. All international participants are required to confirm preregistration and package deal at the time of abstract submission. Bank draft drawn in US dollars, payable to **International Ag Engineering Conference 1997** and sent directly to Professor Shah M. Farouk, Chairman, Organizing Committee and Vice-Chancellor, Bangladesh Agricultural University, Mymensingh 2202, Bangladesh by April 1st, 1997.

Cooperating Countries: Australia, Belgium, Brazil, Canada, China, Egypt, Germany, Ghana, Hungary, India, Indonesia, Japan, Jordan, Kenya, Korea(s), New Zealand, Niger, Philippines, Papua New Guinea, Sri Lanka, South Africa, Sierra Leone, Turkey, Thailand, Taiwan, Tanzania, United Kingdom, USA, United Arab Emirate, Vietnam, Zimbabwe, Botswana and Bangladesh (host).

Proceedings of Tenth International Conference on Mechanization of Field Experiments (France)

IAMFE/FRANCE '96, The Tenth International Conference and Exhibition on Mechanization of Field Experiments, was organized by Institut National de la Recherche Agronomique (INRA) in cooperation with Association Française pour la Mécanisation en Expérimentation agricole (AFMEX, The French Branch of IAMFE) and IAMFE. It took place at the INRA research center Versailles, Paris on July 8-12, 1996.

The conference attracted 230 participants from 35 countries. The main theme was "Agricultural experimentation and methodological constraints relative to equipment" and the discussions dealt with how we can help securing a high level of accuracy in the field research in industrialized as well as in developing countries. There are many limitations in the technology and methods used and IAMFE's work in developing countries is very important in order to spread information about simple but reliable methods and equipment for field research as well as for arranging conferences, exhibitions and training courses in these countries.

Topics discussed at the conference were, among others: Strategies for mechanization of field experiments; Good experimental practice in field trials; Metrology and the evaluation of measured data; Methods and equipment for soil tillage, seedbed preparation, fertilizing, spraying, planting and harvesting; Experimentation in fruit and vegetable production; laboratory equipment and seed processing equipment, Agricultural pollution, Safety risks etc.

At the exhibition and field demonstrations, 43 companies displayed and demonstrated a wide range of machines and instruments for agricultural research. AFMEX has printed a list of the exhibitors. It is in the French language.

The Proceeding of IAMFE/FRANCE '96 consists of 435 pages with 61 papers of authors from 19 countries. The price is US$45 (IAMFE members - US$25) per copy, free postage included. It can be ordered from: The International IAMFE Centre, Mr. Torbjörn Leuchovius, Executive Secretary of IAMFE, PO Box 7033, S-750 07 Uppsala, Sweden. Tel: +46-18.671825, Fax: +46-18.673529, E-mail: IAMFE@LT.SLU.SE.

Questions about IAMFE and IAMFE/FRANCE '96 can be directed to The International IAMFE Centre above or to: The Norwegian IAMFE Centre, Prof. Egil Öyjord, President of IAMFE, PO Box 5065, N-1432 Aas-NLH, Norway, Tel: +47-64.928700, Fax: +47-64. 948810, E-mail: IAMFE@ITF.NLH. NO

FAO: The AGSE Service Lost the Service Chief Position

The Programme and Finance Committees of FAO have broadly approved the recommendations made by the Organization, with a view to containing costs as far as possible. This meant that the AGSE service lost the Service Chief position, its Post-harvest Technology Group was transferred to the Agro-industries Service and was consequently reduced to a Branch. The Branch now only looks after the farm power and mechanization aspects of engineering as well as post-harvest mechanization aspects. This of course does not always ensure proper technical coverage of all aspects connected with land preparation, food production and preservation at the farm level.

Wholecrop Harvester for Rear Mounting on Tractor

This harvester was developed in the UK by the Silsoe Research Institute, Patented by BTG and licensed to a Scottish company, Macantar. Although the main markets for the harvester are the Middle East and North Africa, not Europe, manufacture was recently transferred from Scotland to Cormall A/S in Denmark, and the Danish Government had invited inward missions of delegates from relevant countries to view the harvester on the Cormall stand at Agromek. Cormall A/S is a member of the ABC Hansen Group of companies.

Unlike a combine harvester, the machine collects both the grain and the straw, breaking the latter for use as animal feed. The grain is threshed, cleaned and bagged at the rear, while the straw is finely broken and discharged at the side either into large hessian bags or direct into a trailer pulled alongside.

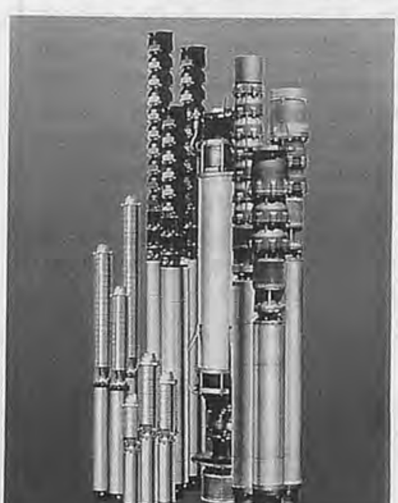
The harvester is designed for rear mounting on tractors from 40 kW with cat. II linkage and front end weights. In its working position the 2-m cutting table assembly is offset to the side of the tractor, but for transport this rotates through 180° and comes "in line" with the tractor, so the harvester can be transported within the tractor's width. In the transport position it is possible to operate the tractor in reverse, and use the table assembly for windrowing. This technique can be used for field opening. For use as a portable thresher a hand feeding hopper can easily be fitted in place of the table assembly. The capacity is up to 2 t/h of clean wheat and the density of broken straw high at up to 80 kg/m^3.

Machines have already been sold into Syria and Morocco and strong interest has been received from Turkey, Iraq and Iran. In addition, harvesters have been extensively trialled in Libya and Kenya.

Contact: Mary Clark, BTG plc. Tel: +44 0171 403 6666, Fax: +44 0171 403 7586, E-mail: BTGUK@btgplc.com Mr. Hans Bossen, ABC Hansen A/S. Tel: +45 86 42 64 88, Fax: +45 86 41 36 22.

New Energy-efficient Submersible Pumps from Ingersoll-Dresser Pump Company

Ingersoll-Dresser Pumps has launched a new generation of PLEUGER submersible pumps and motors designed for water supply and general groundwater applications. The extensive choice of specifications includes six types of NB6 pump for installation in six-inch wells with capacities up to 30 m^3/h and heads up to 400 m as well as twelve sizes of M6 motor with outputs from 5.5-37.0 kW at 50 Hz and 6.4-45.0 kW at 60 Hz.



PLEUGER submersible pump units operate below watter level and are driven by water-filled AC three-phase submersible motors, with pump and motor forming a single enclosed unit. The efficiency of the new NB6 pumps has been optimized by design modifications which have reduced the required motor size at the same time as lowering the customer's energy and investment costs. The latter are also reduced because the pumps have a higher head per stage of about 10-20% which reduces the number of stages by the same proportion. The new six-inch M6 motors feature a new lamination and bearing design providing a high quality, compact and robust product with excellent reliability and therefore reduced maintenance costs.

PLEUGER pumps have a variety of applications. Standard pumps and motors are used to pump drinking water or ground water for water supply utilities, mining, general industry and agriculture. For special applications, there are modified versions of the PLEUGER pumps, such as bottom intake, booster pumps.

The complete PLEUGER pump line offers a range of units with finely graduated performance specificaitons for capacities and heads. PLEUGER pumps are available in many different sizes of radial, semi-axial and axial design for capacities up to 80 000 m^3/h and delivery heads up to 800 m. They are built on submersible motors with outputs up to 5 000 kW for voltages up to 10 kV, to provide the user with a pump unit for almost any requirement. In addition, PLEUGER pumps are designed with modularity in mind, resulting in low costs and easy maintenance.

Contact: Jim Quain, Ingersoll-Dresser Pumps Asia Pacific, Jurong Town, PO Box 88, Singapore 9161. Tel: +65 568 6100, Fax: +65 567 5955.

Fertilizer and Chemical Injector for Crops or Turf

Carrollton TX, 16 July 1996 — The PLUS™ brand of non-electric proportioning injectors, made in Texas, feature flow rates of 30 gpm and 40 gpm, with high dependability claimed.

Used worldwide for fertigation and chemigation, PLUS injectors are easily installed in water lines either singly or in series to supply multiple chemicals.

The Noryl® casing and specially compounded seals can handle acids, fertilizers, disinfectants, fungicides, pesticides, and othedr chemicals commonly used in growing.

The PLUS injector settings can be easily changed to add from 0.2% (500:1 ratio) to 2.2% (45:1 ratio) of a chemical into a water stream.

Sold through distributors by Dosmatic. For free product literature, prices, and distributor locations Write to Paul Wilding, DOSMATIC, 1230 Crowley Circle, Carrollton, Texas, U.S. 75006, or fax 214.245.9000, or telephone (214) 245-9765.

BOOK REVIEW

Annual Research Report for 1995
(Korea)

Agriculture is a "life industry" which produces and supplies the most vital elements for human life; food. With the 21st century just around the corner, global agribusiness conditions are changing rapidly and current advanced countries are tending to arm themselves with agricultural produce. To meet effectively with these changes, it is important for us to achieve advancement in agriculture through technical development and innovation.

Based on the above-mentioned comprehensive program, the Rural Development Administration (RDA) has pursued two developmental thrusts; globalization of Korean agriculture and specialization of local agricultural industry. By enhancing our capacity for self-reliance and international competitiveness, we have strived to develop agriculture into the comprehensive bio-industry last year. To consolidate research systems to meet the needs of agricultural industry and to adapt to worldwide changes, long- and short-term research goals have been revised.

Our last year's efforts to develop state-of-the-art technological advancement through agricultural research has been paid off, and excellent results achieved. To enhance quality and multi-purpose usage of agricultural commodities, to lower production cost and to ensure safety of crop production, labor and energy saving cultural methods have been developed for field crops. Also, fundamental research concerning with environmentally sound farming and value-additional technologies has been conducted in response to the possible emergence of the Green Round. Remarkable results have been achieved from a total of 2 917 experiments. One hundred twenty three recommendations were made, including lodging-resistant, multi-disease-resistant, and super high-yielding rice cultivar "Dasanbyeo" to be applied to improve and strengthen agricultural policies, and 593 others including "newly developed culture medium for cucumbers" have been delivered to the extension services for the use as extension materials.

This annual research report is published with results gathered from the above activities. We hope that those who engaged in agriculture related fields will make full use of this information to contribute to the overall growth of agricultural industry.

Published by Rural Development Administration, Ministry of Agriculture, Forestry and Fisheries, Suwon 441-707, Republic of Korea.

Rice Research in Asia: Progress and Priorities
(Philippines)

by R.E. Evenson, Yale University, R.W. Heart, Director Agric. Sciences, The Rockefeller Foundation and M. Hossain, IRRI

This book reports efforts to use rigorous, quantitative methods for priority setting for conducting research on rice in Asia. The Rockefeller Foundation has drawn on this work to help determine the research goals it should emphasize in its funding. The book begins with a review of the problem and approaches that have been used previously. Several chapters then demonstrate how a number of areas of plant science research have contributed to gains in rice productivity and assess the current challenges of genetic improvement and pest control. The economic framework for priority setting and previous methods are reviewed, before a series of country case studies provide more practical applications of these.

432 pages, hardbound, £55.00 (US$99.00 Americas only)

Published by CAB International, Wallingford, Oxon OX10 8DE, UK.

Cultivated Vegetables of the World: **Latin Binomial, Common Names in 15 Languages, Edible Part, and Method of Preparation**
(USA)

by Stanley J. Kays, Professor of Horticulture, The University of Georgia and João C. Silva Dias, Assoc. Professor of Horticulture, Technical University of Lisbon

An essential reference manual for: libraries; scientists; embassies; agricultural organizations; international travelers; vegetable growers, packers, shippers, produce buyers, and grocery store produce managers; avid gardeners; and gourmet chefs. This authoritative reference provides the commercially cultivated vegetable crops of the world systematically arranged by division, class, family, and species; alphabetically by common name, language, plant part consumed, and method of preparation. The manual provides the common names in 15 languages of the cultivated vegetables from around the world - Arabic, Chinese (Mandarin), Danish, Dutch, English, French, German, Hindi, Japanese, Malay, Portuguese, Russian, Spanish, and Tagalog.

170 pages, softbound, $29.95

Published by Exon Press, P.O. Box 80803, Athens, Georgia 30608-0803, USA. ■■

INSTRUCTIONS TO AMA CONTRIBUTORS

The Editorial Staff of the AMA requests contributors of articles for publication to observe the following editorial policy and guidelines in order to improve communication and to facilitate the editorial process :

Criteria for Article Selection

Priority in the selection of articles for publication is given to those that —

a. are written in the English language ;
b. are relevant to the promotion of agricultural mechanization, particularly for the developing countries ;
c. have not been previously published elsewhere, or, if previously published are supported by a copyright permission ;
d. deal with practical and adoptable innovations by small farmers with a minimum of complicated formulas, theories and schematic diagrams ;
e. have a 50 to 100-word abstract, preferably preceding the main body of the article ;
f. are printed, double-spaced, under 4,000 words (approximately equivalent to 8 pages of AMA-size paper) ; and those that
g. are supported by authentic sources, reference or bibliography.
h. written on floppy disc.

Rejected/Accepted Articles

a. As a rule, articles that are not chosen for AMA publication are not returned unless the writer(s) asks for their return and are covered with adequate postage stamps. At the earliest time possible, the writer(s) is advised whether the article is rejected or accepted.
b. When an article is accepted but requires revision/modification, the details will be indicated in the return reply from the AMA Chief Editor in which case such revision/modification must be completed and returned to AMA within three months from the date of receipt from the Editorial Staff.
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a. Articles for publication (original and one-copy) must be sent to AMA through the Co-operating Editor in the country where the article originates. (Please refer to the names and addresses of Co-operating Editors in any issue of the AMA). However, in the absence of any Co-operating Editor, the article may be sent directly to the AMA Chief Editor in Tokyo.
b. Contributors of articles for the AMA for the first time are required to attach a passport-size ID photograph (black and white print preferred) to the article. The same applies to those who have contributed articles three years earlier. In either case, ID photographs taken within the last 6 months are preferred.
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Format/Style Guidance

a. Article must be sent on 3.5 inch floppy disk with MS DOS format (e.g. Word Perfect, Word for DOS, Word for Windows.....) along with one printed copy.
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f. Tables and figures must be preceded by texts or discussions. Inclusion of such tables and figures not otherwise referred to in the text/discussion must be avoided.
g. Tables must be typed clearly without vertical lines or partitions. Horizontal lines must be drawn only to contain the sub-title heads of columns and at the bottom of the table.
h. Express measurements in the metric system and crop yields in metric tons per hectare (t/ha) and smaller units in kilogram or gram (kg/plot or g/row).
i. Indicate by footnotes or legends any abbreviations or symbols used in tables or figures.
j. Convert national currencies in US dollars and use the later consistently.
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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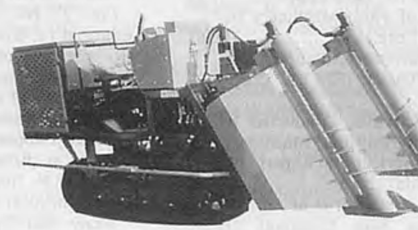

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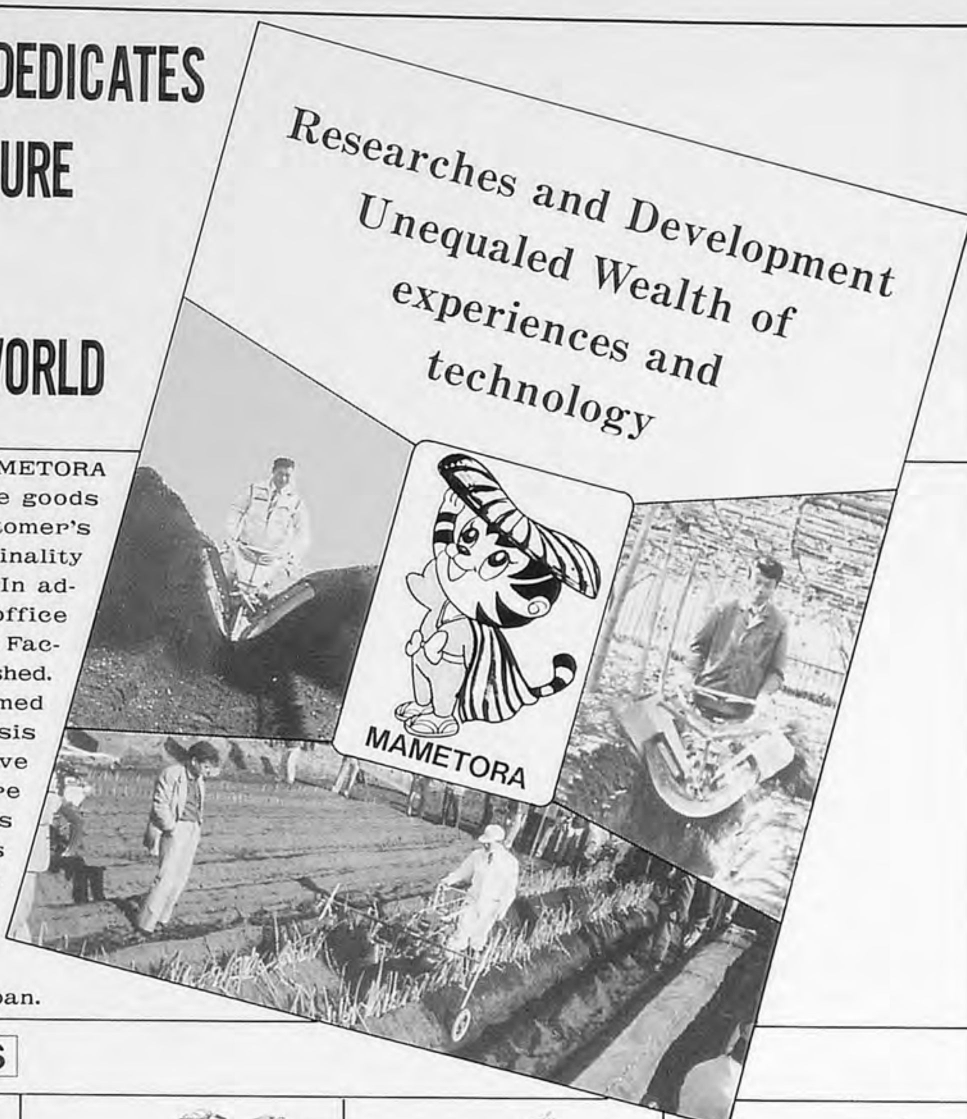

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