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AMA

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VOL.22, NO.4, AUTUMN 1991

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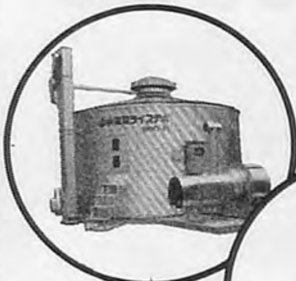
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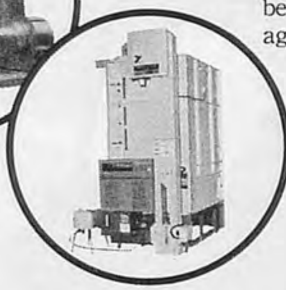
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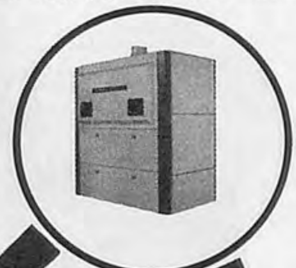
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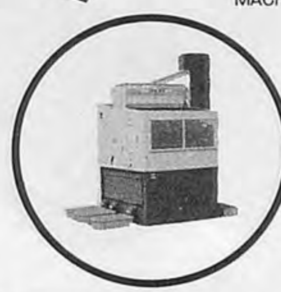
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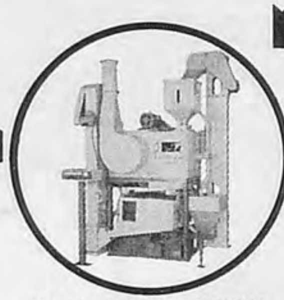
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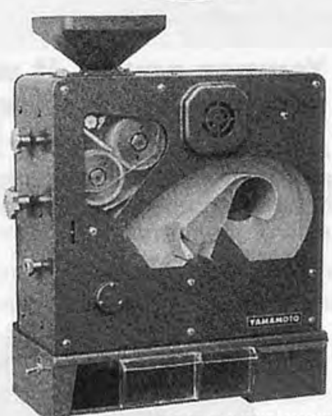
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EDITORIAL

The Paradox of Food Supply

In the July 1990 issue of AMA, we editorialized the *Race Between Population and Food Supply* in an effort to call the attention of policy makers in all countries to the widening gap between population and food supply. In this issue, we are not necessarily repeating the same editorial nor are we content that population is declining world-wide. Instead, it is our intent to voice out our concern for the same global problem in the hope that ways will be found — and soon — to narrow down the gap between population and food supply.

Ideally, available supply for all should equal population growth everywhere. In reality, however, food supply continues to lag behind while population continues at a fast pace. This case is particularly true and alarming in many countries of the Third World. Policy makers even in well-off countries like the United States, Great Britain and Canada are worried that the situation must find a lasting solution before it turns from worst to worse.

The average quantity of food supply per capita has decreased, first, in many African countries; second, in a host of Latin American countries; and by mid-1980, food supply shortage was being felt in many parts of the world. This coming winter, for example, worries the leaders in economically advanced countries that the Russians are facing an acute food shortage.

Behind this paradox of threatening food shortage is a mix of many factors — some beyond the control of man but many within his control. Those outside the control of man are natural calamities as volcanic eruptions that render vast areas of agricultural land useless as in the environs of Mt. Pinatubo in the Philippines lately; the perennial cyclone that hit Bangladesh flooding thousands of hectares of agricultural crops; and similar vagaries of nature that affect agricultural productivity adversely. On the other hand, those factors within the control of man include birth control which outstands as one that man can do much about but which he seems to ignore as more mouths come about each year to feed — without the corresponding source of food to feed them with. Others that man can do much to control if he so desires is the preservation of the ecosystem upon which depends agricultural productivity.

As many intellectuals have warned of dangers for a long time now, the effects of population increase are seen in environment destruction, air and water pollution, acid rain, rising temperature and the like. Add the fact that another phenomenon common in most developing countries is the tendency of the youth to leave farm life in favour of the lure of urban life. As a result, the aging farmers are left behind to till the farm that they once dreamed should be inherited by their sons. And this is where agricultural mechanization comes in to help maintain farm productivity. This means that farming operations that the aging farmers can do with mechanization must be mechanized. But more importantly, the young sons must be shown that farming need not be a back-breaking job for a life-time if they learn to operate farm machineries and see production making gains and prosperity.

Yoshisuke Kishida
Chief Editor

Tokyo, Japan
October, 1991

CONTENTS

AGRICULTURAL MECHANIZATION IN ASIA, AFRICA AND LATIN AMERICA

Vol. 22, No. 4, Autumn 1991

Puddling-type Floating Power Tiller for Small-scale Rice Farms

by
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Abstract

Puddling-type floating (P/F) power tillers have been developed and have gained acceptance by farmers in small waterlogged areas planted to rice in the Philippines. Existing P/F tiller designs have been modified in order to reduce problems related to poor maneuverability and control and uneven field preparation ("furrowing"). The main features of the modified design, called the 'hydrotiller', are the twin pontoons that are used for flotation in order to minimize furrowing and the angled lugs on the puddling wheel which are inclined in such a way that the desired combination of traction, puddling, and lift is attained. A 1-m wide hydrotiller has a field capacity of 1.5-2.0 ha/day and sells for roughly P8000 (US$400), excluding the engine. The hydrotiller is now being fabricated by 25 manufacturers in the Philippines. Farmer acceptance is growing because the hydrotiller is considerably faster and more economical to use in waterlogged areas and even in normal field conditions than conventional alternatives (e.g., water buffalo and two-wheel hand tractor).

Introduction

The idea of using a floating primemover to work in waterlogged areas and deep mud has been assessed at least 50 years ago in the U.S. and by 1962 in China (Wang). However, recently Filipino innovators produced an interesting variation of this concept. Rice farmers in the Philippines generally use the water buffalo (carabao) or hand tractor (power tiller) for tilling their ricefields (Calilung and Stickney 1987). Neither alternative is entirely suitable for waterlogged fields having deep mud and/or water. Filipino scientists (Villaruz 1986) developed a puddling-type floating tiller, consisting of a flotation chamber on which the engine is mounted and a front-mounted powered cagewheel **(Fig. 1)**.

Acceptance of P/F tillers has grown rapidly during the past 10 years — there are now more than 25 manufacturers having a combined sales of over 3000 units per year. Sales are concentrated primarily on the islands of Panay and Mindanao.

Calilung and Stickney (1987) compared different small-farm tillage equipment under field conditions ranging from normal to waterlogged. The results indicate that even under normal field conditions (10-20 cm soil depth), the P/F tiller is faster and more economical than either the water buffalo or the conventional hand tractor. Thus the P/F tiller has been encouraged for use under normal field conditions. But there are even greater advantages in waterlogged fields.

The P/F tiller is suitable for both primary and secondary tillage, when the field has been soaked for at least half a day to soften the soil. The water buffalo and conventional hand tractor also require soaking the field before primary tillage.

Puddling-type Floating Tiller

More than 25 small- and medium-scale manufacturers in the Philippines are known to be fabricating P/F tillers, in one form or another **(Fig. 1)**. The basic components they have in common are the puddling wheel, flotation structure, chain and sprocket transmission, and the engine.

The puddling wheel generally consists of a single or pair of cagewheels mounted on a common shaft, supported and powered by the transmission **(Fig. 1)**. The puddling wheel is usually 1 m wide and diameter around 35 cm. The wheel lugs are usually mounted

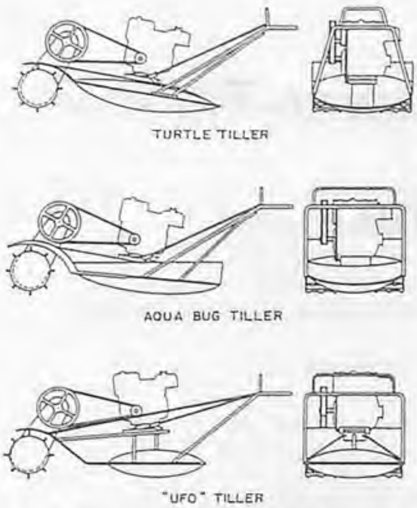

Fig. 1 Puddling-type floating tillers manufactured in the Philippines.

radially. The lugs have triangular-shaped protrusions along the outer edge which provide a more aggressive chopping and tillage action.

The hull or flotation structure varies widely among the numerous P/F designs. The 'turtle tiller' (**Fig. 1**) has an elliptically-shaped, hermetically-sealed hull whereas the 'aqua-bug tiller' has an open top (or 'barge-like') hull. The 'UFO tiller' has a symmetrical float body with the appearance of a 'flying saucer'. The UFO float uses a vertical drive shaft and bearing so it may rotate when the tiller is turned during field operation. These flotation structures provide some buoyancy for the tiller, while serving as a pivot for the handle to adjust the vertical penetration of the puddling wheel and thereby control depth of tillage and forward speed.

The chain and sprocket transmission generally provide a speed reduction of around 3:1. The design is similar to the transmission of many conventional hand tractors, except that it has only a single stage reduction, rather than two stages. The P/F tiller requires a higher cagewheel rotation speed of roughly 250 rpm compared with 60 rpm for a hand tractor.

P/F tillers generally utilize an 8-12 hp diesel or a 10-16 hp gasoline engine. Pulleys and V-belts are used to obtain a speed reduction of roughly 4:1 between the engine and the transmission.

The P/F tiller has these main advantages over the conventional hand tractor or water buffalo (Calilung and Stickney 1987):
- Suitable for waterlogged as well as normal field conditions.
- High field capacity.
- Few passes are required.
- Cost per hectare is low.
- Ability to till edges and corners of fields (a hand tractor with a moldboard plow cannot reach the edge or corners).

On the other hand, field studies of P/F tillers indicate these disadvantages:
- Considerable effort is required to maneuver the tiller and control its forward speed (e.g., in soft soils the tiller must be pushed to avoid bogging down while in firmer soils the operator has to fight to hold the machine back to ensure adequate tillage action).
- The rounded bottom of the float displaces plastic soils, creating wide furrows which need subsequent harrowing and leveling operations and thereby increase time, effort, and costs.
- Unlike the conventional self-propelling hand tractor, the P/F tiller is difficult to transport to the field and cannot be used for subsidiary operations such as harrowing or transportation.

Improving the P/F Tiller

The IRRI-Engineering Department's objective after studying floating tiller concepts was to develop a design that provides better maneuverability and control than existing P/F tillers, while minimizing undesirable furrowing effects. The new design had to be comparable to existing P/F tillers with respect to weight, width, field capacity, and price.

After experimenting with different designs for flotation structure and puddling cagewheel, the hydrotiller design was finalized (**Fig. 2** and **3**). The principal change is the use of twin pontoons for flotation. Pontoons were chosen because they almost totally eliminate the furrowing problem — the narrow width and center tunnel produce less displacement of puddled soil than the bulbous, single flotation device on existing P/F tillers.

The bottom of the pontoon is nearly flat, reducing bearing load and tillage depth by distributing the tiller's weight over a substantial area. There is a slight curvature of the bottom in the longitudinal direction (see side view in **Fig. 2**) because: i) a flat bottom results in excessive drag; and ii) the curvature provides a fulcrum or 'rocking-point' across which the operator can adjust the depth of the puddling cagewheel by exerting a vertical force on the handle.

The angle of "attack" of the pontoon must be small relative to the soil surface in order to reduce excessive drag that results from the accumulation of soil, stubbles and weeds. Furthermore, the shape (V-shaped like a boat's bow) enables it to reduce drag. Ample space is provided between the pontoon and the cagewheel to prevent the accumulation of soil and debris. The pontoon's sides are slightly curved in order to achieve streamlining and facilitate turning.

Although pontoons are not adequate to effect complete flotation of the hydrotiller, they provide sufficient buoyancy and a "hydroplanting effect" to ensure satisfactory operation in waist-deep mud (**Fig. 4**). As with existing P/F tillers, the front section of the hydrotiller becomes partially submerged if the unit stops in deep water.

The angle of the lugs of the

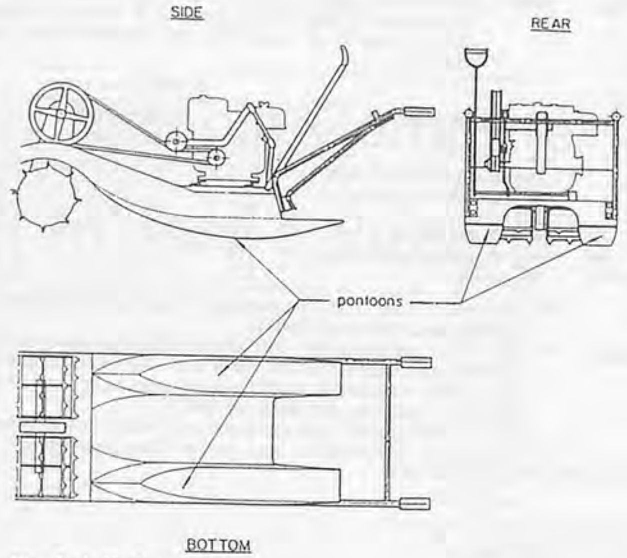

Fig. 2 A Hydro tiller.


Fig. 3 Hydro tiller in operation under normal field conditions.


Fig. 4 Hydro tiller used in waterlogged field conditions.

puddling cagewheel (10 degrees forward of the radial direction) provides lift or 'dynamic flotation' to the front section. Existing P/F tillers generally use a lug angle of 0 degree (i.e., mounted radially); conventional hand tractors have lug angles of 30-45 deg. to provide some "flotation" or lift. The 10-degree angle results in more slippage than 0 degree, thereby increasing the puddling action of the wheel.

The pontoon and lug angle shapes were experimented with until the desired balance of traction and drag, as well as lift and puddling was attained. The final design permits easy operation under conditions ranging from moderate to firm soil to extremely plastic and deep mud.

The transmission, clutch mechanism, and engine requirements are similar to existing P/F tillers. The basic specifications of the present model are:

Width:	1.0 m
Length:	1.9 m
Weight (less engine):	105 kg

Power transmission:
1) Double V-belts between engine and transmission
2) Totally enclosed chain-and-sprocket transmission (3:1 reduction)

Engine power desirable:
7-9 diesel or 10 hp gasoline (Note: engine model that weighs less than 50 kgs is desirable).

Design features such as power and weight of available engines may be adjusted to meet specific requirements of farmer-users. The hydrotiller was designed for easy manufacture by small-scale shops using common tools, materials, and readily available components such as bearing seals, fasteners, V-belts, pulleys, sprockets, and chains.

Detailed drawings are available from the International Rice Research Institute or the Department of Agriculture*.

Technical and Economic Evaluation

Extensive on-farm tests showed that the hydrotiller is easier to maneuver and control than existing P/F tiller models and conventional hand tractors. Moreover, farmers point out the exceptionally high quality of puddling and uniformity of field preparation obtained in a single pass, thereby reducing the number and intensity of subsequent puddling, harrowing, and leveling operations (Ebron 1985). For most field conditions, many farmers confirm that the hydrotiller eliminates the need for any subsequent puddling or harrowing. The IRRI Experimental Farm found that both harrowing and leveling operations can be omitted if two passes of the hydrotiller are used in 1-2 cm standing water in the field with the subsequent pass made at right angle to the previous one.

The hydrotiller also operates satisfactorily in fields with tall stubbles remaining from the rice crop, as well as in weedy fields. The IRRI Farming Systems Department found that the tiller is highly effective in incorporating a green manure crop such as *Sesbania rostrata*, even with standing crop heights of up to 2 m (Ventura et al, 1987).

Table 1 presents results on the field capacity and fuel consumption of the hydrotiller compared with other tillage technologies commonly used in the Philippines (Calilung and Stickney, 1987). The

*Interested firms within the Philippines may obtain technical drawings and assistance by contacting the Engineering Department, Bureau of Plant Industry (Department of Agricultural Engineering) at San Andres St., Malate, Metro Manila.
Firms outside the Philippines may obtain drawings by contacting the Department of Agricultural Engineering, International Rice Research Institute, P.O. Box 933, Manila.

Table 1 Summary of Test Data from First-pass (Primary) Tillage Using Alternative Land Preparation Technologies.[a]

Technology[b]	Soil depth 10-20 cm		Soil depth > 30 cm	
	h/ha	l/ha[c]	h/ha	l/ha[c]
Buffalo/moldboard plow	24.0	0	N.A.[d]	N.A.[d]
HT/moldboard plow	9.3	18	12	29
HT/disk plow	6.6	15	N.A.[e]	N.A.[e]
HT/spiral plow	6.6	15	N.A.[e]	N.A.[e]
HT/cagewheel + harrow	6.6	15	9	23
Hydrotiller	4.2	13	3.7	12

a) Hydrotiller data are based on the present study, other data are from Calilung and Stickney (1987).
b) HT = hand tractor (conventional two-wheel power tiller).
c) Fuel consumption in l/ha for 10 hp gasoline engine.
d) Data for buffalo/moldboard plow not available for soil depths > 30 cm.
e) Equipment not applicable for soil depth > 30 cm.

Table 2 Benefit-cost Analysis of Complete Land Preparation Using Alternative Tillage Technologies at 3 Annual Utilization (25, 50 and 75 d/yr) Levels and Normal Field Conditions.[a]

Technology	Benefit-cost ratio		
	Annual use		
	25 d/yr	50 d/yr	75 d/yr
Buffalo/moldboard plow[b]	0.5,0.3,0.2	0.8,0.4,0.3	1.0,0.6,0.4
HT/moldboard plow[c]	0.8	1.1	1.3
HT/disk plow	0.9	1.2	1.4
HT/spiral plow	0.9	1.2	1.4
HT/cagewheel and harrow	1.0	1.3	1.5
Hydrotiller[d]	1.5	2.1	2.4

a) Hydrotiller data from the present study, other data are from Calilung and Stickney (1987).
b) The three estimates for buffalo/moldboard plow based on different assumptions (10%, 50% and 100%) for opportunity cost of labor required for buffalo maintenance (i.e., watering and grazing). (Calilung and Stickney 1987).
c) HT = hand tractor (conventional two wheel power tiller).
d) Two passes by hydrotiller and one levelling pass using a water buffalo.

primary tillage data were gained on fields having both normal (10-20 cm) and extreme (> 30 cm) soil depths (Calilung and Stickney, 1987). The results show that the field capacity of the hydrotiller is substantially higher than those of traditional technologies, especially in deeper soils where buffalo and hand tractors are not operable. The most important finding was the hydrotiller's creditable performance in normal soil depths. This showed that the machine was not only suitable for waterlogged areas but for normal fields as well.

A benefit-cost comparison of alternative land preparation technologies is presented in **Table 2.** The data and computations of Calilung and Stickney (1987) were utilized — benefits are the current charges (income) levied by equipment owners for land preparation while costs include capital investment, interest, fuel, labor, and other expenses associated with each type of equipment. A technology is profitable only when the benefit-cost ratio is greater than 1. The benefit-cost ratio for using the hydrotiller is more than 50% higher than corresponding values for other technologies **(Table 2).**

Concluding Remarks

The present study has shown that the hydrotiller is technically and economically superior to existing small-farm tillage equipment in the Philippines. Moreover, it is suitable over a wider range of soil and water conditions and is not limited to waterlogged areas. The design overcomes problems of maneuverability, control, and furrowing associated with existing P/F tiller models, while maintaining the same cost and field performance levels.

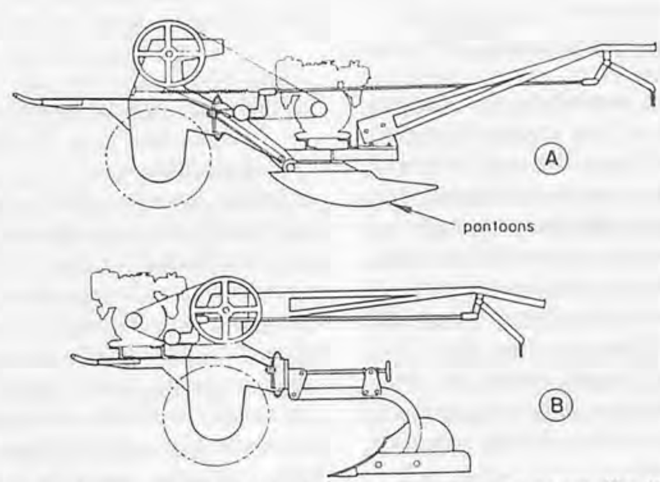

Fig. 5 Convertible tiller (A) is obtained from traditional power tiller (B) by adding pontoons and moving engine and handlebar to rearward positions.

Through a collaborative effort with the Philippine Department of Agriculture, the hydrotiller is now being introeduced to farmers and manufacturers. Eight firms have begun commercial production in Mindanao, Panay and Luzon.

The IRRI Engineering Department has initiated the distribution of the hydrotiller design in appropriate areas in Asia, Africa, and Latin America. It will prove particularly attractive in waterlogged areas in Africa which cannot be tilled by conventional equipment.

Efforts are continuing to overcome the problem of poor transportability, maneuverability and limited versatility of P/F tillers and to further improve the hydrotiller. A prototype design is now under preliminary field testing (the convert-a-tiller model) that readily converts a conventional hand tractor into a model embodying many advantages of the hydrotiller. The conversion is accomplished by adding the pontoon assembly and relocating the engine and steering handles **(Fig. 5).** It is not yet clear, however, if farmers will accept a dual-purpose machine requiring a conversion operation.

(Continued on page 17)

Portable Computer and Data Logger to Test a Prototype in the Fields

by
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Abstract

A portable field computer "Epson HX-20" and a data logger/meter, "Squirrel SQ-8-4V" were selected and used successfully to test a hand-operated grain crops harvesting machine on Bangladesh farms.

Brief description of the machine, portable computer and data logger/meter are given. Details of instrumentation outline and the circuit diagram of the power supply box are also provided.

Field experiment techniques are described and a sample output during wheat harvesting is presented.

Introduction

Instrumentation and selection of equipment to test prototype agricultural machines in the field is always a difficult task. The selection of equipment for simulated field tests in the laboratory does not face such difficulties as the size and weight of the equipment is often not as great. Also, in the laboratory tests, delicate equipment can be used easily where the safety of the equipment is not at high risk against adverse weather, vibration and field operations.

But researchers who want to test their machinery in the field have to consider the factors concerning the safety, portability, compactness, robustness, acceptable accuracy, power supply, etc. of the equipment used in the tests.

Considering all these factors "Epson Hx-20" portable computer (Epson, 1985), and "Squirrel SQ-8-4V" data logger/meter (Grant Instruments, 1985) were selected and used successfully to test a hand-operated grain crops harvester (Mollah, 1988) on Bangladesh farms.

Materials and Method

In 1986 a hand-operated prototype grain crops harvester (Mollah, 1988) was developed at Silsoe College, U.K. to harvest rice and wheat in Bangladesh.

The Machine

The line diagram of the machine is shown in **Fig. 1**. This machine was designed to cut grain crops like rice, wheat, etc. at the base and windrow the cut crops on the ground in a swath or bunch them in a collection box. The cutting part of the machine consisted of two counter rotating discs overlapping each other. One of the discs was powered from the ground drive and the other was freely rotating. The machine cuts one row of crop in a single run or pass and windrow only to its right.

Instrumentation Outline

In order to measure the test parameters of the pushing force, torque to the disc, and disc separating force, strain gauged beams were designed and mounted onto the machine. The gauge bridges were powered from a specially designed portable power supply box.

The voltage outputs from the bridges were measured and recorded by the digital data logger/meter "Squirrel SQ-8-4V" (Grant Instruments, 1985). The data logger was built specially to record a sample rate of 10 readings per second. It could record voltages from 0 to 10mV. The data recorded into the data logger was then transfered to the portable computer, "Epson Hx-20" (Epson, 1985) for analysis and storage. **Fig. 2** shows the computer, data logger and the power supply box connected to each other. The computer was

Acknowledgement: I acknowledge the valuable guidance and help of my PhD supervisor, Mr. C.D. Watt, Silsoe Campus, Silsoe, Bedford, U.K. in selecting portable computer and data logger for field experiments.

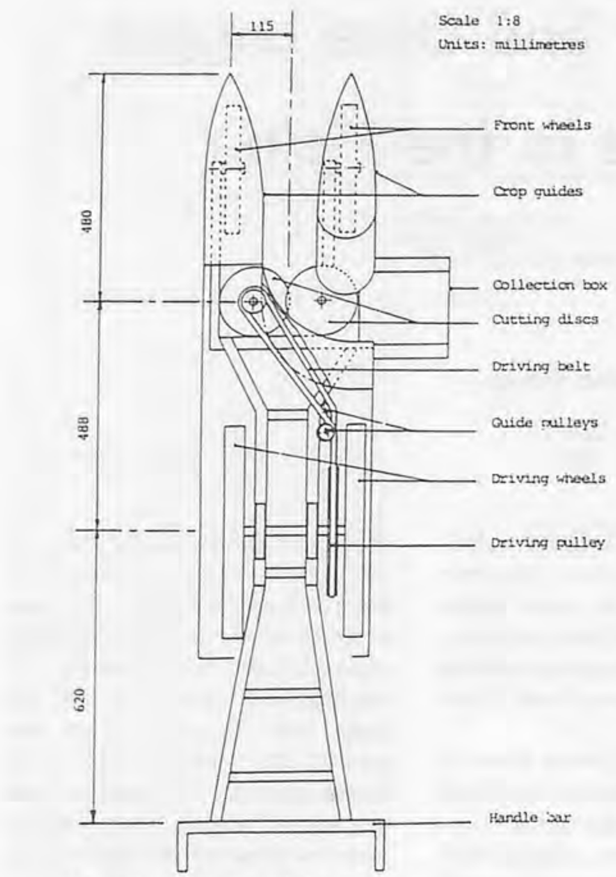

Fig. 1 Line diagram of the machine.

Fig. 2 The power supply box, data logger and the portable computer.

Fig. 3 Power supply box and data logger connected to early version of the machine.

portable and the power supply box could be carried on the shoulder by means of a strap while holding the data logger in the hands (**Fig. 3**). The person carrying the instruments connected to the gauge bridges on the machine can walk along side the machine during field trials.

Strain Gauges

Self-temperature compensation "student" gauges having a gauge factor of 2.045 (Code No. 913321) were used to measure the test parameters (Micro-Measurements Division, 1985).

Fatigue life of the gauges were 10^8 cycles at $\pm 1200_{\mu\epsilon}$. The normal use temperature for static strain measurement is $-100°$ to 350°F. Strain limits are 3% to 5% for 3 mm and 6 mm gauge lengths, respectively, for single cycle use.

Power Supply Box

The power supply box contained a 12V RS dryfit A200 Lead Acid rechargeable battery (RS Components Ltd, 1985), a regulator, 20 turn 10 ohms preset potentiometer, 5 pin DIN sockets on extension leads to connect the strain gauge bridges, leads to connect the squirrel data logger, points to connect the recharger and an on/off switch.

The regulator was used to reduce the battery voltage from 12V to 5V to excite the gauge bridge. A preset potentiometer was used to set zero before any measurement. **Fig. 4** shows the circuit diagram of the power box containing the regulator and the preset potentiometer. **Fig. 2** shows the power supply box itself with the data logger and the computer.

Squirrel Data Logger/Meter

The Squirrel was a combined meter and data logger with four channels (Grant Instruments, 1985). Used as a meter, the unit provides a continuous indication which is updated every second. As a logger, the unit takes readings at preset intervals between 1/10 second and 100 minutes and was set at an interval of 1/10th of a second. Up to 1900 readings can be stored. The meter and logging functions can be used at the same time or independently.

The "Squirrel" is powered by a 9V alkaline battery (Duracell MN1604) contained in a compartment underneath. The unit gives an indication of the remaining battery life (in days, for a complete recording cycle).

The unit is controlled by a microprocessor which is programmed to perform all the required measurements, display and storage operation (Grant Instruments, 1985).

An interface socket is fitted which enables stored readings to be transfered to a separate computer for display and storage.

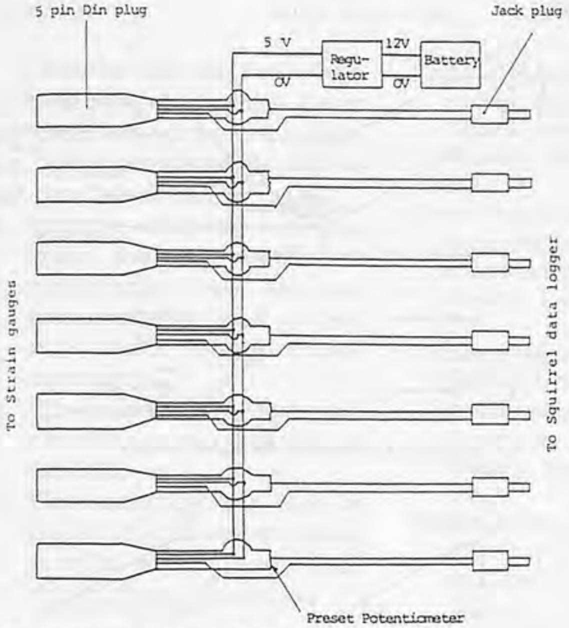

Fig. 4 Circuit diagram of the power supply box.

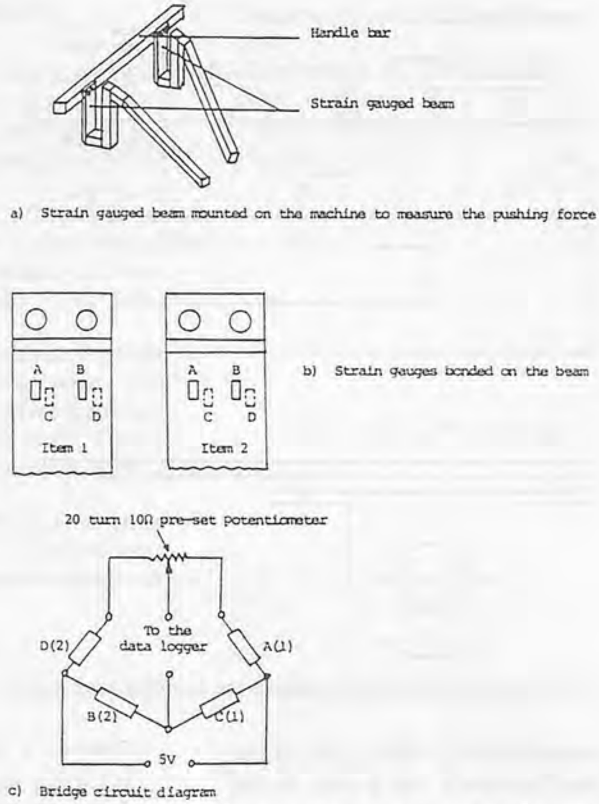

Fig. 5 The bridge circuit diagram for the handle bar strain gauged beams.

Epson HX20 Portable Computer

"Epson HX20" is a filed portable computer installed with software to transfer data from the "Squirrel", perform analysis and print results by a built-in printer. The computer is also able to copy a data file to a micro-cassette by a built-in micro-cassette recorder or transfer a data file to another computer for analysis.

There is a built-in rechargeable battery which can be used for 50 hours of average use for a full charge.

The liquid crystal display screeen displays 20 characters by 4 lines at any one time, but it would automatically scroll horizontally to show more characters.

Instrumentation of the Machine

Three different types of strain gauged beam were designed and mounted onto the machine to measure the test parameters.

To Measure Pushing Force

Two strain gauged beams (143mm × 26mm × 4mm each) were used at the handle bar to measure the pushing force. Two gauges on each beam were connected to a single bridge **(Fig. 5)**.

To Measure Disc Separating Force

The overall length, position of the strain gauge and centre of the disc bearing (shaft) is shown in **Fig. 6**. The overall length of the beam was restricted by the framework of the machine. The beam was assumed to have uniform cross section along its length and the stiffening effect of the bearing bolted to the beam was ignored.

To Measure Disc Torque

A special strain gauged beam fitted with a guide pulley was used to detect the vertical force exerted by the belt on the pulley and then the belt tension was calculated from the geometry of the belt and pulley to determine the disc torque **(Fig. 7)**.

All the gauge were wired and connected to a 5-pin socket on extension leads for easy handling. The strain gauges on the beam were provided with an extra coating against any mechanical damage.

Field Experiments

All the strain gauged beams were calibrated against known dead loads before being used in the fields, using a data logger for recording readings. The values of "r" squared for all calibraiton graphs were above 0.98.

Data Recording

The voltage outputs from four strain gauged beams (handle bar, cutting disc, belt tension-tight side and belt tention-slack side strain

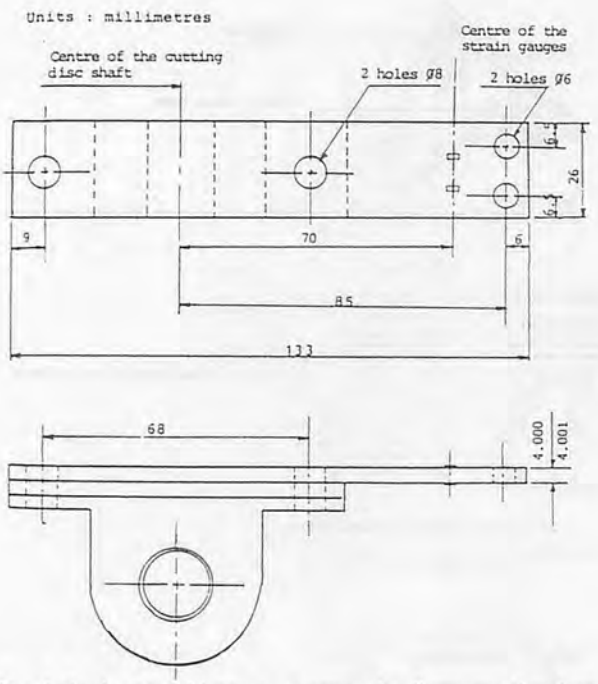

Fig. 6 Strain gauged beam to measure the disc separation force.

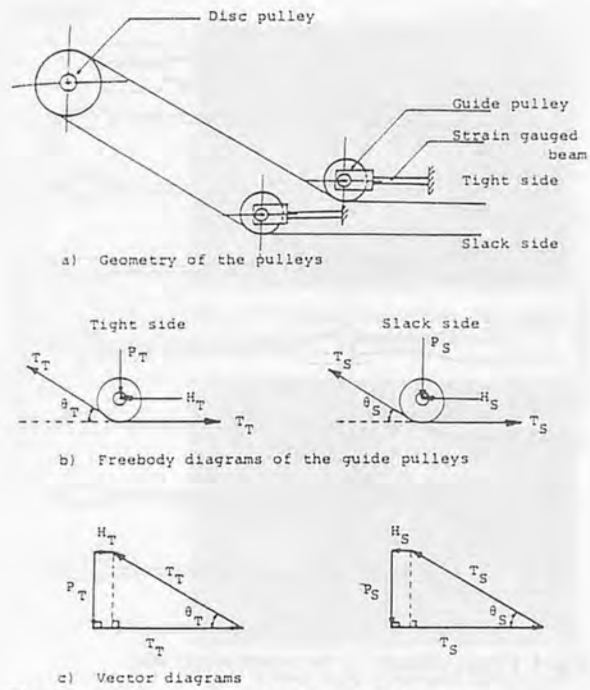

Fig. 7 Arrangements to measure the belt tensions.

gauged beam) could be recorded simultaneously for a run as the data logger had four voltage input channels.

The voltage output from the bridge circuits were zeroed by a preset potentiometer before putting the belt on. For each run the data logger was used to record the voltage outputs (10 readings per second) after the machine started to harvest and was stopped before the machine stopped harvesting. This technique was used to avoid the transient readings caused by the acceleration and retardation of the machine.

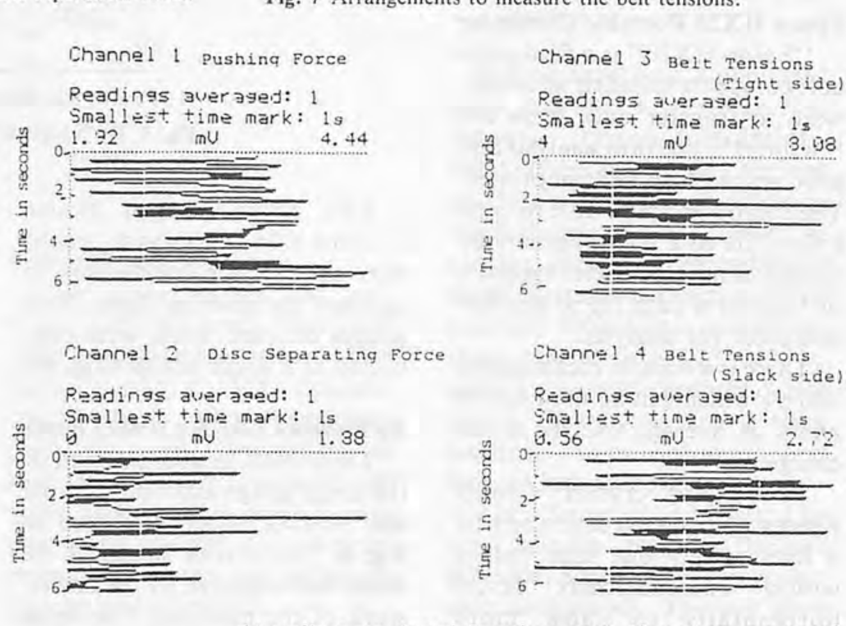

Fig. 8 The sample outputs during wheat harvesting.

Data Analysis and Discussion

The sample output during wheat harvesting experiments are shown in **Fig. 8**.

The computer Data Analysis Menu has options of printing, plotting, providing mean/standard deviation, max/min and threshold values of data. The program is able to print the data with exact time of occurrence if the right date and time is set before using the data logger. It also enables the researcher to see the number of minimum and maximum readings with time of occurrences which helps to detect the cause should anything go wrong.

The mean and standard deviation of the data can also be obtained on-the-spot before being used on the main frame computer. A desirable threshold value can be set to view the readings greater or lower than that value.

The readings can be plotted automatically by built-in printer, either on expanded or non-expanded scale. **Fig. 8** shows plotting on expanded scale.

Conclusions

From the experience of using a portable computer and data logger on wheat and rice harvesting during field experiments, the following conclusions can be made:

a. Portable computer and data logger are very useful and sophisticated tools that can successfully be used in field experiments of agricultural machinery.
b. This equipment is compact and portable for easy handling in the field.
c. This equipment can also be used in adverse weather conditions like rains and gusty winds with minimum protection.

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(Continued from page 12)

Puddling-type Floating Power Tiller for Small-scale Rice Farms

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Energy Expenditure Pattern of Power Tiller Operator

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Introduction

The ability of an indivisual to operate the power tillers depends on the physical structure of his body skeletal muscles, nervous system and metabolic process. In the case of skeletal muscle system, the bundle of muscle fibre serves to convert chemical energy into mechanical work. The neutral control of muscle metabolism is the collective chemical processes of the conversion of food stuff into mainly mechanical work and heat. The mechanical work is reflected by respiration and digestion. The contraction of muscle requires energy, the basic source of which is glycogen that gets converted into lactic acid. If the activity of the operator is continued, the body needs replenishment of nutrition from blood along with a supply of oxygen. If the level of activity requires more oxygen than it is produced by the normal rate of blood flow, the system itself adjusts to fulfill the increased demand either by increasing the breath rate or increased heart beat. So the basal metabolic rate, heart beat (Cureton[1]) and oxygen consumption are the pertinent parameters for assessing human energy required for operating the power tillers. The present study is undertaken to estimate the energy expenditure required for a power tiller operator while operating the power tiller under different conditions.

Methodology

The methodology of the study comprises of selection of subjects for operating the power tillers, their standardization and calibration and their evaluation while performing various operations with selected power tillers.

Three subjects were selected based on their anthoropometry, physical strength and other standard medical investigations. The medical investigations consist of ECG, blood pressure and pulse rate, bio-chemical parameters and blood analysis and these subjects were screened for normal health. These subjects were standardized and calibrated in the tread mill **(Fig. 1)** (Leblanc[2] and Clarke[3]) for their oxygen consumption and heart beat relationship. The calibration was to compute the oxygen consumption in terms of the heart beat as measured on the operators while operating in the field (Brockway[4] and Durnin[5]). The basal metabolic rate was also measured by using the Benedict Roth apparatus for calculating the actual energy expended by the subjects for any particular operation **(Fig. 2)**.

Having selected and calibrated the subjects, three makes of power tillers, namely; Mitsubishi (A), Kubota (B) and National (C) were selected for the study based on their weight distribution, power and characteristic dimension.

The selected power tillers are operated in red and black cotton soils with recommended higher and lower tyre inflation pressures for dry ploughing with mould board plough and puddling with rotary cultivator. All the other parameters viz., depth of operation, moisture content, soil conditions were controlled to be constant throughout the study.

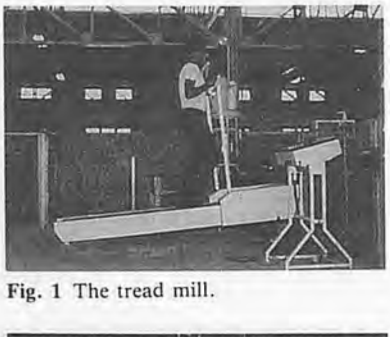

Fig. 1 The tread mill.

Fig. 2 Benedict Roth apparatus.

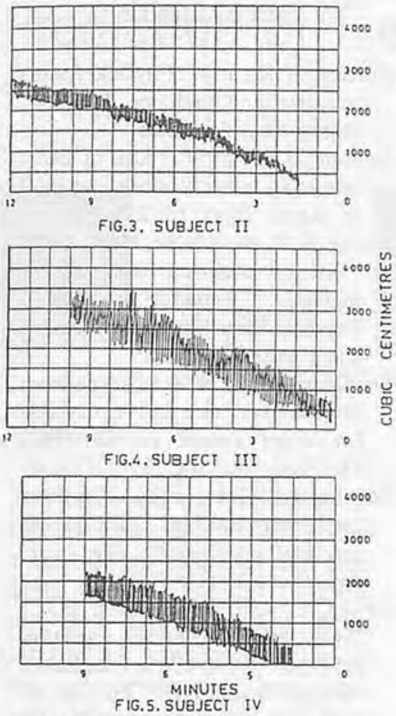

Fig. 3-5 Basal metabolic rate.

The subject operator was monitored for his heart beat at constant time interval, until they reached their fatigue level or stabilization (Davies[8]). The recorded basal metabolic rate of the three subjects are shown in **Figs. 3** to **5**. The recorded oxygen consumption for the three subjects while working on tread mill is given in **Figs. 6** to **8**.

Results and Discussion

Energy requirements for different operations with power tillers under different inflation pressure and soil conditions are presented in **Table 1**. These values were obtained after deducting the basal metabolic rate from the value of oxygen consumed in the field. Energy expenditure varied from a minimum of 1.43 Kcal/min to 6.94 Kcal/min. There is considerable variation in energy expenditure rate with respect to power tiller, inflation pressure, type of soil and subject.

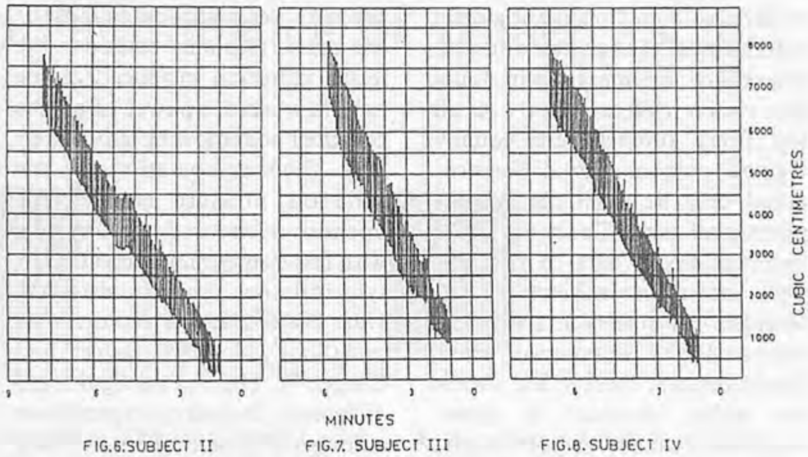

Fig. 6-8 Oxygen consumption of subjects while working in tread mill.

Table 1 Energy Expenditure of Subjects

Item	Energy expenditure in kcal/min		
	Subject II	Sugject III	Subject IV
A. Black cotton soil			
Power tiller A			
i) Tire pressure 1.8 ksc	4.99	3.37	5.74
ii) Tire pressure 1.4 ksc	4.99	3.08	5.74
iii) Tire pressure 1.0 ksc	4.99	2.54	5.74
Power tiller B			
i) Tire pressure 1.7 ksc	4.99	6.94	5.74
ii) Tire pressure 1.3 ksc	4.99	6.94	5.74
ii) Tire pressure 0.9 ksc	4.99	4.48	3.74
Power tiller C			
i) Tire pressure 1.8 ksc	2.86	3.66	2.99
ii) Tire pressure 1.4 ksc	2.67	3.66	3.91
iii) Tire pressure 1.0 ksc	2.86	3.66	3.47
B. Red soil			
Power tiller A			
i) Tire pressure 1.8 ksc	4.99	6.94	2.56
ii) Tire pressure 1.4 ksc	4.99	3.07	2.79
iii) Tire pressure 1.0 ksc	2.48	3.90	2.99
Power tiller B			
i) Tire pressure 1.7 ksc	4.99	3.90	5.74
ii) Tire pressure 1.3 ksc	4.99	3.37	4.63
ii) Tire pressure 0.9 ksc	3.93	2.54	4.17
Power tiller C			
i) Tire pressure 1.8 ksc	4.99	1.43	3.91
ii) Tire pressure 1.4 ksc	4.99	1.82	2.80
iii) Tire pressure 1.0 ksc	4.99	1.43	3.23
Pudding — Black cotton soil			
i) Power tiller A	3.35	3.08	5.74
ii) Power tiller B	3.09	5.30	5.74
iii) Power tiller C	4.99	3.37	5.74
Puddling — Red soil			
i) Power tiller A	3.47	4.19	5.74
ii) Power tiller B	4.99	5.00	5.74
iii) Power tiller C	4.99	4.19	5.74

Analyzing the energy expenditure rate for different subjects, there is significant difference in energy expenditure in most of the operations. The general trend shows less energy expenditure by the subject III and greater energy expenditure by the subjects II and IV for dry ploughing operation. Theoretically speaking, the energy expenditure rate for a particular operation should depend on the energy requirement of that particular operation and hence should be constant. The difference in energy expenditure rate indicates that the energy expended by different subjects for carrying out the same operation are different. This can only be justified by analyzing the anthropometric suitability of the subjects and also the skill of the operators in doing the work with least effort. As all the three subjects are equally trained persons the differences could only be from the anthropometrical angle. The study of the anthropometric data of the subjects shows that subject III had stronger physical features than subjects II and IV, especially chest circumference, chest width, shoulder width, shoulder to elbow length and elbow to waist length. His larger physical features make it possible for him to operate the power tiller with least effort.

It is interesting to note that subject II exhibited almost a constant energy expenditure rate for all dry ploughing operations irrespective of the type of power tiller and inflation pressures. This may be attributed to the subconscious ability of the operator to adjust himself to the operation and exert the constant effort irrespective of the operation.

Comparing the energy expenditure while using different power tillers in black cotton soil the energy expenditure rate was constant for the power tillers A and B for both the subjects II and IV which are 4.99 and 5.74 kcal/min, respectively. The energy expenditure in operating power tiller C is around 55 to 65% of that required for power tiller A or B.

For the red soil and for subject III, the energy expenditure rate for power tiller A and B was higher than that of power tiller C by around 40% confirming the results obtained from black cotton soil.

This is attributed to the fact that this power tiller C with plough is relatively light in weight (165 kg) and easier to handle compared with power tiller B (485 kg) and power tiller A (335 kg). The energy required to operate all the three power tillers at low inflation pressure was minimum in most of the cases. This may be due to the lesser vibration produced at the handle when the power tiller was operated under low inflation pressure. Hence, operation at low inflation pressure enables the operator to endure for a long time with less energy requirement. Low tyre inflation pressure also provides better traction and drawbar pull (Shol[7], McLead[8], Owyer[9] and Carper[10]). There is no significant difference in energy expenditure among the power tillers in black cotton soil and red soil for puddling operations.

The ceiling limits for the energy expenditure standards are 3, 4, 5 and 6 kcal/min and the generally adopted standard is 4 kcal/min. (Murrell[11]). For puddling operations in most of the cases the energy required was more than standard value of 4 kcal/min which indicates that puddling operation is heavier than dry ploughing as walking itself requires greater energy.

Conclusion

1) The energy requirement of a power tiller operator is based on his general physical strength, anthropometric suitability and his sensitivity for working in different soil conditions.

2) The weight of power tiller is the major power tiller design feature which has a major influence on the energy expended by the operator.

3) The low inflation pressure of power tiller has an impact in reducing the energy requirement for power tiller operation.

4) Puddling operation is of heavy nature, since walking in puddled soil itself requires greater energy.

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Prediction of Field Performance of Wheel Tractors


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Abstract

A study on the prediction of tractor field performance was conducted on Latif Farm of Sind Agriculture University, Tandojam. The data pertaining to the parameters concerned with the field performance of rear wheel drive tractor on firm soil were collected and analysed. The curves of firm soil were used to determine slip travel ratio, speed and the ratio of DBHP to the AHP. The results obtained were in close agreement with the experimental results.

The rear wheel drive tractor may be used on firm soil at 10% slip to obtain maximum drawar power. Also, future research work with rear wheel drive tractor may be conducted on various types of soils to evaluate the field performance of various makes of wheel type tractors so that standardization of wheel type tractors may be made in accordance of varying soil conditions which will be guideline for implement designers, in general, and users, in particular.

Introduction

Agricultural mechanization in Pakistan is on the up slope of the curve, but the import of the tractors up to the present time is made without checking their field performance. This has created a lot of misunderstanding among farmers as to which model of the tractor is most suited to the conditions of his farm. Proper standardization of tractors has not yet been followed by the policy makers of the counttry, resulting in the import of non-standardized tractor.

The best way of standardization of farm tractors is to test and evaluate the tractor field performance under varying soil conditions of the country. The criterion which best describes the performance depends largely upon the intended use of the tractor. The performance of farm tractors can be expressed in many ways.

The maximum drawbar pull is often used in comparing or evaluating tractors. Unfortunately, drawbar pull is seriously affected by the soil or test track conditions and also by the gear ratio and the ballast being carried. Since power is a function of both velocity and drawbar pull, it is obvoius that this only partly describes the ability of a tractor to do work.

The objective of this study was to evaluate the field performance of the tractor based on pull and speed. This type of study would provide useful information to the farm power and machinery manager as to how the tractor field performance is evaluated. Also based on the pull and speed relationship, the comparison between appropriate size of farm tractor depending upon the factors responsible for selection of tractor.

Method of Study

Knowledge of tractor field performance is of great interest to the users as it provides the basic tool for selecting the tractors depending upon various factors involved in selection. To evaluate tractor field performance it is imperative that the parameters pertaining to the performance must be determined. Among many parameters, drawbar pull is the most popular method of describing tractor field performance. This method, as compared with others, is easy and quick and has been used by RNAM. This is extensively used to determine the pull of mounted implements as well. It was, therefore, decided to use RNAM procedure for the measurement of drawbar pull of three-bottom mounted moldboard plow. The material required for the project were tractors; i) Fiat 480 used as towed tractor; ii) Ford 4600 used as a power source, hydraulic dynamometer, soil sampler, soil sample containers, stop watch, white chalk, measuring tape, steel ruler, oven, ranging poles, weighing bridge, and three-bottom mounted moldboard plow. The research was conduct-

Table 1 Moisture Percentage on Dry Weight Basis

Sample	Depth (cm)	Weight of wet sample + Container (g)	Weight of dry sample + container (g)	Weight of container (g)	Wt. of wet soil (g)	Wt. of dry soil (g)	Wt. of water (g)	MC% = $\frac{(Ww - Dw)}{Dw} \times 100$
1	5	164.6	152.9	31.6	133.0	121.3	11.7	9.60
2	10	118.0	107.0	31.5	86.5	75.5	11.0	14.56
3	15	114.9	129.0	30.1	114.8	98.9	15.9	16.07
4	20	144.3	126.4	31.3	113.0	95.1	17.9	18.82
5	25	137.5	118.6	30.4	107.1	88.2	18.9	21.42
Average	15	135.8	126.8	30.9	110.9	95.8	15.1	16.94

ed on Latif Farm, Sind Agriculture University, Tandojam and the parameters studied to evaluate the tractor field performance were soil moisture and type of soil, slip and travel ratio, depth and width of cut, pull and speed.

In order to evaluate the moisture content of the experimental plot the soil samples before plowing were collected and weighed in physical balance and then were kept in an oven for 24 hours at 105°C. After 24 hours the soil samples were weighed and moisture content was calculated **(Table 1)**.

Type of soil has profound effect on tractor performance, slip decreases on clay soils and increases on sandy soils. To evaluate the effect of the type of soil on tractor performance, the soil samples were collected and analysed in the department of Agricultural Chemistry, Sind Agriculture University, Tandojam and it was found that the soil of the test plot was clay loam.

Weight Distribution and Stability

It is obvious that the location of the centre of gravity determines the weight distribution on the wheels, the distance ahead of the rear axle determines weight distribution under normal operations. Rear wheel drive tractors usually have 55-80% of the weight on the rear wheels. Front end weight on a two wheel drive tractor does not contribute directly to traction performance but provides a source for weight transfer to the rear wheels as the tractor pulls and is necessary for steering. Excessive front weight would decrease both the tyre efficiency and dynamic ratio (coefficient of traction) obtained from the tractor tyres. The rear tractor weight improves tractive performance. The greater the rear tractor weight, the better is tractive performance. For the purpose of the study, Ford 4600 was weighed on weighing bridge. The weight of the tractor was 2230 kg along with operator. Also, the front end of the tractor was weighed at 765 kg, which was 34% of the total weight.

Slippage and Travel Ratio

The slip of a driving wheel defines the magnitude and efficiency of tractive performance. Slip (travel reduction) of the drive wheels is the primary independent variable, both tyre efficiency and dynamic ratio are a direct function of the amount of slip of the drive wheels. Dynamic ratio may vary from over 0.80 at 15% slip on concrete to as low as 0.30 at approximately 30% slip in sand. Slip is calculated by the following formula:

$$\text{Slip} = 100\ \frac{(1 - As)}{So}$$

where:

As = Actual travel speed in miles/h with load

So = No load travel speed

or

Slip = 100(Wheel revolutions with load − Revolution without load) /(Wheel revolutions with load)

Slip = 100(Distance travelled with load − Distance travelled with load)/ (Distance travelled without load)

Slip = 100(Velocity without load − velocity with load)/(Velocity without load)

Travel ratio = (Velocity with load)/(Velocity without load)

Table 2 Percent Slippage

Gear	Revolutions to cover 43.5m distance without load	Revolutions to cover 43.5m distance with load	Slip
1	10	11.0	9.1
2	10	11.5	13.1
3	10	12.0	16.7
Average	10	11.5	12.96

Table 3 Depth and Width of Cut (cm)

Gear	Depth of cut	Width of cut
1	14.8	86.6
2	13.3	88.6
3	16.0	87.6
Average	14.7	87.6

Table 4 Pull (kg) = H-R

Gear	Speed m/sec	Pull (kg)
1	0.543	568
2	0.725	568
3	0.96	613

The slip is expected to increase with increasing load. Power efficiency decreases as slip increases. To evaluate the slippage of the rear wheel drive tractor, loamy soil on Latif Farm of the University was selected. A 43.5-m distance was measured and the tractor without load and with load was operated over that distance. The revolutions without load and with load were counted and recorded to calculate the slip in percent **(Table 2)**.

Depth and Width of Cut

Pull increases as the depth of cut and width of cut increase. It was, therefore, imperative that the depth and width of cut be measured to evaluate tractor field performance. The depth and width of cut were measured *in situ* with steel tape **(Table 3)**.

Drawbar Pull and Speed

Pull is the total force exerted by

the power unit on the implement. With tillage implements it is generally at some angle above the horizontal, mostly at 13 1/2° to the ground and it may or may not be in a vertical plane parallel to the line of motion. Pull also defines both the magnitude and efficiency of tractive performance. The pull is affected by the depth of cut, width of cut, tool shape, tool arrangement, type of soil. moisture content and travel speed. Various methods have been developed to measure drawbar pull of tillage implements, but in the present study RNAM procedure was used **(Table 4)**.

The speed of operation has profound effect on tractor implement performance. The maximum permissible forward speed is related to such factors as the nature of the operations, the condition of the field and the amount of power available. Increased forward speed increases the draft with most tillage implements because of the rapid acceleration of any soil that is moved appreciably. The soil thus accelerated produces acceleration forces that increase normal loads on soil engaging surfaces, thereby increasing the frictional resistance. To evaluate the speed, a distance of 43.5 m was measured and ranging poles at the two ends of the measured distance were fixed, and time taken for that distance was recorded by a stop watch and the speed was calculated from the formula:

Speed m/sec = (43.5 m test run)/[time taken for test run](sec)

The results thus obtained are shown in **Table 4**.

Results and Discussion

Tractor field performance is of great assistance in selecting the farm tractor. This study provides basic data required by the farm managers so as to run the agricultural mechanization business on sound scientific basis. The data pertaining to the moisture content of the test plot, type of the soil, slip, speed, depth and width of cut, and pull were collected and tabulated. The results obtained from the analysis of the data are discussed in the following paragraphs.

Table 1 shows the moisture content of the test plot. The average moisture content was 16.97%, which more or less agrees with the results of Nichols (1931).

To evaluate the type of the soil, the soil samples obtained from the test plot were subjected to the mechanical analysis by the Department of Agricultural Chemistry, Sind Agriculture University, Tandojam. From the analylsis it was found that the soil type of the test plot is clay loam.

The centre of gravity determines the weight distribution of the wheels. Therefore it was imperative to locate the centre of the test tractor. The weight of whole tractor was determined on the weighing bridge and was recorded at 2230 kg. The front and rear ends of the tractor were also weighed on the weighing bridge and were 765 kg and 1465 kg, respectively, which agree with the conclusions of Barger *et al* (1967).

Both tyre efficiency and dynamic ratio are a direct function of the amount of slip of the drive wheels. Power efficiency also decreases as slip increases. The maximum drawbar power occurs at 30% slip and then drops. "No go" point occurs at 100% slip where pull will be equal to zero. In the present study the slip in the third gear was 16.7%. Therefore, it is obvious that the test tractor was underloaded. To utilize full capacity of the tractor, the only way is to increase the pull up to maximum where 30% slip occurs.

Most available evidence indicates that the specific draft of a plow generally decreases as the depth is increased to some optimum depth/width ratio and then increases as the depth is increased further. The initial of specific draft with increased depth is logical because the total force for cutting the bottom of the furrow slice should be independent of depth. The increase in specific draft beyond the optimum depth is probably due in part to choking of the thick furrow slice in the curvature of the moldboard.

The results of the soil bin tests in sand of the past research indicate that for this one soil condition, varying the width of cut with a 12 in. bottom and a 16 in. bottom had little effect on the specific draft for the bottom alone. But landslide friction, draft due to coulter, and rolling resistance of the plow wheels would change very little and hence would cause the implement specific draft to increase as the width of cut is reduced. However, present results **(Table 4)** indicate that the pull is increased as the depth of cut and width of cut increase, which is in agreement with results presented by Randolph and Reed (1938).

Speed has profound effect on the tractor performance because increased speed will increase draft of plow due to acceleration forces. Thus more pull, more power will be required by the tractor. **Table 4** shows that increased speed has increased the pull which is consistent with the results of Kepner *et al* (1972).

The use of graphical solution based on tyre performance criteria has been explained on **Fig. 1**. The values of slip, kilowatts/ axle power, rear static weight/ axle power have been obtained from the **Fig. 1** and these values match with the values obtained by Frank (1972).

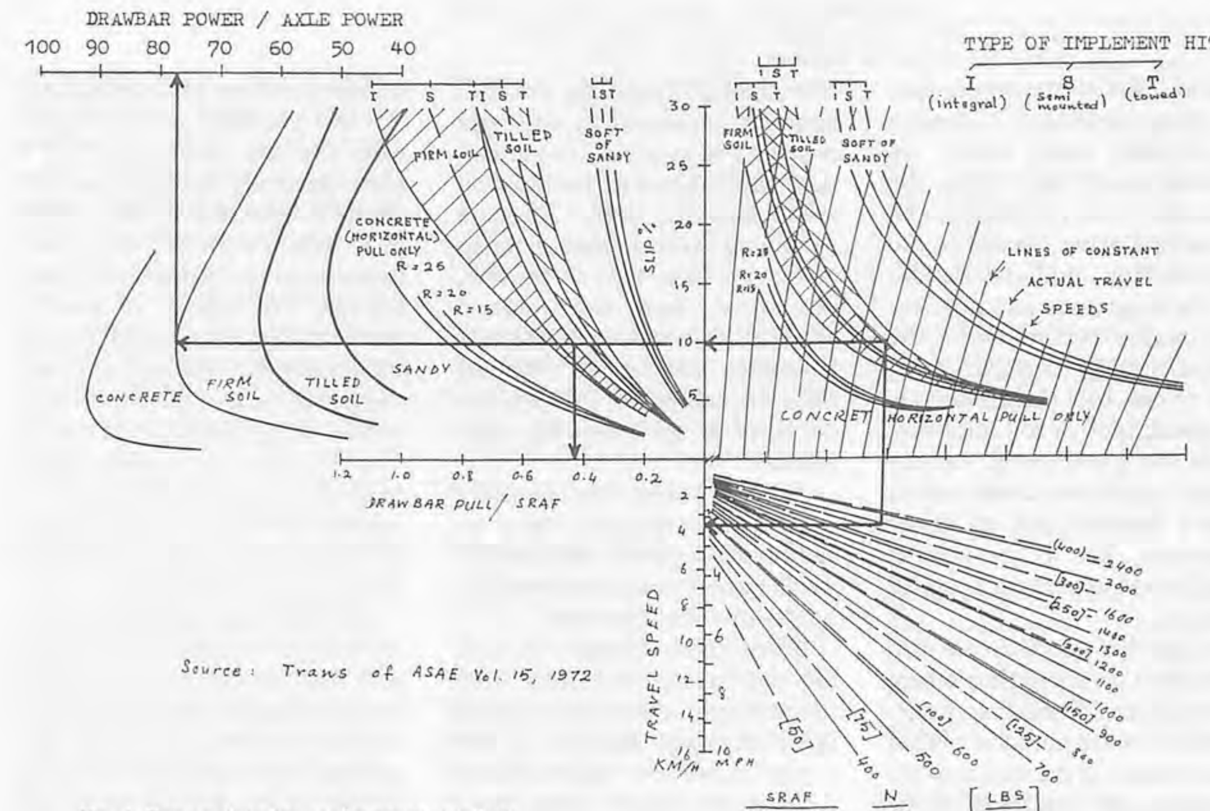

Fig. 1 Tractor drawbar performance predictor.

Conclusions and Suggestions

The tractor field performance of rear wheel drive tractor with rubber tyres has been predicted on the basis of tractor drawbar performance predictor. The ratio of pull to the static weight of the rear wheels has been computed which is 0.43. The speed of the tractor in the third gear was 3.456 km/h. Since the test plot was firm, therefore, firm soil curves were used, and the following conclusions are drawn:

1. The slip on the tractor field performance predictor chart was 10% whereas experimental slip in third gear was 16.7%. The increase of 6.7% in slip was due to the load of towed tractor which was attached in between the pulling tractor and the implement. Frank (1972) also concluded that maximum tractive efficiency (DBHP/AHP) occurs at 10% slip on firm soil. The ratio of DBHP to AHP in the present study was 0.78 which matches with the results presented by Frank.

2. The travel ratio of 0.9 in the present study also occured at 9% slip and also agrees approximately with the findings of Frank.

3. The ratio of rear static weight of the tractor to the axle power in the present study is found to be 1943 which matches with the findings of Frank.

4. The experimental speed was 3.456 km/h and the speed on predicter chart is also 3.5 km/h.

5. It is, therefore, suggested that on firm soils the rear wheel drive tractor should be operated at 10% slip to obtain maximum horsepower. Operation either at low or high slip reduces the power available from the tractor. Since power drops more sharply if too little slip is obtained (excessive rolling resistance) and since part load operations cause the slip to decrease, it is usually desirable to ballast for a wheel slip greater than that required for maximum power.

Further, it is suggested that the graphical solution based on tyre performance criteria may be used to determine the expected drawbar pull, drawbar horsepower, travel speed and slip of any tractor under various soil conditions.

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Diesel Engine Performance Tests Using Oil from Jatropha Curcas L

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Abstract

Oil from seeds of *Jatropha curcas* L. has potential as a diesel fuel substitute to promote agricultural development in Burkina Faso. Agronomic research since 1985 has established the feasibility of jatropha oil production in Burkina Faso. Transesterification using ethyl alcohol has been applied to reduce oil viscosity; also fuel properties of the raw oil and esterified product have been compared with other seed oil and diesel fuels. The results of successful short term experiments to determine diesel engine performance showed that jatropha oil ethyl ester produced 81% of the maximum power, 86% of the maximum torque and 115% of the specific fuel consumption rate of diesel fuel No. 2. No distinction could be made between the superiority of either jatropha or rapeseed oils as alternate fuels.

Introduction

The importance of diesel fuel substitutes from seed oils has been demonstrated by utilization in engines for agricultural purposes during the past decade (1). Burkina Faso (formerly Upper Volta) is a hot and dry country located in West Africa. Traditionally, water availability has been the main problem of human life and agricultural development. In spite of the building of dams and opening of wells throughout the country to provide the inhabitants with water, small farmers are unable to utilize this water to increase food crop production because of the uneven topology of the land. In order to promote agricultural development in Burkina Faso, the price of fuel needs to be affordable to farmers. While Burkina Faso is one of the poorest countries in West Africa, energy prices ($0.2857/kW-h) there are high because of cost of the imported diesel is approximately $0.93/ℓ. The objective of the work presented in this paper was to establish the suitability of fuel from oilseeeds to replace part of the imported petroleum.

The oil from seeds of *Jatropha curcas* L. has potential as a diesel fuel substitute. This shrub was brought to Africa in the 15th century from Brazil and is now widespread in tropical areas (2). Research on *Jatropha curcas* L., a member of the Euphoriaceae family began in Burkina Faso in January 1985 by plant identification, seed collection and studies over a two-year period of plant survival, growth and seed production at two research stations in Burkina Faso (3). The work culminated with the demonstrated use of raw jatropha oil in a small diesel engine. Since viscosity is reportedly a major problem in the use of raw seed oils as fuels in diesel engines (4), transesterification has been used to reduce the viscosity and make the oil more suitable as a fuel substitute. Reported here are results of successful short term experiments to compare diesel engine performance using raw and transesterified oils with analogous rapeseed products and No. 2 diesel fuel.

Oil Production

A private research organization, SERAGRI, in Burkina Faso successfully demonstrated the use of jatropha oil as a fuel substitute by running a diesel engine for approximately 60 hours from 1985 to 1986. The characteristics of the Laombardini Genelic series 9040 diesel engine used in the previous experiments were as follows: one vertically mounted cylinder with a piston stroke of 68 mm and a

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cylinder bore of 82 mm (displacement of 359 cm^3) generated 5 kW at 300 rpm and powered a 220 volt mono generator. After this encouraging result, studies were undertaken to collect more information on plants growing under natural conditions. Experimentation was made to carry out plant adaptation to the Salielian climate zone of the country and to etimate plant production. The preliminary results presented here are mainly concerned with seed production; plant survival and growth studies are reported elsewhere (3).

The two experimental sites were the "Bosquat de 5 Juin" near Boulminou Dam at Quagadougou (site A) and the SO.FI.TEX Weather Station at Sourou near the Sourou River (site B). Both were situated along the same rainfall line of 800 mm of water per year (**Fig. 1**). The differences between the two sets of plantings were as follows:

Item	Plot A	Plot B
Plot size	1296m^2	1600m^2
Elevation	plateau	valley
Transplanting	7/6/85	
Sowing		9/3/85
Plant spacing	4m × 4m	2m × 2m

Survival rates were greater at plot B than at plot A. Plant death at site A was due more to termites than to drought. All performance parameters were better for plants in the valley than on the plateau. The plants were taller, stems were larger and space taken by the foliage were greater at plot B than at plot A. No seeds were harvested from plot A in the first year, but plants at plot B produced seeds with characteristics similar to those reported by Martin and Mayeux (2). The first year production was short of that reported in the literature for plants at maturity (5 to 6 years), from which 200 to 800 kg oil per hectare could be expected (**Table 1**). Oil used in subsequent studies was obtained by hexane extraction of embryos from seeds harvested from plot B in 1986.

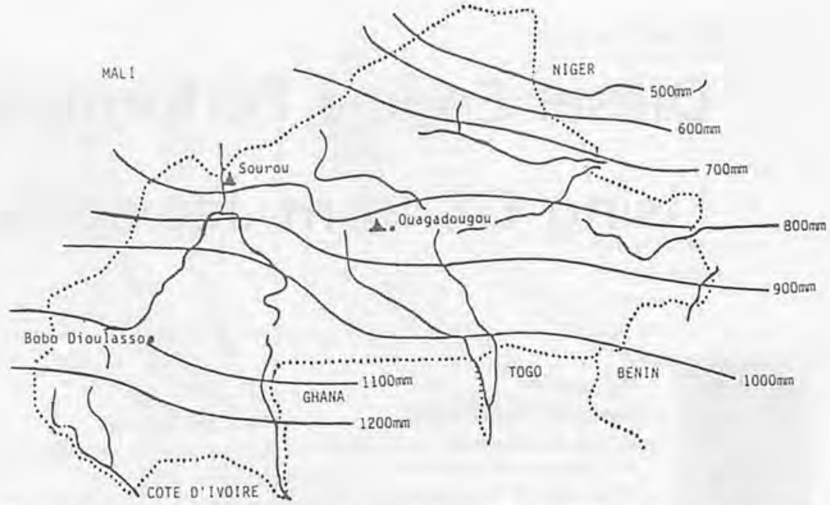

Fig. 1 Map of Burkina Faso showing equal rainfall lines and sites of two agronomic studies in 1986 and 1987. Ouegadougou and Sourou lie 269 km apart.

Table 1 Plant Production Data and Oil Yields from *Jatropha curcas* L.

		Plant Production Data		
		Fruits	Seeds	Capsules
Burkina Faso (3)	kg	14.58	7.89	6.69
Burkina Faso (3)	%	100	59±3	41±3
Literature (2)	%	100	53-62	38-47
		Oil Yields		
	Fruits	Seeds	Embryo	Oil
Kg per plant (2)	4-5		3-3.5	1.5-2.0
Kg per ha (2)		650-2 000	400-1 200	200-800
Kg per ha (3)	91.15	49.33		

*First year production - 1986.

Transesterification

Jatropha oil is a triglyceride; the transesterification reaction fundamentally replaces the glycerol by short chain alkyl alcohols (R-OH).

CH_2-OOC-R_1	CH_2OH + R-OOC-R_1
CH-OOC-R_2 + R-OH →	CH-OH + R-OOC-R_2
CH_2-OOC-R_3	CH_2OH + R-OOC-R_3
Triglyceride catalyst	ethyl esters
alcohol	glycerol

Since a sorghum alcohol industry exists in Burkina Faso, ethanol could be used in the transesterification reaction mixture to produce three long chain fatty acid ethyl esters from each triglyceride molecule. An excess of alcohol is used and both acid and alkali catalysts have been applied, but alkali is preferred in preparation of oils for fuel purposes. The reaction can occur over a range of temperature (20-85°C). In the present case, 1.0 weight percent sodim hydroxide, a 6:1 stoichiometric alcohol: oil ratio, anhydrous conditions and vigorous stirring at room temperature was used (5).

Jatropha oil was prepared by hexane extraction of dehulled embryos, as described above. Rapeseed oil, kindly provided by R.A. Korus from the University of Idaho, was prepared by pressing seeds a *Brassica napus* L. (var. Dwarf Essex). Using the conditions for transesterification described above, ethyl esters of jatropha oil and rapeseed oil were prepared in 0.5 liter quantities. Chromatography of the reaction products after exhaustive washing with water was conducted using a 1 m × 0.22 mm id glass column packed with 60/80 mesh Chromosorb WHP and 5% C20M stationary phase (Alltech Associates, Deerffield, IL) in a HP5840A gas chromatograph programmed from 150 to 175°C at 2.5°C/min with a helium carrier gas flow rate of 23.9 mℓ/min. The injector temperature was maintained at 200°C and the flame ionization detector temperature was 225°C. In **Fig. 2**, the chromatogram shows that myristic (14:0), palmitic (16:0), oleic (18:1), and linoleic (18:2) acids predominated in jatropha oil. The major fatty acids in rapeseed oil were stearic (18:0), oleic (18:1), linoleic (18:2), linolenic (18:3) and erucic (22:0).

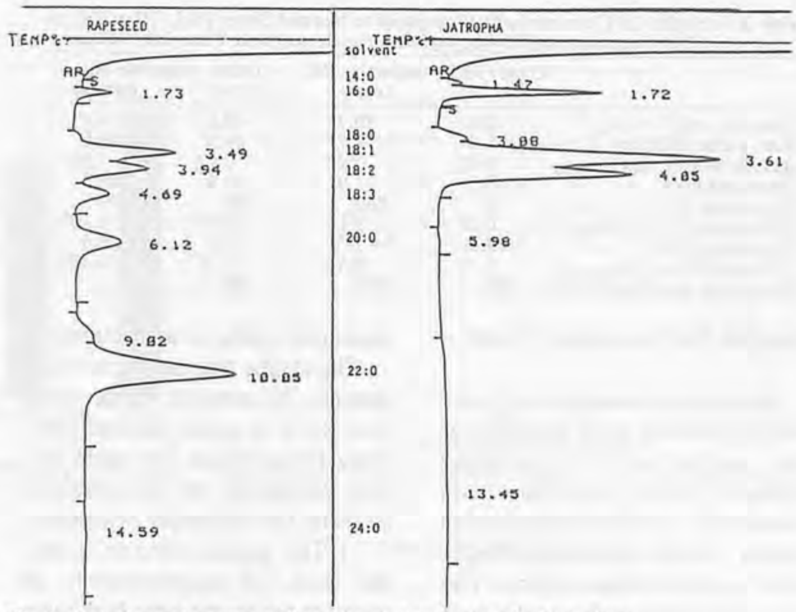

Fig. 2 Gas chromatograms of jatropha and rapeseed oil ethyl esters.

Table 2 Fatty Acid Composition and Fuel Value Index of Six Seed Oils.

		Fatty acid composition							Fuel value index g
		16:0	18:0	18:1	18:2	18:3	20:1	22:1	FVI
Diesel No. 2.									130-246
Winter Rapeseed	(6)	3.01		18.45	14.84	7.08	10.46	45.8	203.06
	(10)	3.49	0.85	64.4	22.3	8.23			223.78
Safflower (H.O.)	(6)	4.75		74.12	19.74				157.83
	(10)	5.46	1.75	79.36	12.86				138.01
Safflower (H.L.)	(6)	5.87		8.84	83.76				349.75
	(10)	8.6	1.93	11.58	77.89				333.67
Soybean	(11)	8.0	3.0	26.0	49.0	11.0			321
	(10)	11.75	3.15	23.26	55.53	6.3			310.68
Sunflower	(11)	6.4	1.3	21.3	66.2	0.1	4.0	0.8	299.96
	(11)	6.08	3.26	16.93	73.73				321.19
	(12)	6.0	4.2	18.7	69.3	0.3	0.1		

FVI = 1.0 [÷16=0] + 1.0 [÷18:0] + 1.0 [÷18=1] + 4.0 [÷18=2] + 8 [÷18=3] + 1.1 [÷20:1] + 1.2 [÷22=1]

Oil Quality

The fatty acid composition of jatropha oil published in the literature (Martin and Mayeux, 1984) agrees with that shown in **Fig. 2**. Compared to other seed oils (safflower, soybean, sunflower and rapeseed) which have been studied in the United States, jatropha oil contains higher percentages of C16:0 and C18:0 fatty acids than to other oils. According to these values, jatropha oil is more saturated than others. The suitability of jatropha oil for a diesel fuel substitute is supported by the fact that the fuel value index (FVI), calculated according to the equation found at the bottom of **Table 2**, is in the range of comparable fuels.

In long term diesel engine stand tests, engine failure is commonly reported with the use of raw rapeseed and soybean oils (4). As has been described by many authors, fuel with high viscosity is difficult to pump, modifies the lubricating oil properties upon leaking past piston rings and sometimes causes ring sticking. In addition, viscous oils are incompletely burned in the combustion chamber, which results in carbon buildup that covers the piston surface and plugs the injector nozzle (3, 6).

Fuel properties of No. 2 diesel and raw and esterified seed oil, that have been determined using ASTM and AOCS tests and compiled from publications and reports, are summarized in **Table 3**. The main problem with jatropha oil is the relatively higher viscosity compared to that of other fuels. Transesterification of jatropha and rapeseed oils using ethyl alcohol as described above reduced the viscosity of 67% and 83%, respectively, as shown in **Table 4**. Viscosity was determined using a Brookfield Viscometer and rotating LV-2 spindle standardized against Cannon Certified Viscosity Standard Oil S600 at 20°C. The viscosity of the oil was reduced in both cases by formation of the ethyl esters, but not to the extent expected from literature information, represented by the average value tabulated in **Table 3**. Nonetheless, these esterified products were used in engine performance tests described below following filtration through Whatman No. 42 filter paper. Densities of the washed reaction products were also determined in triplicate using 10 ml pycnometer flasks (**Table 4**). These results show that transesterification reduced the density of both the raw oils by approximately 5 %.

Table 3 Jatropha Oil Characteristics, Compared to Normal Diesel Fuel, Other Vegetable Oils (rapeseed, safflower, soybean, sunflower) and Their Ethyl Esters.

	Diesel fuel (No. 2)	Jatropha Oil (raw)	Other vegetable oils (raw)	(esters)
Viscosity (cp)	2.4	50.7	40.6	4.7
Heat Value (MJ/kg)	45.3	39.7	39.5	40.0
Specific weight (kg/l)	0.87	0.92	0.92	.88
Cetane number	45.8	51.0	40.6	49.7
Flash point (c)	78	240	265	179
Sulfur content (% mass)	0.25	0.13	0.05	0.101
Cloud point (°C)	−19.0	−5.0	+1.1	−4.0
Carbon residue (%)	0.14	0.64	0.26	0.495
Exhaust gas temperature (°C)	690	508	704	—

Table 4 Viscosity and Density Measurements on Rapeseed Oil; Jatropha Oil and Their Ethyl Esters

Sample	Viscosity centipoise	Density kg/L
rapessed oil	71.8	.923
rapeseed ester	11.9	.883
jatropha oil	46.2	.920
jatropha ester	15.1	.884

Engine Performance Tests

Engine performance tests were conducted with each fuel type at the Agricultural Engineering Research Center, Colorado State University. The engine used in the testing was an air-cooled Petter AA-1 4 cycle diesel engine. The rope start diesel engine had a bore of 7 cm (2.75 in) and a 5.7 cm (2.25 in) stroke with a total engine displacement of 86.5 cubic cm (13.4 cubic in). This single injector engine (using direct injection) was rated at 2.8 kW maximum power at a 3600 rpm engine speed. A 100 ml pipette was attached to the entrance of the fuel pump to allow testing of the different fuel types. The return flow line was placed in a beaker.

A Go-Power model DY-70 dynamometer was used to test the engine performance with each fuel type. This dynamometer uses a hydraulic brake to convert the rotating torque (at the engine crankshaft) to a stationary torque (at the vaned housing). The power absorption unit consists of a vaned impeller rotating in the stationary housing that is partially filled with water. As the vaned impeller rotates, water is accelerated outward until the water strikes the outer edge of the housing and is deflected against stationary vanes. (This will cause the housing to try to rotate). A load cell restricts the housing from rotating and measures the torque the water exerts on the housing. By controlling the amount of water in the housing (using a water valve) the engine load can be varied. The engine crankshaft rotational speed is measured using a tachometer.

The engine was run for approximately 30 minutes using diesel fuel No. 2 to attain normal operating temperature. For each fuel type the engine test was conducted using the following procedure:

1) The engine was run at partial load for approximately 10 minutes using the new fuel type.

2) The load was removed, the engine was set at full throttle and the engine speed was measured.

3) The amount of fuel consumed for approximately 20 seconds of opertion (precisely timed with a stop watch) was determined using difference readings of oil contained in a 100 mℓ buret.

4) The engine load was increased and the torque, engine speed and fuel consumption measurements were taken.

5) The load was increased by increments and the measurements were repeated.

6) About 10 load settings were measured as the engine was loaded from high idle (approximately 3750 rpm) to approximately 2200 rpm.

7) The load was then incrementally decreased with measurements taken at each load setting until the engine reached high idle.

An engine performance test was conducted with the diesel fuel initially. The fuels were then tested in the following order: rapessed oil, rapessed ester, jatropha oil and jatropha ester. Finally, another diesel fuel test was conducted.

Results and Discussion

Table 5 summarizes the results of the engine performance tests. The maximum power for the diesel fuel tests were both 2.1 kW. These values were considerably lower than the maximum power listed in the manufacturers' specifications (2.8 kW). The Agricultural Engineering Research Center is located at an elevation of 1525 m (5000 ft), thus lower engine performance is expected. The rapeseed and jatropha oils produced maximum powers of 1.3 and 1.5 kW, respectively. This represents a drop in power of about 30%. The rapeseed and jatropha esters produced maximum powers of 1.85 and 1.7 kW, respectively. The esters produced about 15% less power than the diesel fuel. **Fig. 3** presents the relative maximum power produced by each fuel type. The engine performances on the first and second diesel tests were very similar. Results from previously published reports indicate that vegetable oils

Table 5 Summary of Engine Performance Tests

	Maximum power (kW)	Rated speed (rpm)	Maximum torque (N-m)	Specific fuel consumption (kg/kW-h)
Rape oil	1.30	3 200	4.10	0.57
Jatropha oil	1.50	3 250	4.55	0.56
Rape ester	1.85	3 200	5.90	0.53
Jatropha ester	1.70	3 150	5.60	0.53
Diesel (first)	2.10	3 150	6.50	0.45
Diesel (second)	2.10	3 300	6.50	0.47

Note: Specific fuel consumption data is an average value taken between the rated engine speed and full load.

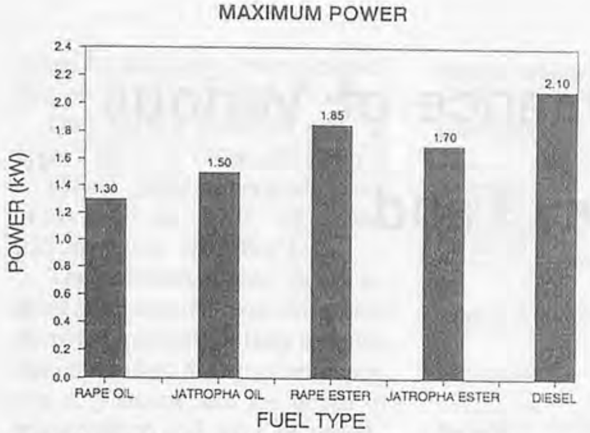

Fig. 3 Comparison of maximum power determinations from engine performance tests.

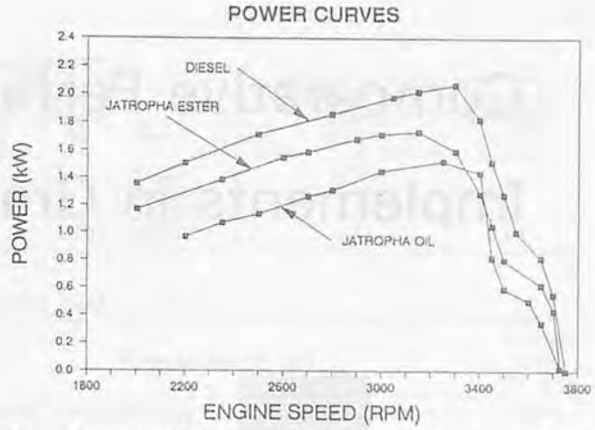

Fig. 4 Power curves for diesel, jatropha oil and jatropha oil ethyl esters.

and their esters produce similar power characteristics as diesel fuel (7, 8, 9). The decline in maximum power found in the experiment conducted in this study could possibly be explained by the high altitude at which the tests were performed. However, it is noted again that the jatropha and rapeseed fuels produced similar power characteristics.

The rated engine speed for each fuel type was approximately 3200 rpm. The rated engine speed was the engine speed that produces the maximum power. The power curves for the jatropha oil, the jatropha ester and the final diesel fuel tests were typical **(Fig. 4)**, except that a discontinuity in the power curves from 3400 to 3600 rpm was observed, possibly because the governor was not operating correctly at these speeds. The loading curves show that the relationships of diesel, the oils and their esters are consistent with respect to engine speed.

The maximum torque produced using the oils and esters were also lower than the maximum torque produced using the diesel fuel **(Talbe 5)**. This reduction in torque is directly responsible for the lower power values. Also, the specific fuel consumption values for the oils were higher than those for the esters, which in turn were higher than the specific fuel consumption for the diesel fuel **(Fig. 5)**. The rapeseed and jatropha oils produce similar engine performance characteristics, and the same was observed for the rapeseed and jatropha esters.

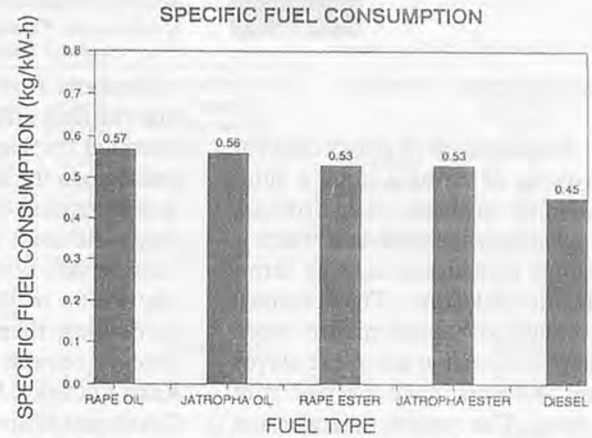

Fig. 5 Comparison of specific fuel consumption rate determinations from engine performance tests.

Conclusions

In lesser developed countries, such as those in West Africa, the development of biomass processing for fuel substitutes production will contribute to a better life in rural communities. Potential for using seed oils, such as jatropha oil that grows naturally in West Africa, as diesel fuel has progressed through studies on oils of rapessed, soybean, sunflower and safflower. The viscosity of these seed oils has been shown to cause engine fouling in long term use. The chemical process, transesterification with ethanol, has been used widely to lower the viscosity of such oils. This technique has been succesfully applied in preliminary bench scale trial to prepare jatropha oil for diesel fuel substitute.

The rapeseed and jatropha oils and esters produce lower power characteristics than diesel fuel. Also, the esters exhibit better power characteristics than the raw oil. Compared to a diesel-fueled engine, the loss in power resulting from using raw oil or ester is about 30% and 50%, respectively. From these results, no distinction can be made regarding the superiority of either rapeseed or jatropha as an alternate fuel.

(Continued on page 32)

Comparative Performance of Various Implements in Grassy Land

by
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Introduction

Development of grassy land for growing of crops is quite a cumbersome problem with ordinary implements like cultivator which is mainly owned and used by farmers in Pakistan. The problem becomes of acute nature when there is Sudan grass in the clayey soil. Grasses have fibrous root system. The roots are slender and fiber-like and extend laterally and downward deeply.

The cultivator, a conventional implement, becomes too difficult to operate in heavy soil with thick grasses. It simply uproots the grass hardly at the depth of 10 to 12 cm in clayey soils under ordinary conditions. The partically uprooted grass entangles between the tines and makes the cultivator operation difficult.

There are many other commercially available implements which may take up the development of grassy land in a better way. The main reason for not adopting them seems a lack of introduction and confidence on the part of farmers in their workability. The Director General Agriculture (Extension and Adaptive Research) desired that the performance of different implements should be tested in grassy land with clayey soil in the form of demonstration to farmers. It is an admitted fact that the field efficiency of tractor-operated implements heavily depends upon the size of fields. If the demonstration is given on small field, the field efficiency lowers considerably which gives negative impression to farmers instead of motivating them. In the meantime, a request of farmer from Rakh Ladheke, Mauza Jia Bagha, Tehsil and District, Lahore came to the Director General Agriculture (Extension and Adaptive Research) office for the demonstration of agricultural implements.

In March, 1986, the site was visited. It was found that the requisite type of soil, i.e.; heavy clay covered with Sudan grass was available there in three adjacent plots comprising of 270 m × 90 m, 270 m × 120 m and 225 m × 90 m sizes where the demonstration/trial was to be arranged.

Having confirmed the availability of land in the required size and form the study was arranged for demonstration and evaluation of operational effectiveness of rotavator followed by cultivator, disc harrow and chisel plow in developing the grassy land.

Review of Literature

No such study specific to clayey soil covered with Sudan grass is available for reference. Evaluating the comparative performance of rotavator followed by cultivation, disc harrow and chisel with specific reference to clayey soil covered with Sudan grass seems an original work.

Materials and Methods

Having ascertained that fairly large plots were available at the site of demonstration, it was decided to take departmental rotavator, disc harrow and chisel plow along with tractor MF-265. The farmers could arrange 47 HP tractor. As such it was further decided that demonstration/trial of rotavator, disc harrow and chisel plow would be arranged with the departmental tractor and that of cultivator with the tractor to be made available by farmers. Brief specifications of implements are given below:

Implement	Specifications	Width (cm)
Rotavator	Howard, 36 blades	150
Disc harrow	Nardi, Offset 16 discs	155
Chisel	Agric, 3-tines	119
Cultivator	local made, 9-tines	206

Tractor and implements were moved to the site on 10.3.1986.

Following treatments were put to test:

Rotavator operated once followed by cultivator once in plot of 270 m × 90 m (24300m^2) size.

Disc harrow operated once in area, 270 m × 120 m (32400m^2).

Chisel plow operated twice crosswise in plot of area, 225 m × 90 m (20250m^2).

The selection of area for the individual treatment was done randomly. Operational time to cover the earmarked area by the respective implement and the depth of manipulated soil were recorded.

The distance travelled in metres by the tractor in 5 min was recorded. Ten readings were taken with each implement operation working. Each reading was converted into m/sec and the average speed was computed. The condition of soil and grass worked with were also noted.

Results and Discussions

Effective Filed Capacity

Time taken by each implement to cover the earmarked area once and effective field capacity such calculated are given in **Table 1.**

Working Speed

Average working speeds of tractor derived from ten readings taken during each operation are given in **Table 2**.

Theoretical Field Capacity

Theoratical field capacities calculated on the basis of width of each implement and average speeds are given in **Table 3**.

Field Efficiency

Field efficiencies computed on the basis of effective and theoretical field capacities of individual implement operation are given in **Table 4**.

Working Depth

Observations recorded at 10 places and average depths computed are given in **Table 5**.

Table 1 Effective Field Capacity

Implement	No. of Plowing	Time per plowing (h)	Area m^2	Area Acres	Effective field capacity (AC./h)	Operational time/acre (h)
Rotavator	1	8	24300	6	0.75	1.83
Cultivator	1	3	24300	6	2.00	
Disc harrow	1	8	32400	8	1.00	1.00
Chisel plow	2	4.50	20250	5	1.11	1.80

Table 2 Average Working speed

Implement	Observations (m/sec)	Average speed (m/sec)
Rotavator	0.75, 0.77, 0.74, 0.76, 0.76, 0.73, 0.75, 0.75, 0.74, 0.75	0.75
Cultivator	1.45, 1.44, 1.43, 1.47, 1.46, 1.43, 1.48, 1.45, 1.44, 1.45	1.45
Disc harrow	0.91, 0.94, 0.95, 0.89, 0.87, 0.90, 0.88, 0.91, 0.95, 0.90	0.91
Chisel plow	1.23, 1.23, 1.25, 1.20, 1.20, 1.24, 1.25, 1.23, 1.24, 1.23	1.23

Table 3 Theoratical Field Capacity

Implement	Width (m)	Average speed (m/sec)	Theoretical field capacity (acres/h)
Rotavator	1.50	0.75	1.00
Cultivator	2.06	1.45	2.66
Disc harrow	1.55	0.91	1.25
Chisel plow	1.19	1.23	1.30

Table 4 Field Efficiency

Implement	Field efficiency (%)
Rotavator	75
Cultivator	75
Disc harrow	80
Chisel plow	85

Table 5 Depth Working

Implement	Observations	Average
Rotavator	7.00, 7.20, 6.50, 6.51, 7.50, 7.50, 7.00, 7.00, 6.79, 7.00	7.00
Cultivator	12.20, 12.50, 12.00, 11.50, 11.75, 12.35, 12.00, 11.80, 11.90, 12.00	12.00
Disc harrow	14.50, 15.60, 15.30, 14.80, 14.60, 15.10, 15.20, 15.10, 15.00, 14.80	15.00
Chisel plow	39.50, 38.00, 40.50, 41.60, 42.00, 39.00, 38.20, 40.00, 41.20, 40.00	40.00

Condition of Field after Operation

The condition of fields observed after each implement operation is recorded in **Table 6**.

The effective and theoretical field capacities (2.00 and 2.66 acre/h) were higher cultivator operation. This is due to wider width and attainment of high working speed made possible with the rotavation of grass from the soil surface. However, the combined operation of rotavator followed by cultivator consumed maximum time per acre, i.e., 1.83. No doubt, the soil was cleared of grass with rotavator up to an average depth of 7 cm which was further loosened up to 12 cm with the successive operation of the cultivator. But roots of the grass were found below the working depth. A field efficiency of 75% each was obtained in rotavator and cultivator operations.

The operation of disc harrow resulted in 1.00 and 1.25 acre/h effective and theoratical field capacities, respectively. The operational hours per acre was computed as 1.00 h. The field efficiency of 80% was obtained with this implement. Disc harrow cut grass roots up to an average depth of 15 cm and reasonably pulverized the soil. Grass roots were found below the working depth.

The use of chisel plow gave 1.11 and 1.30 acres/h effective and theoretical field efficiencies, respectively. The operational

Table 6 Field Conditions After Manipulation

Implement	Field condition
Rotavator	Field surface become clear of grass. Grass was cut into small pieces and mixed with soil up to an average depth of 7 cm. Below working depth, tremendous roots existed
Cultivator	This implement followed rotavator, which had crused grass and its roots up to 7 cm. So, the cultivator operation was eased substantially. The Cultivator loosened the soil up to an average depth of 12 cm. The roots of grass below 7 cm were removed partically. Many roots were observed uncut below the working depth.
Disc harrow	This implement manipulated soil up to average depth of 15 cm. Grass was partically cut into working depth. Many roots were observed existing below the operational depth of disc harrow. The cut grass was reasonably shattered and mixed with soil due to side shifting of soil with the first gang of discs and reverse action of rear one.
Chisel plow	The soil was opened up to an average depth of 40 cm. Grass was completely uprooted as observed after the second pass of chisel plow. Uprooted grass did not entangle with plow and blocked its operation due to wide spacing between tines.

hours of 1.80 acre/h was obtained. The operation of chisel plow resulted in field efficiency of 85% which was highest. The chiesel plow operated twice, opened soil up to an average depth of 40 cm and completely uprooted the grass roots.

Conclusions

Sudan grass has deep roots and the use of deep plowing implements is essential for its uprooting.

In clayey soil, Rotavator crushes grass up to an average depth of 7 cm.

The cultivator following rotavator increases the manipulation of soil up to an average depth of 12 cm but still leaves tremendous roots uncut below the working profile.

The effective and theoratical field capacities of rotavator (150 cm width) in clayey soil covered with grass are 0.75 and 1.00 and that of following cultivator (206 cm width) are 2.00 and 2.66 acre/h respectively. The field efficiency of each of these implements is 75%.

The use of disc harrow pulverizes the soil up to an average depth of 15 cm. It partically crushes grass and its roots up to the working depth. Many roots remain uncut. The effective and theoretical field efficiency of 155 cm width disc harrow is 1.00 and 1.25 acre/h. The field efficiency is 80%. The chisel plow operated twice uproots Sudan grass in clayey soil completely. The chiesel plow of 119 cm width gives effective and theoretical field efficiencies of 1.11 and 1.30 acre/h. The field efficiency is 85% which is of the highest magnitude of other implements viz; rotavator, cultivator and disc harrow. As such, the use of chisel plow is most efficient and ideal to work clayey soil covered with Sudan grass. ■■

(Continued from page 29)

Diesel Engine Performance Tests Using Oil from Jatropha Curcas L

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A New Design Concept for Animal-drawn Harvester

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Abstract

Animal-drawn cereal harvesters are extremely important in India to meet the harvesting needs of various crops. The earlier attempted designs on animal-drawn harvesters have limitations of ground traction and high draft requirements. In this paper attempt has been made to reduce the animal draft requirement. Also, the cutting force available at the cutter bar had been increased by 6-8 times approximately.

Introduction

Harvesting is one of the most important operations in crop production. If it is delayed, it will have certain detrimental losses. The optimum period of harvesting of a crop is at its biological maturity, when yield is maximum and losses are minimum. Some crops are highly susceptible to shattering e.g., soybean, mung bean and some varieties of paddy. The specific benefits of early harvesting soon after attaining the biological maturity has innumerable benefits; more crops could be grown in a year, turn around time (time between harvesting the first crop and sowing of next crop) is available and sort duration crops could be grown effectively. In most cases some harvesting equipment are needed. The harvesting of rice is done still using sickles in India. The combines are few in number. There is a large gap left between sickles and combine as there is no suitable intermediate equipment available for harvesting. The efficiency of manual harvesting system using sickles is considerably low. It requires about 180-200 man h/ha to harvest paddy and wheat (8, 13). Time required mostly depends upon the type of crop, its physical strength and the seasonal variation during the harvesting period.

It has been observed that energy required for harvesting crops after attaining the biological maturity is minimum (5). This, however, poses operational problems. The heavy machines cannot enter into the field due to high soil moisture and green vegetation trends to choke the machines. A design of low-weight machine is a hard task for the designers.

There have been many attempts to develop tractor and power tiller-operated machines for the harvesting of crops by Devnani and Pande (1985), Garg and Sharma (1984), Howson and Devnani (1981). Through RNAM (Regional Network of Agricultural Machinery) some institutions in a few countries: Philippines, China, Korea, Pakistan and India have made great efforts to develop tractor and power tiller operated harvesting machinery and trying to popularize them. It has certainly enhanced the mechanization level to some extent within a few years. However, it has its own limitations as these machines can be used by 10-12% of the farmers only when in full operation. In addition, efforts were made by many to design and develop manually-operated harvesters; Rahman and et al (1980), Mollah and Kilgore (1986), Srivastava and Dyck (1978).

Earlier efforts of Devnani and Nag (1970) for the development of power tiller-operated harvesters and Verma and Garg (1971) on tractor-operated harvesters had been quite encouraging. However, due to some problems in the adoption of these machines it could not make a dent. The design efforts made by IARI in developing the IARI reaper and TNAU for making tractor- and power tiller-operated harvesters were good. However, none of these machines could reach to the industrial stage

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for the benefit of the farming community. The recent efforts made at CIAE, Bhopal and PAU, Ludhiana in design and development of the IRRI-designed tractor-operated, front-mounted vertical conveyer reaper had a very good beginning and getting quite popular for some crops.

A positive design effort was made, in another study, at the Central Institute of Agricultural Engineering, Bhopal to develop a need-based tractor-operated soybean harvester (Yadav and Yadav, 1985). Basically, the machine was designed to meet the needs of soybean growers in the state of M.P. and in the country. Since this machine is equally good for harvesting wheat and other cereals and pulses, it has become very popular. More than a dozen manufacturers have taken up its manufacture and marketing. However, it is also in limited use due to the requirement of tractor power.

Research Efforts on Animal-drawn Reaper

There are some good efforts made at PAU, Ludhiana in the 1960s to develop animal-drawn harvesters based on the McCormick horse-drawn reaper/mower for lawns and grasses. The unit had high draft (129-180 kg) for a pair of bullocks and higher still in the field with irrigation bunds.

The work reported at Ludhiana had an engine mounted on the machine. The animal could not accommodate the engine sound and machine vibrations. A similar effort was also made at JNKVV Jabalpur by mounting an engine on the harvester. The above mentioned animal-drawn harvesters were invariably heavy and the ground drive wheels were not able to develop sufficient traction to operate the mowing system. The metal wheel causes more of ground friction and high draft force. The efforts made by Singh (1981) using the pneumatic wheels and oscillating mechanism had some possibility in lowering the draft and making a workable system. However, as the system still has an inclined cutter bar drive and the ground traction for drive was taken from the pneumatic wheels, enough force was not available to cut the crops. To achieve clean cutting it requires high draft.

It is reported that 42% of the agricultural power in India is still obtained from the draft animals (10, 12, 16). This potential power is readily available in the country within the means of most farmers. In fact, during harvesting season this rare but most useful power source is kept idle. If suitable harvesting machine is available, this power source could be effectively utilized in addition to the present uses of the power sources in harvesting.

There is still strong conviction that a better designed animal-drawn harvester will work satisfactorily for harvesting cereals.

Keeping in view the non-availability of suitable harvesting machines for use of draft animals to benefit the peasant farmers, it was decided to develop an animal-drawn cereal harvester. Various failure points of the earlier machines were carefully observed before taking up the new design.

New Design Specific Points

1. The total weight of the machine should be low and should be carried on the least friction type wheels;

2. A large mechanical lever system should be used in the application of cutting force; and

3. The cutter bar mechanism should develop zero or near zero vertical load to reduce the knife and cutter bar friction.

A small force and a long lever can be effectively utilized for performing the given task easily and effectively in order to produce a large force using a small lever.

Objective

The specific objective of this paper was to analyze the theoretical design benefits of the following mechanism being used on the cutter bar type cereal harvesters which, in fact, is being used in various makes of harvesters: i) offset crank mechanism; ii) in-line crank mechanism; oscillating lever mechanism; and lever crank mechanism.

The offset crank mechanism shown in **Fig. 2** is common and found in use on most conventional harvesters. It is easy to adopt.

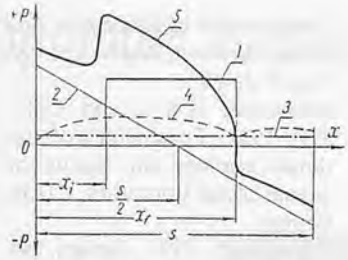
The in-line crank mechanism is not common **(Fig. 3)**. It can be used on the harvesters where cutting is required at certain height above the ground, assuring sufficient clearance.

The oscillating lever mechanism **(Fig. 4)** is used for reducing the friction force on the cutter bar and reduces it considerably.

The forces acting on the knife section during harvesting is given as follows (Klenin et al, 1986);

$$Ps = Pav + Pj + f\,Gk \quad (1)$$
$$F = F1 + F2 \quad (2)$$
$$F1 = Gk\,f \quad (3)$$
$$F2 = (Pav + Pj + f\,Gk)\,\mathrm{Tan}\,B \times f/(1 - f\,\mathrm{Tan}\,B) \quad (4)$$
$$Pav = e\,F1\,Z/Xc \quad (5)$$
$$Pj = Mk\,jk \quad (6)$$
$$jk = rw^2\,\mathrm{Cos}\,wt^2 = rw^2\,[1 - (x/R)] \quad (7)$$
$$Pj = Mk\,rw^2\,(1 - (x/r)) \quad (8)$$

It could be observed from **Fig. 1** that the friction force (4) caused due to pressure on the connecting rod, could be eliminat-



1. cutting force, 2. inertia force, 3. friction force due to weight of knife, 4. friction force due to pressure of connecting rod, 5. resultant force

Fig. 1 Variation of forces acting on knife as a function of its displacement.

ed or reduced to a bare minimum value.

Since a pair of draft animals can produce a maximum of 1 HP, all the necessary force has to come from this power source only during the operation of the harvester. The existing machines having the mechanism as shown in **Figs. 2-4** can add the undesirable draft force. The power for the cutting of crops is available through the ground drive wheels only. This soil thrust is limited to the friction between the traction add and soil. Animal-driven machines are short of such tractive power. The reduction forces caused due to faulty design of the machine is critical. In addition a suitable provision for increasing the thrust through the same ground traction is most important.

The lever crank mechanism for the application of higher force to the knife section is shown in **Fig. 5**. All the design features of the machine are the same. The only change was adapted in the connecting rod and its linkages to connect the knife section. The crank lever oscillates about the fulcrum point in the middle, causing a movement of the knife section. The small length of the crank lever (1) oscillates and pushes the knife section horizontally on the cutter bar. This lever is free to slide up and down in the vertical slot attached at the inner end of the knife section. The

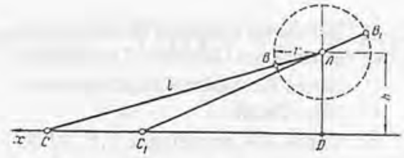


Fig. 2 Offset crank mechanism.

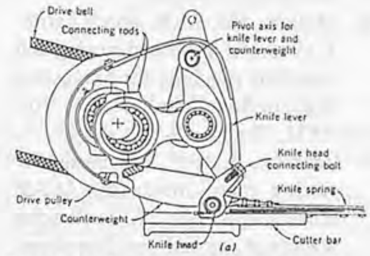


Fig. 4 Oscillating lever mechanism.

effective length of the crank lever increases at each end of the stroke. The minimum length (1) is in the middle of the stroke.

Assuming force (f) is produced to the interaction of the ground drive wheel and the lugs provided, the force developed at the rated speed of the ground drive wheel is given as:

$$p \times L = P \times l$$

Thus the cutting force can be increased at least 6-8 times of the crank force depending upon the length of the lever well with in the operational constraints of the machine.

Notations

Ps → sum of all forces acting on the knife section during harvesting,
Pav → average resistive force to cutting, N
F → friction force, N
F2 → force applied by the connecting rod, N
F1 → friction force between knife section and ledger plate
f → coefficient of friction
p → force developed at the crank wheel due to the ground drive traction,

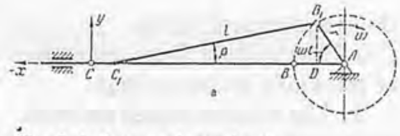


Fig. 3 Inline crank mechanism.

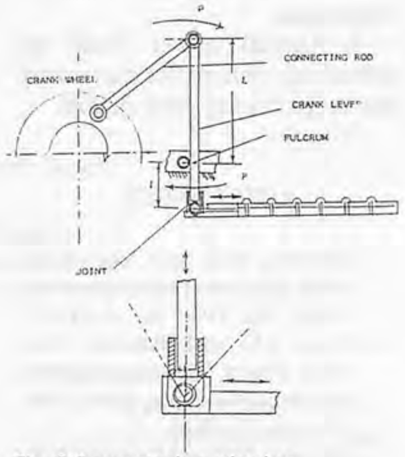


Fig. 5 Lever crank mechanism.

L → long length of the lever at the fulcrum end,
l → short length of the lever at the fulcrum point,
P → large force developed at the knife section,
Gk → weight of the knife section, N
Pj → inertia force at the knife section, N
Mk → mass of the knife section,
jk → acceleration of the knife section, cm/s^2
w → angular velocity, rad/s
x → knife section displacement, cm
r → crank radius, cm
e → crop coefficient
B → inclination angle between the connecting rod and the base
Xc → total displacement of the knife section, cm
Z → number of the cutters,

Conclusions

The new design has the following advantages:

1. The weight of the machine can be reduced as the ground trac-

tion requirement is low.

2. The cutting force available at the knife section is high.

3. The friction caused between knife section and ledger place due to the connecting rod thrust is eliminated.

4. Animal power could be effectively utilized for harvesting the crops during peak period.

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Optimum Replacement Time of Combine Harvesters

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Abstract

To predict the time of combine replacement, three mathematical models, namely; the present worth concept, orthogonal polynomials and compression of discounted worth of the cumulative return of both old and new machines have been taken into consideration. Data regarding the use of three different models of combine on Pantnagar University farm were analyzed. The optimum replacement period of John Deere combines was between 8-9 years (4059-6756 h) whereas that for SKPR-4 combine was between 8 and 10 (3700-4140 h) years.

Introduction

For a farm as large as that of Pantnagar, various machines, including combine harvesters, are needed for timely operations to fetch maximum return. The combine models John Deere 630, John Deere 330 and SKPR-4 on the University farm have deteriorated to a condition quite noticeable in terms of their economy and performance efficiency. The study was, therefore, undertaken to determine the optimum replacement periods of these combines.

Taylor (1923) and Hotelling (1925) used the statistical theory of depreciation to establish the relationship between the average unit cost of the output of a machine over the years and the cost of operation of a new machine. Preinreich (1940) showed that the whole chain of future replacements should invariably be considered in determining the replacement period of a unit in this chain. Shukla and Trishal (1980) collected data for the tractors of Pantnagar University farm and proposed replacement periods for IH-856, IH-706, Ford-7000 and MF1035 tractors through the use of operations research technique employing present worth concept.

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Materials and Method

Combines deteriorate with time during the course of their use. In order to predict the replacement period, data in respect of following combine harvesters of the University farm were collected:

1. John Deere 630
   : 5 combines No. JD-12, JD-13, JD-14, JD-83 and JD-98
2. John Deere 330
   : one combine
3. SKPR-4
   : one combine

The models used for predicting the replacement period are described below:

Present Worth Concept Model

This model envisages the replacement of machine by taking into account the minimization of the present worth of the total capital and running investments to be made for the whole chain of equidistant replacement units. The model is of the following form:

$$C_n - \frac{A + \sum_{k=0}^{n-1} C_k r^k}{\sum_{k=0}^{n-1} r^k} > 0$$

$$> C_{n-1} - \frac{A + \sum_{k=0}^{n-2} C_k r^k}{\sum_{k=0}^{n-1} r^k}$$

where,

A = capital cost of the machine
C = yearly maintenance cost
r = $(1 + i)^{-1}$ = discount rate

i = rate of interest
n = years

The term $\dfrac{A + \sum_{k=0}^{n-1} C_k r^k}{\sum_{k=0}^{n-1} r^k}$ is the weighed average cost of money associated with the expenses incurred on an item if its replacement is planned after n years.

Orthogonal Polynomial Model

This model (Hotelling, 1925) expresses the relationship between time intervals (years) and the corresponding weighted average costs. The general form of the model is as follows:

$$Y(x) = a_o P_{no}(x) + a_l P_{nl}(x) + \dots + a_n P_{nn}(x)$$

where, a_o, a_l and a_n are coefficients, x and Y are the years and weighted average cost as fraction of purchase price, respectively.

$Pn_o(x)$, $Pn_l(x) \dots P_{nn}(x)$ are the orthogonal polynomials, and $n = x - 1$. x is counted from the zero year.

The values of $Pn_o(x)$, $Pn_l(x)$ etc. are calculated from the following general equation:

$$Pnm(x) = \sum_{x=0}^{m} (-1)^j \binom{m}{j} \binom{m+j}{j} (x/n)^j,$$

$$m = 0, 1, 2, 3, \dots n$$
$$j = 0, 1, 2, 3, \dots m$$

and

$$\sum_{x=0}^{n} P^2{}_{nj}(x) = [(n + j + 1)^{j+1}] / [(2j + 1)^{n(j)}]$$

The coefficient $a_0, a_1, \dots a_n$ in the model are calculated as follows:

$$aj = [\sum_{x=0}^{n} P(x) P_{nj}(x)] / [\sum_{x=0}^{n} P_{nj}{}^2(x)]$$

$$j = 0, 1, \dots, m$$

The replacement period is obtained by minimizing the polynominal and equating it to zero.

Compression of Discounted Worth of the Cumulative Return Model

This model takes into account the cumulative return of the old machine (if not replaced) and that of the new machine (challenger) which is proposed for purchase. A new machine to replace the old machine should be purchased only if

$$P(No) + M_t > Q(No)$$

i.e.

$$\sum_{t=1}^{No} E_t r^{(t-1)} + S\,No\,(r)^{(No-1)} - C + M_t > \sum_{t=1}^{No} F_t r^{(t-1)} + M_{No} r^{(No-1)}$$

where,

$P(No) + M_t$ = present values of the total asset
$Q(No)$ = present value of the total asset if no new machine is purchased
$P(N)$ = Present worth
r = rate of interest
E_t = net revenue generated in t^{th} year
S_N = resale value after N years
No = value of N at which P (N) is maximum
C = cost of the machine
F_t = net revenue generated in t^{th} year by the old machine
M_t = resale value of the old machine at t^{th} year.

Data pertaining to the various parameters such as purchase price, working hours, repair costs, POL costs, area harvested and harvesting rate were collected from the University farm records. These data for JD 630 (No. 12, 13 and 14) combines are for the years 1972-82. The data for JD 630 (No. 83 and 98) are for the years 1973-82 while the data for John Deere 330 combine are for the years 1978-1982. The data of SKPR-4 combine are for the years 1971-1981. The data for John Deere 630 (No. 13) combine are presented in **Tables 1, 2** and **3**. Similarly, data for other combines can be presented. Data in respect of resale values during the life of combines (1974 and 1975) and also 1983 were collected from local dealers and combine owners. On the basis of these data, the resale values for other years have been interpolated (**Table 4**). The rates of custom hiring prevailing, during different years of machine use in the area were taken. The rate of interest varied from year to year. However, for simplicity, a constant value of rate of interest (i = 12.5%) prevalent at the time

Table 1 Working Hours and Costs

Year	1972	1974	1976	1978	1980	1982
Working hours	344	398	486	525	496	561
Diesel (litre)	2 727	4 016	5 403	4 207	4 013	5 783
Engine oil (litre)	131	123	117	22	111	132
Other oil (litre)	82	113	151	153	91	107
Annual Charges (Rs.)						
Repair costs	5 840	15 701	23 400	40 364	66 063	97 798
Diesel costs	2 863	4 418	7 402	6 101	8 327	
Engine oil and other oils costs	2 130	2 360	2 680	1 760	2 101	2 868
Total POL cost (Rs)	4 993	6 778	10 082	7 861	10 428	21 663
Total running costs (Rs)	10 833	22 479	43 482	48 225	76 491	119 461

Purchase Price: Rs. 168370.
Machine: Combine Make: John Deere 630 Serial No. JD 6078 (No. 13)
Horse Power: 100 Date of Purchase: April, 1972 year begins: Ist Jan.
Cutter bar size: 3.90 m.

Table 2 Estimated Cumulative Returns of John Deere 630 Combine for Purchase Years of 1980-1

Year		1980	1981	1982	1983	1984	1985	1986	1987	1988
Area harvested (hectare)	(1980)	560	535	549	532	511	486	457	425	—
	(1981)	—	560	555	549	532	519	486	457	425
Harvesting charges (Rs./hectare)		475	475	500	500	525	550	575	600	625
Gross revenue (Rs)	(1980)	266 000	263 340	274 400	266 000	268 170	267 080	262 430	254 640	—
	(1981)	—	266 000	277 200	274 400	279 300	280 950	279 200	273 840	265 250
Running cost (Rs)	(1980)	22 600	39 600	56 600	99 355	118 355	139 255	155 277	176 277	—
	(1981)	—	22 600	39 600	56 600	99 355	118 355	139 355	155 277	176 277
Net revenue (E_t) (Rs)	(1980)	243 400	223 740	217 800	166 645	149 835	127 745	98 135	96 363	—
	(1981)	—	243 400	237 600	217 800	179 945	162 585	139 885	118 563	88 973
Purchase cost (Rs)		714 400	785 000	843 300	901 600	959 900	1 018 200	1 076 500	1 134 800	1 193 100
Resale value (S_N) (Rs)	(1980)	—	—	—	—	639 243	614 477	696 715	584 984	—
	(1981)	—	—	—	—	—	—	649 661	629 032	615 038
r^{t-1}	(1980)	1.000	0.889	0.790	0.702	0.624	0.555	0.493	0.438	0.389
	(1981)	—	1.000	0.889	0.790	0.702	0.624	0.555	0.493	0.438
$E_t.r^{t-1}$	(1980)	243 400	198 905	172 062	116 985	93 497	70 898	48 389	30 381	—
	(1981)	—	243 400	211 226	172 062	126 321	101 453	77 636	58 452	38 970
$\sum_{t=1}^{N} E_t.r^{t-1}$	(1980)	243 400	442 305	614 367	73 135	824 849	895 747	944 136	974 517	—
	(1981)	—	243 400	454 626	626 688	753 003	854 463	932 099	990 551	1 029 521
$S_N.r^{N-1}$	(1980)	—	—	—	—	398 887	341 035	294 180	266 223	—
	(1981)	—	—	—	—	—	—	360 562	310 113	269 387
P (N)	(1980)	—	—	—	—	509 336	522 382	523 916	516 340	—
	(1981)	—	—	—	—	—	—	507 661	515 665	513 908

Life of the combine (challenger), purchased in 1980 = 7 years.
Life of the combine (challenger), purchased in 1981 = 7 years.

Table 3 Estimated Cumulative Returns of John Deere 630 (No. 13) Combine

Year	1980	1981	1982	1983	1984	1985	1986
Area Harvested (ha)	356	305	379	343	336	326	314
Harvesting charges (Rs/ha)	475	475	500	500	525	550	575
Gross revenue (Rs)	69 100	144 590	189 400	171 400	176 400	179 300	180 090
Running cost (Rs)	76 491	142 254	119 461	129 461	141 461	155 461	171 461
Net revenue (Rs) F_t	92 609	2 336	69 936	41 939	34 939	23 839	8 629
r^{t-1}	1.00	0.889	0.79	0.702	0.624	0.555	0.493
$F_t.r^{t-1}$	92 609	2 077	55 252	29 441	21 802	13 231	4 254
$\sum_{t=0}^{N} F_t.r^{t-1}$	92 609	94 686	149 938	179 379	201 181	214 412	218 666

Table 4 Resale Values of John Deere 630 Combine Purchased in 1972

Years	Resale value (Rs)
*1974	250000
*1975	300000
1976	309000
1977	317800
1978	326600
1979	336000
1980	343000
1981	352000
1982	357000
*1983	370000
1984	351500
1985	333900
1986	317720
1987	3011400

* These are the authentic values obtained from various sources. On the basis, other years' resale values have been interpolated.

of analysis for agricultural loans was taken. Since the combine harvesters are tax free, the taxes have not been taken into account in the calculation of the gross revenue. Similarly, the deductions have not been made for these machines as they are not insured.

All the three models described above have been used to predict the period of replacement for John Deere 630 combines. For John Deere 330 and SKPR-4 combines only the first two models have been used. This is because of the fact that the data in respect of resale values of these machines could not be obtained which is necessary in the use of third model, i.e., compression of discounted worth.

Results and Discussion

Studies have been conducted on John Deere 630, John Deere 330 and SKPR-4 combines to predict their periods of replacement. Data in respect of only one John Deere 630 combine (No. 13) has been presented in **Tables 1-3**. However, data for all the combines have been plotted in **Figs. 1** to **4**. Though the basic objective of this study was to predict the periods of replacement of different combines, data for POL consumption, repair cost and overall cost have been collected and analyzed. These data have been used in the replacement models.

Relationship Between Cumulative Hours of Use and Cumulative POL Cost

Figure 1 shows that with an increase in cumulative hours of use the POL cost also increases. This is because of the increase in POL consumption this is due to increase in age and also the rate of POL. The data in respect of John Deere 630 combines for POL costs at different hours of working are slightly different from each other. The consumption figure appears to be on the higher side for the year 1979. This is due to the reason that the engine of this combine was not overhauled before the season during this year. The

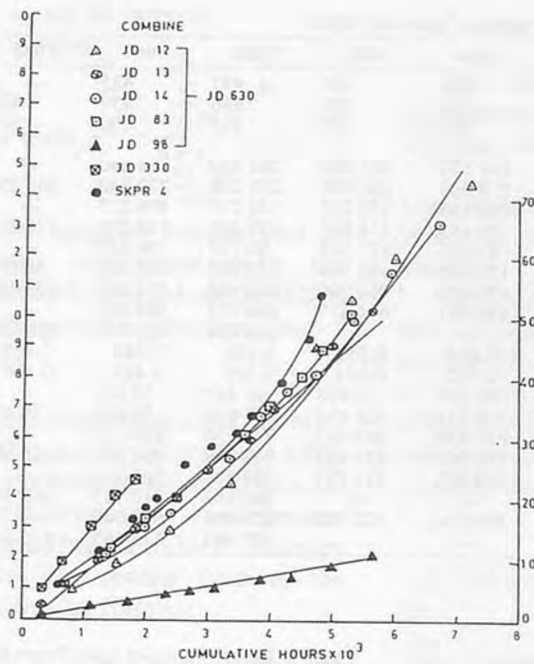

Fig. 1 Cumulative hours and cumulative POL cost.

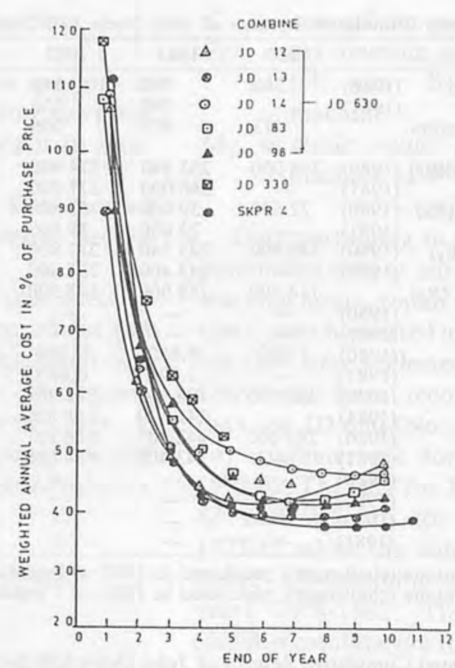

Fig. 2 Weighted average annual cost at end of year.

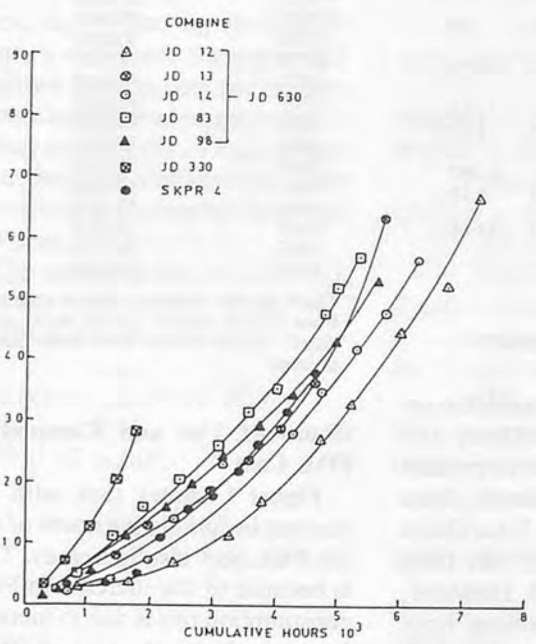

Fig. 3 Cumulative hours and cumulative costs.

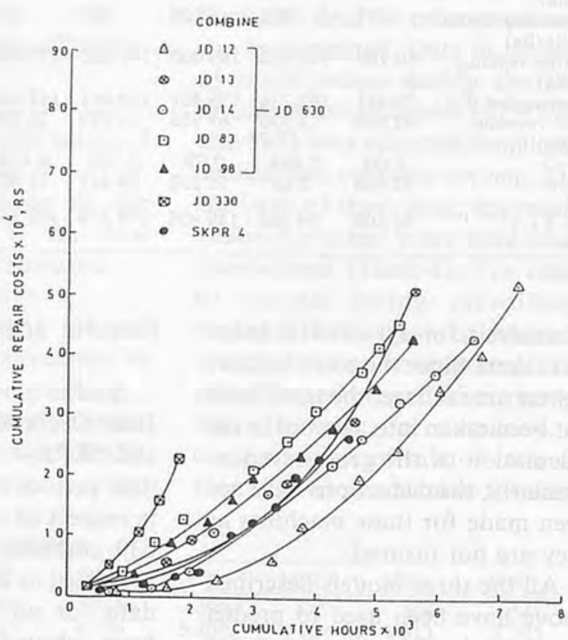

Fig. 4 Cumulative hours and cumulative costs.

maximum POL cost of about Rs. 10.5×10^4 was incurred on John Deere 630 (No. 83) combine whereas the minimum cost of Rs. 8.5×10^4 for the same period has been incurred on John Deere 630 (No. 12 and 14) combines. The highest working hours of 6756 were obtained in the case of John Deere 630 (No. 12) combine. For the similar period of operation, the POL cost in the case of SKPR-4 combine is higher than that of the John Deere 630 combines. If the data from 1978-1982 are compared for all the combines, the cost of POL consumed is the lowest for John Deere 330 combine because of lower hp of the machine. Out of all the combines, the John Deere 630 (No. 12) combine appears to give the best performance.

Relationship Between Cumulative Hours and Repair Costs

The cumulative repair cost increases with an increase in the use of combines (**Fig. 2**). The minimum repair cost has been incurred on John Deere 630 (No. 12) combine whereas the relatively higher repair cost has been incurred on John Deere 630 (No. 83) combine. Comparing the data of repair cost for John Deere 330 combine for the year 1978-82 with the other combines it is seen that the repair cost in John Deere 330 combine is less than that of John Deere 630 combines but more than that of SKPR-4 combines.

Relationship Between Cumulative Hours and Cumulative Cost

The overall cost increases with an increase in the hours of use of combine (**Fig. 3**). The overall cost is the sum of POL cost and repair costs. (The reasons have already been explained in the preceding discussion.) In terms of overall cost also, the John Deere 630 combine (No. 12) appears to be the most economical machine.

Relationship Between Weighted Average Annual Cost and Year of Use

The weighted averae cost in percent of purchase price is greater than 100% at the end of first two years for all the combines except John Deere 630 (No. 13) where the major breakdown had taken place (**Fig. 4**). This is because of inflation which increased the purchase cost of new machine each year and also due to overall less wear of the various components of the machine. However, as the hours of use increase, the combines become old and lose their working efficiency which causes reduction in these weighted annual average costs. The weighted annual average cost almost becomes constant after the fifth year's use.

Replacement of Combines

The combine-wise replacement periods are presented in **Table 5**.

The higher period of replacement of John Deere 630 (No. 12) combine is due to the low overall running cost as compared to other John Deere 630 combines. Further, the performance of John Deere 630 combines was better as compared to John Deere 330 and SKPR-4 combines.

Conclusion

On the basis of analysis of data of the combines at the Pantnagar University farm, the following conclusions are drawn:

1. The optimum replacement periods for all the John Deere combines range between 8-9 years when the combines have covered 4000-6800 h of working life. The replacement period of John Deere 330 combine is also at the end of 8 years.

2. Combine SKPR-4 may be replaced in 8-10 years of its use when it has covered 3700-4140 hours of work.

3. The models used for varoius combines yielded almost the same result. It is, therefore, suggested that the present worth concept model be used to predict the period of replacement as it is simpler compared to the other two models.

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Development of Axial-flow Thresher in Southern Vietnam

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Abstract

The axial-flow thresher (AFT) was introduced to Southern Vietnam in 1974 by the Vietnam Agricultural Machinery Co. (VIKYNO) through fabricating 50 units from IRRI drawings. Since then, farmer mechanics in the Mekong Delta provinces have adapted the principle to local conditions. The main modifications have been: different shapes and arrangement of threshing teeth, and simpler yet effective cleaning systems. It is estimated that in 1988, there were about 50000 units in Vietnam, produced by hundreds of small-scale manufacturers. From the development of AFT in Vietnam, the lessons and experiences learned for promoting a new machine are: (1) the machine should satisfy farmers' real needs; (2) there should be an efficient prototype for farmer-mechanics to imitate and the machine should be compatible with locally available skills and materials, and local land and field conditions; (3) the cost of machine use must be lower than equivalent manual work; and (4) extension activities should be conducted in such a way that people in all provinces engage themselves in fabrication and using the machine.

Acknowledgement: The author wishes to express sincere thanks to Dr. Graeme R. Quick, Head of IRRI Engineering Division for his valuable comments and editing of the manuscript.

Introduction

This paper on the development of the axial-flow threshers (AFT) is confined to Southern Vietnam, especially to the Mekong Delta Provinces where 2.3 million hectares (Mha) are planted to rice (out of a total rice area of 5.7 Mha for the whole country). The Mekong Delta is a flat-land region: the soil is fertile and annually flooded by the Mekong River **(Fig. 1)**. The region is criss-crossed with small rivers and man-made canals; boat navigation is the main method of transporation of goods and supplies. This is a drawback when moving heavy machinery.

The population in this region is 14M, 85 percent living in the countryside. In 1986 the population of Vietnam was 61 M. In terms of the number of inhabitants per rice-planted hectare, the Mekong Delta is lowest (6.1), compared to the national average (10.7) or to the Northern Porvinces (12.8). This low population density provides an impetus to mechanization of the Region as the most advanced in the country. In some provinces of the Region, 90% of land preparation is done by tractor (the country average is 27%). It is in this environment that the axial-flow thresher was promoted and rapidly adopted as an important agricultural machine for the rice farmers—a machine that now handles almost 90% of the threshing work in the Region.

Fig. 1 Mekong Delta region in Vietnam.

Development of the Axial-flow Threshers

Before 1974 in Southern Viet-

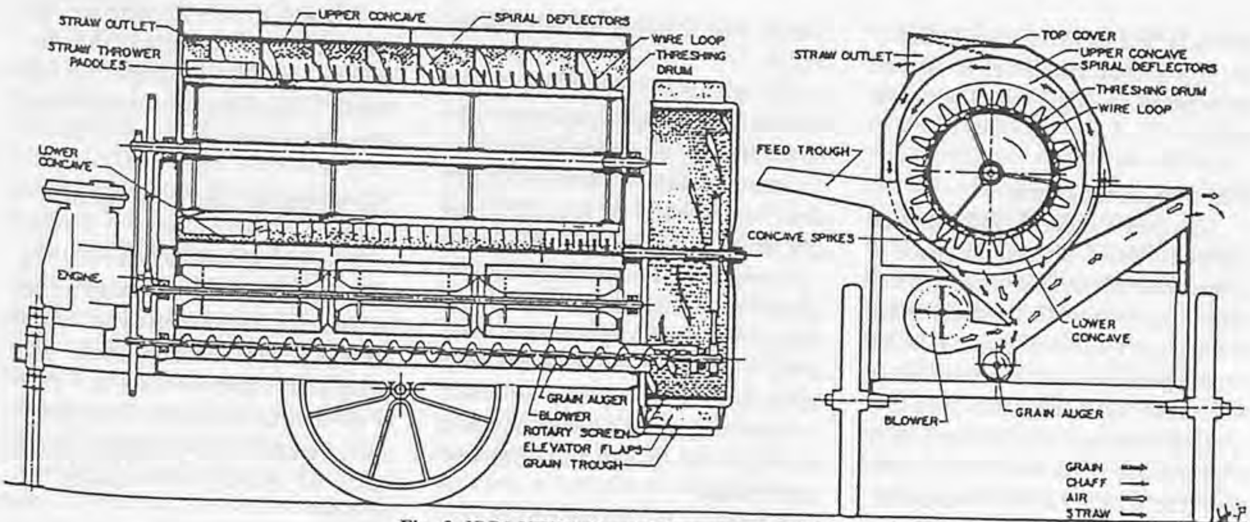

Fig. 2 IRRI-VIKYNO axial-flow thresher (1974).

nam, there were only simple threshers, either of the hold-on type, or of the through-flow type (like the conventional spike-tooth drums on the combine-harvesters). These machines had low efficiency in terms of output (kg/h) or of labour productivity (kg/man-h). A typical 10-HP machine threshed about 400 kg/h with a 5-man crew. The threshing quality is not good, usually with more that 3% unthreshed grains, and a significant amount of threshed grain lost and tangled in the straw.

In 1974, the Vietnam Agricultural Machine Company (VIKYNO) introduced the first axial-flow thresher (AFT), fabricated according to IRRI drawings. Its output was 700kg/h using a 7 HP diesel engine. The machine was tested in 4 provinces of the Mekong Delta. Enthusiasm among farmers was high, some of them wanting to buy the demonstration machine on the spot! So, VIKYNO decided to fabricate the first batch of 50 machines. The company prepared all necessary jigs and fixtures for late mass production.

Between 1975 to 1979, farmer-mechanics in al least 5 provinces began to fabricate the AFT for their own use in their fields, and next for selling to their neighbors. These machines, which essentially looked like the VIKYNO model, were modified as time went on in order to select optimum components under actual use. This period might be called the "Imitation Phase". One point of interest was that for the same threshing output, the selling price of a VIKYNO machine was more than twice the price of a machine built by local farmer-mechanics.

Between 1980 and 1989, there was an "explosion" in the fabrication of AFTs, each district of every province having some metal shops producing AFTs. These threshers are diverse in appearance and performance, and now look quite different from the original IRRI model. This period might be called the "Innovative Phase".

In 1988, the author estimated that there were more than 50000 AFT in the Mekong Delta, coming from hundreds of manufacturers. The machine is also expanding into other region of Vietnam, such as the Eastern Provinces and Central Coastal Provinces. The AFT ranks second in quantity only to the small popular water pump. The pump, AFT, and disk plow are now the three most numerous and important agricultural machines in use in the Meong Delta, far in front of any other powered implement.

Fig. 3 Custom-made Vietnamese axial-flow thresher (1988).

Construction, Modification, and Evolution

The AFT, either an old VIKYNO model **(Fig. 2)**, or a 1988 custom-made model **(Fig. 3)**, basically comprises two assemblies: the threshing drum and the cleaning-conveying system. The operation is as follows:

Rice bundles are laid on the feeding table and hand-fed through the feeding port. Grains are threshed by the drum, and move spirally along the drum axis together with the straw. The grains fall through the concave, while the straw moves to the other end of the drum, where it is ejected by the straw paddles to fall at a distance of 5-10 m away from the machine.

The grain is cleaned by screen and fan, and finally discharged at the bagging-spout.

The axial movement of the

straw is due either to the deflecting louvers on the cover or to the teeth set at an angle with the drum axis.

The Threshing Drum

The first VIKYNO 1974 model followed IRRI pattern and used a drum with 128 cylindrical teeth arranged in 8 rows. Cylindrical teeth are easy to fabricate on a lathe working with round-bar steel (even without a lathe, one can obtain a tooth by cutting off the head of a commercial M10 × 100 bolt. The problem is that 128 teeth together with 128 drilled holes on the drum constitutes a major fabrication cost. Farmer-mechanics, after using first prototypes with cylindrical teeth, began a search for another tooth profile which is simpler to manufacture or requires less of them.

Sometime in 1977 a thresher with flat teeth appeared **(Fig.4)**. The teeth were cut from automotive leaf spring, shaped and bolted to the drum. In early models, the drum was simply made from standard 100mm pipe, and there were as many as 30 teeth on a 1.4 m long drum. The trend is now toward larger drums with fewer teeth. For example: The 1.6 m drum in 1982 had 25-30 teeth, this decreased to 20-25 by 1985, and by 1988 was down to between 11-18 teeth.

The reduction in teeth (with a subsequent reduction in machine weight and cost) is due to an improved tooth profile and tooth arrangement on the drum. The tooth shape and settings are shown in **Fig.4**.

*Angle α (25-30°), the axial-flow angle is for conveying the straw spirally along the drum axis. In some models, α varies from the foot to the head of the tooth, as the tooth is twisted.

*Angle β, the overlapping angle is for continuously transferring the straw from one tooth to the next. The typical value of β is 70°, although some teeth in the central section may have $\beta = 90°$ to retard the flow of straw and improve separation.

*Angle γ, the anti-wrapping angle, causes the straw to slide off the tooth and prevents wrapping, in case the threshing drum slows down. γ may vary from 0-45°, as the tooth is curved.

At present, we can arbitrarily divide the flat-tooth AFT into 3 categories, based on the drum length:

Category	Drum length m	Output ton/h	Power HP	Note
1	0.9-1.2	0.8-1.0	5-7	Can be carried by 2 men
2	1.4-1.8	1.2-1.5	9-12	
3	2.0-2.4	2.0-3.0	13-20	Tractor powered

To the author's knowledge, the "record" for the biggest thresher goes to a new 2.5 m drum that can thresh 8 ha of high yielding rice in a working day.

The Cleaning and Conveying System

The first IRRI-VIKYNO system consisted of a centrifugal blower, an auger conveyor, and a rotary screen. This system was complicated and inefficient. Farmer models consisted of two different systems, now widely used, both of which are equally effective.

a) Axial-flow fan and screen with longitudinal oscillation **(Fig. 5a)**: The screen pan conveys the grain to one end for bagging. The transmission for screen is through simple eccentrics and a 4-bar linkage. The axial fan, driven by belt from the threshing drum, blows away light impurities.

b) Suction centrifugal fan and screen with transverse oscillation **(Fig. 5b)**: The screen oscillates without a vertical amplitude as it is restrained by 4 springs pulling downward. Grains zigzag down the sloping screen, and fall across the suction port of the fan where

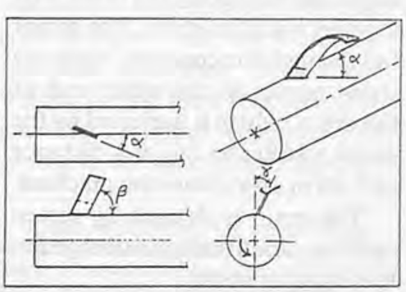

Fig. 4 Flat-tooth shape and setting angles.

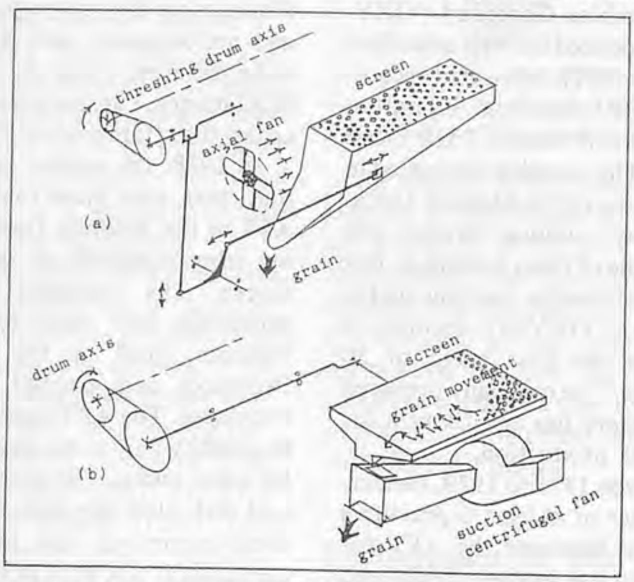

Fig. 5 Cleaning systems on the axial-flow threshers. a) Axial-flow fan and screen with longitudinal oscillation. b) Suction centrifugal fan and screen with transverse oscillation.

light impurities are sucked out.

In both systems, drive mechanisms are simple, using only one or two belts. In fact, the component arrangement is governed by the drive design. Both cleaning systems work well with a cleaning efficiency between 96 and 98%.

Since 1982, two research institutes began to design AFT, using teeth similar to farmer's threshers. The models proved no better than local threshers, notwithstanding minor mechanical defects, due to lack of intensive testing and use.

In 1979 the Faculty of Agricultural Machinery, University of Agriculture and Forestry, Hochiminh City, began constructing AFT based on the IRRI model. In 1981, a larger model was developed and tested with various crops: rice, sorghum, corn, barley, soybean. These crops were successfully threshed with this machine. Output was 1.5t/h for rice, 6.0 t/h for soybeans. While the multicrop use of AFT was a step forward, as far as rice is concerned, this model proved no better than farmer models since it still uses the original cylindrical teeth.

Lessons and Experiences

In less than 10 years (1974-1983), the AFT established itself as a leading agricultural machine in Southern Vietnam and served as a major tool of mechanization. Why this rapid spread of AFT? Who contributed to its rapid development? And how did this process occur? Answering these questions will provide some clues about how to introduce an equipment to mechanize agriculture in developing countries similar to Vietnam. Concerning the birth and growing of the AFT, the author believes it can be traced back to the following reasons:

a) First, there was an urgent need to mechanize the rice threshing operation. This has been an arduous task that requires 20 man-days/ha and constitutes a labor bottleneck, since land preparation following the rice harvest also requires considerable labor.

b) There was a prototype with principles simple enough yet effective in operation, namely, the IRRI model built by VIKYNO. It was important that farmer-mechanics have something to imitate at the outset. Although in this case, the role of the research institutes seems to be dim in the whole process, it cannot be denied that VIKYNO played the role of the research institute in providing the first prootype.

c) There were available materials for fabrication even at the province/district level. The materials were:

*Imported components, available in local markets: single-groove 6200-series ball bearings, sheetmetal. Some farmers even flatten corrugated galvanized steel sheet for use in the sheet metal work.

*Components fabricated in the country: foundry parts (pulleys, bearing supports.), Vee-belts, round steel 6 to 25 mm dia, bolts and nuts.

*Old automotive parts: leaf springs, gearboxes for self-mobile threshers.

d) Fabrication was based on appropriate locally available processes, namely: welding, cold working, minimal machining such as drilling and turning. No milling, shaping, or finish grinding was required.

e) The availability of the prime mover for multiple use. It is estimated that in 1975, there were more than 200,000 small gasoline or diesel engines in the Mekong Delta. They are used for pumping, rice milling and, rotary tilling. The thresher could use the same engine.

Once fabricated, the threshers continued to be used widely, due to the following advantages:

a) It is a high output machine. Its output surpassed any threshing machines ever imported and surpassed, including the original IRRI-designed machine.

b) It is robust, and "trouble free" in the sense that if trouble occurs, the farmer-mechanics is close by, ready to correct faults or supply readily available spare parts.

c) It is compatible with local land and field conditions. The machine can be carried or moved across levees and ditches, or transported by boat into far-away fields.

d) Overall depreciaton in using the machine is low. The engine is for multiple use, thus spreading depreciation cost. The thresher itself is built by low wage workers, using cheap and simple materials. (In 1988, the price of a 1.0-ton/h thresher was about 400 US$, without engine.)

In summary, the above factors, reasons, and features all contribute to this final point:

The cost of threshing by machine (4%) is lower than manual threshing cost (6%) (The percentage number in parenthesis is the threshing cost, as paid by the farmer, compared to the overall value of the threshed grain)

The above "equation" can rarely be solved in the conditions of a country where labor is abundant, and the wage rate is less than 1$/day. It was solved, however, in the case of the AFT. Several contradictory conditions have converged in this simple, low-cost, yet robust, reliable, high-output machine.

Conclusion

The development of the AFT in Southern Vietnam spurs our thinking and discussion about tactics for mechanization in a developing country with difficult conditions like Vietnam. The course of action seems to be:

a) To identify the real needs, the real problems to be solved.

b) To shorten the adoption time, there should be a research institute to release a "workable" prototype that is a fairly efficient machine right at the start. A model should be available for copying by farmer-mechanics or small manufacturers. Afterwards, the Institute should follow up the development of the machine among the various enterprises, metal shops. And assist in its adaptations and modifications. Poor farmers or small manufactures cannot afford patent right, just like they didn't pay the copyright to a research institute. (IRRI charges no royalty for their original designs).

c) Extension activities should be conducted in such a way that people in all the provinces engage themselves in fabricating and using the machine. Finally, it can be said that the work is by the people, and of course, for the people.

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Design and Development of Straight Through Peg Tooth Type Thresher for Paddy

by
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Abstract

In the traditional rice growing areas of India, paddy threshing is mostly done by manual beating and treading with bullocks or tractor. A suitable thresher which can meet the requirements of the paddy farmers of all regions, has become very essential and hence a straight through peg-tooth type thresher (Model IEP-2) has been designed and developed at CIAE-IRRI Industrial Extension Project, Coimbatore, Tamil Nadu. The salient features of the thresher are its threshing efficiency of 99.98%, output of 640 kg/h, cleaning efficiency of 98.0%, low cost, $765 and low power requirement (6 hp). The straw comes out of the thresher with almost full length as required by the farmers. The design details and the performance data of the thresher are discussed in this paper.

Introduction

In the northern parts of India, where wheat is the major crop, about 1.5 million threshers are being used for threshing wheat. In the eastern part of the country, some pedal operated threshers are being used for paddy. But even in the northern areas where wheat threshers are common, paddy is threshed by bullock or tractor treading and by beating manually. In Southern India, where paddy is a major crop, threshing is mostly done by the age old conventional method which is labour intensive. Some multicrop threshers having raspbar type cylinder are being used for threshing paddy in some pockets of South India where the paddy crop can be reasonably dried before threshing. These raspbar type threshers have problems like low threshing efficiency, loss due to unthreshed grain going out with the straw, and choking of threshing cylinder, when used for threshing wet paddy crop without drying. One of the main crop harvesting seasons in South India coincides with the North-East monsoon. The farmers are not able to dry the harvested crop in their fields and the crops remain heaped in stacks in the fields for several days after harvesting due to the absence of a suitable paddy thresher and non-availability of labour. This increases the losses in the quality and quantity of paddy, due to cyclones, floods, rains and rodents. If a suitable paddy thresher is provided for these regions where paddy-paddy rotation is popular, it is expected that the demand for paddy threshers will be as high as that for wheat threshers.

The axial flow threshers (TH-S) were introduced by the CIAE-IRRI Industrial Extension Project, Coimbatore by encouraging manufacturers and by arranging several field demonstrations in the four Southern States of India. The sales of this axial flow thresher remained very much limited because the straw was broken by the thresher into two or three parts which is not acceptable to the farmers of South India and the power requirement was high which increased the total cost of the machine.

To overcome the problem of bruising of straw; and to obtain high capacity of threshing with low power requirement and higher threshing and cleaning efficiencies, a new thresher for paddy (Model IEP-2) was designed and developed at the Central Institute of Agricultural Engineering, Industrial Extension Project, Coimbatore. The new thresher is based on straight flow and peg-tooth type threshing cylinder principles.

Design Considerations

Based on the feedback from the farmers during the various demonstrations conducted with the axial flow threshers, the general requirements of the rice farmers for paddy threshers were assessed as follows:

1. The capacity of the thresher should be about 6-8q/h of grain.

2. The cost should be less than \$900 (without engine) as compared with the cost \$1050 for axial flow thresher and \$1300 for raspbar type multicrop threshers of similar capacity.

3. There should be minimum damage to the straw as the straw has some sale value (20% of grain) if it is in full length.

4. No unthreshed grains should go with straw.

5. The grain coming out of grain outlet should be clean and unbroken.

6. Even wet crop should be threshed in the machine without chocking of the cylinder.

Design and Development

Based on the above design considerations, a straight through peg-tooth type thresher was designed for a capacity to thresh 6-8 quintals of paddy per hour and for a 6 hp Diesel engine as power unit. The design consists of a frame, feeding chute, threshing cylinder, concave, centrifugal blower, oscillating sieves, grain collection box and straw walker sieve.

The various design parameters, namely; the threshing cylinder diameter and speed, concave width and construction, blower speed, straw walker sieve, number of sieves and sieve area, etc were optimised with respect to output, threshing and cleaning efficiencies, breakage of grains and breakage of straw, by functional design methods. A prototype thresher developed was used in the laboratory for optimizing the different machine and crop parameters. The feeding chute was designed as per the Bureau of Indian Standards (IS: 9129, 1979).

Description and Technical Details

The final improved version of the straight through peg-tooth type thresher (Model IEP-2) is shown in **Figs. 1** and **2**. The detailed specifications of the thresher are given in **Table 1**.

The frame (A) is made of 50 × 50 × 8 mm and 40 × 40 × 8 mm MS angles. The body cover (B) is made of 18 g thick MS sheet. The prime mover (C) is mounted on the chassis frame, just below the feeding chute (D). The threshing cylinder (E) is placed just after the feeding chute. The concave clearance is fixed in the front end of the concave (F) and adjustable at the rear end. The cylinder-concave clearance could be varied from 5 mm to 21 mm at rear end. The crop which is fed through the feeding chute is threshed in the cylinder — concave portion. The threshed grain with chaff, impurities and dust are collected at the forward oscillating upper sieve (G). The front portion of the upper sieve below the concave does not have the holes and hence the grain with other materials is moved forward on the sieve. Upon reaching the rear sieve portion, the grain with impurites and chaff starts falling on the lower sieve fitted in the oscillating grain collection box (H). The centrifugal blower (I), which is mounted just below the feeding chute on the vertical frame, sends the blast of air, in between the upper and lower sieves and through the upper sieve. The chaff is blown off through the opening between upper and lower

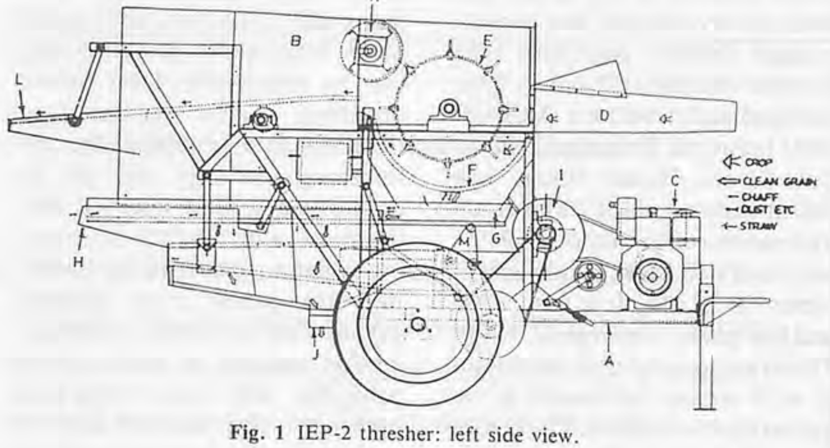

Fig. 1 IEP-2 thresher: left side view.

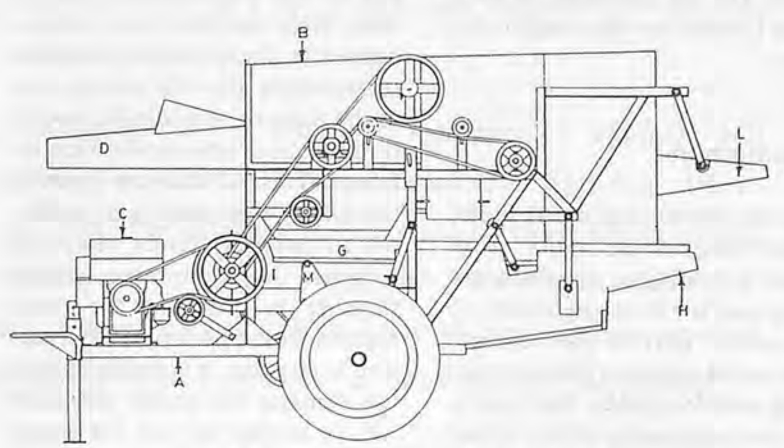

Fig. 2 IEF-2 thresher: right side view.

Table 1 Specifications of IEP-2 Thresher

Type	:	Straight through peg tooth type
Prime mover	:	6 hp Diesel engine (Greaves Lombardini Model - 523, 1800 rpm)
Overall dimensions	:	3300 mm length 1480 mm width 1720 mm height
Weight	:	375 kg
Threshing cylinder	:	a) 600 mm dia × 400 mm width b) 12.7 mm dia m.s. rod peg tooth type c) Speed - 550 rpm
Concave	:	Grate type of m.s. rods and flats Width 500 mm Curved length 940 mm Clearance: Fixed: 40 mm front Adjustable: 5 mm to 21 mm at rear
Beater cylinder	:	Speed - 160 rpm
Blower	:	Number - One Type - Centrifugal type with shutters for adjusting air flow rate Speed - 900 rpm Number of blades - 4
Sieves	:	Number - 2 Length - Upper 1850 mm (1040 mm Sieve portion) 12 mm holes Lower 940 mm - 8 mm holes Oscillation speed - 225 strokes/min Stroke - 35 mm
Straw walker sieve	:	Forward oscillating type - 375 strokes/min Stroke - 25 mm
Transport	:	Two pneumatic wheels and arrangement for hitching to jeeps, tractor or bullock cart.

sieves. The clean grain falls through the lower sieve and is collected in bags from the clean grain outlet (J).

The beater cylinder (K) which is mounted immediately after and just above the level of the threshing cylinder, pushes the threshed straw on to the straw walker sieve (L), fitted above the upper sieve. The straw walker sieve is provided with oscillations and is extended beyond the upper sieve area. The straw is pushed outside through the straw walker. The loose grains going with the straw are collected through the straw walker sieve on the tray below it and fall on the upper sieve.

All the mechanisms and systems in the thresher are driven through V-belts and pulleys. This drive for the blower is taken from the prime mover. The drive for the threshing cylinder is taken from the blower shaft and for the beater cylinder, from the threshing cylinder shaft. The forward and backward oscillating motions for the upper and lower sieves are provided by Bell crank assembly (M) for which the drive is provided by an eccentrric arm fitted on a rotating shaft connected to the blower shaft through belt and pulley drive. The drive for the straw walker assembly is from the threshing cylinder-beater cylinder drive and the forward oscillating motion is provided by an eccentric crank assembly.

The prime mover used for this thresher is a 6 hp Greaves Lombardini Diesel Engine (Model 523 — rpm 1800). The whole thresher was mounted on an axle with pneumatic tyre wheels with provision for towing it by any vehicle, tractor or bullock cart.

Performance

The IEP-2 thresher was intensively field tested for paddy with different varieties at different moisture levels. The performance tests were carried out as per the Bureau of Indian Standards (ISI) — Indian Standard Test Code for Stationary power thresher for wheat (IS: 6284, 1975), comprising of one hour duration and long-duration tests. Samples were collected at regular intervals: (i) to determine grain losses from different outlets namely, straw outlet and chaff outlet; (ii) to determine the chaff and foreign matter coming out of clean grain outlet; (iii) to determine breakage of grains; and (iv) to determine the damage to the straw. Fuel consumption was measured by actually measuring the fuel at the time of start and close of trials. The grain-straw ratio and the moisture content of grain and straw were also determined. The average feed rate and average output of each test were recorded. The performance data is presented in **Table 2 (A)**.

It can be seen from **Table 2 (A)** that the average output was about 637 kg/h of grain and the threshing efficiency was 99.95%. There was no breakage of grain in all these tests. The average sieve-blower loss was 0.74% and this loss was actually collectable as these grains were collected just below the chaff outlet. The average cleaning efficiency was 94.8% and the average fuel consumption was 0.95 l/h.

Initially, the thresher was not provided with straw-walker sieve. It was found that, since the threshed straw was pushed over the upper sieve, the cleaning efficiency could not be improved beyond 95%. To achieve better cleaning efficiency, the straw walker arrangement was provided over the upper sieve. The results are presented in **Table 2 (B)** showing that the cleaning efficiency of the thresher increased to 97.8% by providing the straw-walker arrangement but had increased the machine cost additionally by $35 only.

The straw length analysis before and after threshing in the IEP-2 thresher for paddy crop is

Table 2 (A) Performance of IEP-2 Thresher for Different Varieties of Paddy

Variety	No. of Trials	Grain: Straw ratio	M.C. % of Grain (wet basis)	M.C. % of Straw (wet basis)	Feed rate kg/h	Output kg/hr	% of Broken	% of Unthreshed	% of Sieve-blower loss (collectable)	Threshing efficiency (%)	Cleaning efficiency (%)	Fuel consumption l/h
IR-20	3	1:2.70	16.5	33.4	2100	560	Nil	0.02	0.63	99.97	94.3	0.95
		1:2.80	25.0	52.5	3 000	780	Nil	0.03	1.20	99.98	95.1	0.97
IR-50	1	1:1.8	—	—	2 700	960	Nil	0.03	0.48	99.97	95.2	1.00
IR-50	2	1:2.2	19.0	34.0	2 000	620	Nil	0.05	0.72	99.95	94.8	0.95
IR-60	2	1:2.6	25.0	55.8	1 920	540	Nil	0.02	0.59	99.91	94.9	0.94
		1:3.2	21.0	74.4	—	570	Nil	0.03	0.80	99.98	95.0	0.92
Vaigai	1	1:1.5	—	—	1 800	575	Nil	0.07	0.72	99.93	94.3	0.92
Ponni	1	1:3.2	23.8	64.6	1 800	490	Nil	0.06	0.76	99.94	94.4	0.92
Average						637	Nil	0.04	0.74	99.95	94.8	0.95

Table 2 (B) Performance of Thresher with Straw Walker Arrangement

Variety	No. of Trials	Grain: Straw ratio	M.C. % of Grain (wet basis)	M.C. % of Straw (wet basis)	Feed rate kg/h	Output kg/hr	% of Broken	% of Unthreshed	% of Sieve-blower loss (collectable)	Threshing efficiency (%)	Cleaning efficiency (%)	Fuel consumption l/h
IR-50	1	1:1.5	17.8	33.0	1 750	690	Nil	0.01	0.48	99.99	97.5	1.00
IR-50	1	1:1.6	17.0	32.5	1 550	595	Nil	0.02	0.31	99.98	98.0	1.00
IR-60	1	1:2.5	21.0	41.0	1 900	635	Nil	0.02	0.30	99.98	98.0	1.00
Average						640	Nil	0.02	0.36	99.98	97.8	1.00

given in **Table 3** showing that the damage to the paddy straw length was only about 9.6% in the thresher.

Table 3 Straw Length Analysis in IEP-2 Thresher*

	Straw length, cm					
	below 21	21-30	31-40	41-50	51-60	61-70
Before threshing						
Total No. 707						
Number	—	6	99	304	284	14
Mean length, cm	—	30	37.34	47.00	57.99	63.93
% of total number	—	0.85	14.00	42.99	40.17	1.98
After threshing						
Total No. 775						
Number	50	159	234	276	65	—
Mean length, cm	26	27.39	37.41	47.17	54.28	—
Percent of total number	6.45	20.52	30.20	34.45	8.39	—
Percent straw cutting			9.6%			

*Variety: IR-50. Cocave clearance: 7 mm at rear.

Conclusion

The IEP-2 thresher has many distinct advantages over the axial flow threshers and the raspbar type multicrop threshers for threshing paddy crop as follows:

1. Even the wet crop with moisture content of grain and straw as high as 25% and 74%, respectively, could be handled by the thresher without any choking and with very high threshing efficiency of 99.98%.

2. The capacity of the thresher is high in the range of 5.5 to 9.6 quintals per hour of grain output, (Average 6.4 q/h).

3. The threshing and cleaning efficiencies were 99.98% and 97.8%, respectively.

4. There was no breakage of grains and sieve blower loss was 0.36%.

5. The straw length analysis has shown that there is only minimum damage to the straw. The length of straw obtained after threshing in the IEP-2 thresher was acceptable to farmers.

6. The 6 hp Greaves Lombardini Diesel engine, Model 523 has adequate power for IEP-2 thresher. Same engine can be used in the self-propelled vertical conveyor reaper developed by the CIAE-IRRI Industrial Extension Project and hence, a farmer can own both a self-propelled reaper and a IEP-2 thresher with one engine.

7. The power requirement per quintal of grain threshed was about 1.0 hp in the IEP-2 thresher while it is around 2.0 hp in the case of TH-8 axial flow thresher.

8. The command area of the thresher is 200 acres of paddy area in a year which can be easily commanded in the same village by even a small farmer by custom hiring. Therefore, the thresher will be a profitable proposition even to small farmers.

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Effects of Different Clearances between Two Rubber Rolls on Dehusking of Paddy

by

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Abstract

Using a Satake Grain Testing Dehusker, a series of dehusking tests were carried out with three different Indian varieties of paddy, Panke (short and round); T-141 (medium and bold); and Lalat (long and slender) at four levels of clearances between the rolls (20%, 30%, 40% and 50%) of paddy thickness. Evaluation of results was made in terms of hulling index. A clearance of 20% to 30% of paddy thickness is recommended for maximum hulling.

Introduction

The purpose of dehusking is to remove husk from paddy without damage to the bran layer and rice kernel. In the process of dehusking, the action of tension, compression and friction are applied to the paddy (Araullo et al, 1976). Because of the structure of the paddy grain, there is some percentage of broken rice which cannot be avoided, but efforts have to be made to minimize this broken percentage. If the paddy cracks during dehusking or in the field due to sun drying or other reasons, then brokens in the dehusking cannot be avoided. The percentage of head rice is a major criterion in rice milling. The yields of head rice, broken rice, whole plus broken rice or total recovery of milled rice depend on the variety, purity, percentage of mature kernels, moisture content, percent crack, duration of storage (Chakraverty and Ojha, 1975), milling techniques, premilling procedures, i.e., harvesting, cleaning and drying methods and degree of polishing (Bhattacharya, 1980). The maximum compressive strength can be resisted by paddy grain at a moisture content of 14 to 16% wet basis (Chakraverty and Ojha, 1975). In dehusking, there are many types of machines which can be used. According to make and design, different machines have different dehusking methods. In India more than 78 000 hullers are used in dehusking as well as polishing of paddy. The average yield of rice for the huller is 62-64% with 25-30% brokens giving only 50% head yield (Indudharaswamy and Bhattacharya, 1979). Thus the Government of India is trying to popularize rubber roll shellers for dehusking paddy by establishing Regional Extension Service Centers (rice milling) in paddy growing states. The main objective of these centres is to replace or modernize the traditional hullers by rubber roll shellers for extra yield.

In using this types of machine, like the disc sheller type, the grain thickness and the machine's clearance between the rolls through which the paddy is dehusked, must have certain ratio relation so as to perform the dehusking well with higher percentage of hulling index (Anonymous, 1981).

In this study, three different varieties of paddy, Panke (SR), T-141 (MB), and Lalat (LS) which are very popular in Orissa, were selected for dehusking in a Japanese-type laboratory model rubber roll sheller at four levels of clearances (20%, 30%, 40% and 50%) of paddy thickness to determine the effects of these clearances on percentage of unhulled paddy, whole rice, broken rice and hulling index.

Materials and Methods

Definition of Analytical Terms Used

Unhulled paddy: rice grain which is completely or partially covered by husk

Whole rice: whole or nearly whole kernel (more than 3/4 size) without any husk covering

Broken rice: less than 3/4 size of the whole kernal

Mealy waste: waste materials other than husk obtained at the outlet

The Laboratory Model Rubber Dehusker

A Satake Grain Testing Dehusker (THU 35A Model of Satake Engg. Co. Ltd., Tokyo) was used for the experiments. It consists of two rubber rolls having 100 mm diameter and 35 mm face width, rotating in opposite directions with different speeds. The speed of the faster roll was 1900 rpm and that of slower roll 1000 rpm. The slow moving rubber roll was adjustable laterally in order to increase or decrease the clearance between the two rolls. The other important components were hopper with a shutter to control the flow rate, a husk aspirator and paddy cleaner. The model was operated with a single-phase built-in motor.

Paddy Samples

Rice varieties Panke, T-141 and Lalat were collected by hand stripping from experimental farm of O. U. A. T., cleaned properly and dried to about 15% m. c. (wb) using drying air at room temperature. The samples were then stored at room temperature in plastic containers containing naphthalene until tested.

Methods of Measurement

The moisture content (m. c.) of the paddy samples were determined by air oven method in which 10 g pf paddy samples were dried at 105° for 24 h. Dimensions of paddy and brown rice were determined by making physical measurements of lenght, width and thickness by means of a micrometer having least count of 0.01 mm. For this 15 grains were randomly selected from each sample and their dimensions were determined by taking the average of 15 measurements. The samples were classified as per classification by the Food and Agricultural Organization of the United Nations, 1972.

Test Procedure

The clearance between two rubber rolls were adjusted at 20%, 30%, 40% and 50% of paddy thickness by using a filler gauge. After necessary adjustments, the rubber roll sheller was connected to the power source. The 50 g of cleaned and sound paddy sample was then fed to the hopper of the sheller. Three replications were made at each clearance between the rubber rolls. After dehusking, separation of unhulled paddy, whole rice, broken rice and mealy waste were done by hand sorting method and then the weight of each component was determined. The hulling index for each sample dehusked at different clearances were determined by the following formula (Anonymous, 1981).

Hulling Index

$$= 100\left(1 - \frac{W_1}{W_2}\right)\left(\frac{W_3}{W_3 + W_4 + W_5}\right)$$

where,

W_1 = weight of unhusked paddy in the product
W_2 = weight of paddy in the feed
W_3 = weight of whole rice
W_4 = weight of broken rice
W_5 = weight of mealy waste in the product

Results and Discussion

The characteristic of paddy samples used in the experiments are shown in **Table 1**. There was no significant variation in the m.c. of different samples. Pre-milling procedure, i. e., harvesting, drying and duration of storage, etc. were similar for all the samples and only cleaned and matured grains were selected for the experiments. Therefore, it was presumed that they did not influence the results.

Fig. 1 shows the effects of clearances between two rubber rolls on percentage of unhulled paddy and whole rice. The percentage of unhusked paddy was found minimum at a clearance of 20 to 30% of paddy thickness and thereafter it increased with increase in clearance between the rubber rolls. Beyond 30% clearance of paddy thickness sharp increase in unhulled paddy was noticed for all the varieties. The percentage of unhulled paddy was minimum for MB variety, followed by SR and LS varieties at all the clearance except 20%. The trend of the result shows that the stripping action was more pronounced on medium and bold variety than short and round variety. The stripping action was least on long and slender variety resulting in quantity of unhulled paddy.

The maximum percentage of whole rice for SR, MB and LS varieties were 67.3%, 65.5% and 61.7%, respectively, at a clearance of 20% of paddy thickness. No significant variation in whole rice was noticed by increasing the clearance from 20% to 30%. Beyond 30% clearance there was a sharp decrease in percentage of whole rice with an increase in clearance between rubber rolls. The maximum percentage of whole rice was shown by the SR variety followed by MB and LS varieties. This may be due to better resistance of short and round variety against tensile and compressive stresses.

Fig. 2 shows the effects of

Table 1 Characteristics of Paddy Samples

Variety	Moisture content (% wb)	Thickness of paddy (mm)	Parameters of whole brown rice		
			Length (mm)	Length/width	Type
Panke	15.6	1.8	3.88	1.68	short and round (SR)
T-141	15.45	1.91	5.86	2.56	medium and bold (MB)
Lalat	15.54	1.95	7.00	3.01	long and slender (LS)

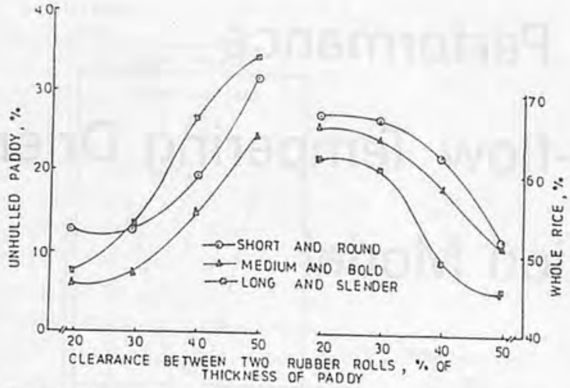

Fig. 1 Effect of clearance between two rubber rolls on unhulled paddy/whole rice.

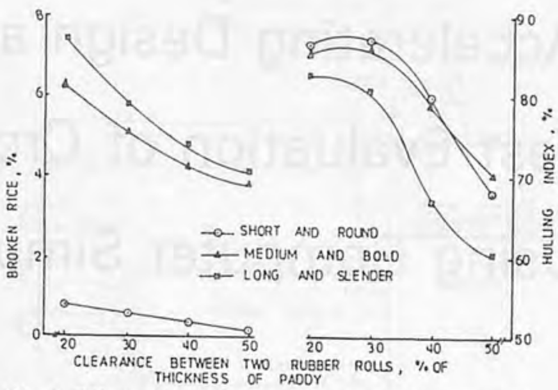

Fig. 2 Effect of clearance between two rubber rolls on broken rice/hulling index.

clearances between two rubber rolls on percentage of broken rice and hulling index. In general, the percentage of broken rice decreased with an increase in clearance between two rubber rolls. The SR variety produced the least percentage of broken rice (less than 1%) followed by MB and LS varieties.

The maximum percentage of broken rice for SR, MB and LS varieties were 0.17%, 7.02% and 7.43%, respectively, at 20% clearance which are much less than that generally obtained in the case of hullers.

The maximum percentage of hulling index for SR and MB varieties was 86.7% and 86%, respectively, at a clearance of 30% and for LS variety it was 82.6% at a clearance of 20%. No significant variations in percentage of hulling index was noticed between 20% and 30% clearance. A sharp decrease in the percentage of hulling index was found in all the varieties beyond 30% of clearance. This may be due to sharp increase in percentage of unhulled paddy and decrease in percentage of whole rice beyond 30% clearance. Though the percentage of brokens decreased with an increase in clearance beyond 30% **(Fig. 2)**, they did not put much effect on percentage of hulling index in comparison to the percentage of hulled paddy and whole rice.

Conclusion

A clearance of 20 to 30% of paddy thickness between two rubber rolls may be recommended for maximum hulling indices of SR, MB and LS varieties of paddy.

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Accelerating Design and Performance Test Evaluation of Cross-flow Tempering Drier Using Computer Simulation Model

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Abstract

The use of computer simulation has broad applications in agricultural engineering, especially in design and performance test. Improvements in agricultural machinery performance are being realized through the use of facts obtained during simulation runs. One aspect of research efforts in Ehime University is lowering grain processing costs through development of more efficient grain dryers and the most profitable grain drying management practices using computer simulation model.

The main purpose of this paper is to present some of the results of a study by computer simulation of the performance of a simple cross-flow tempering drier (a drier in which a moving stream of grain is successively dried by heat air flow at right angle for a short period time, then moves on to the tempering tank, where the stress developed in the kernel is relieved).

Introduction

Artifical drying of agricultural products is one of the most common methods of preservation. Proper drying procedures eliminate the potential of spoilage during subsequent storage and help to improve the quality of the product. It would be of great benefit if heated air driers could be designed and managed in ways to reduce or minimize drying damage to agriculturl products. Understanding and controlling the drying process is very important step in establishing improved design guidelines for drying systems.

The use of computer simulation technique is a very powerful tool in analyzing and explaining the complicated, coupled phenomena of heat and mass transfer and the drying principle of biomaterials. Computer simulation models also help to predict the performance of a drier for evaluating new drier designs, drying efficiency and energy use.

For such a study, computer simulation has three main advantages: (a) A saving of several orders of magnitude in time and cost; (b) Important variables, which in practice are subject to uncontrollable variations, can be held constant. Also, when new truths are leared about the drying process, it would be a simple matter to incorporate them into the simulation model; and (c) Changes can be calculated in those quantities, grain temperature, for example, which are either not easy or impossible to measure by experiment.

According to Chandler (1979) current harvested grain losses of about 30% are apparently occurring throughout large areas of the world, particularly in the tropics and subtropics. These losses occur due to improper post harvest handling of the grains. There are two main methods of drying: natural and artifical drying. The common practice in most areas of the world particularly in the third world countries is sun drying which has a lot of shortcomings. Artifical drying with forced heated air not only reduces the time taken for drying but also gives better milling quality, by eliminating the "sun-checking" or "sun-cracking" effect of sun drying.

The need for artifical drying today is strongly felt both in developed and developing countries due to the second generation

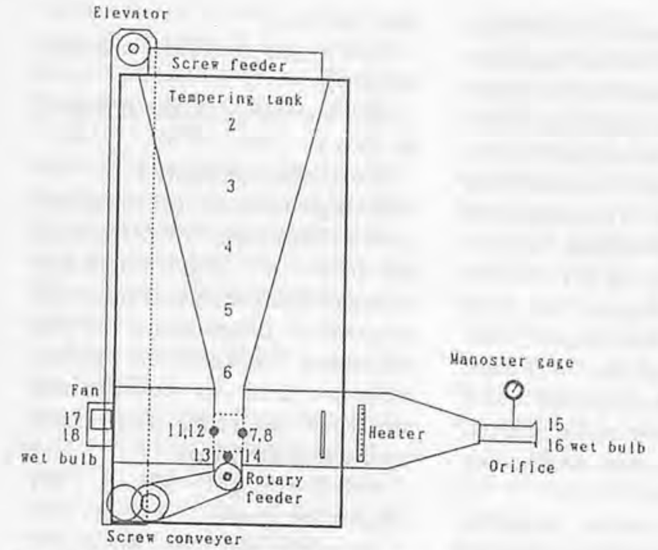

Fig. 1 Diagram of the tempering cross-flow dryer.

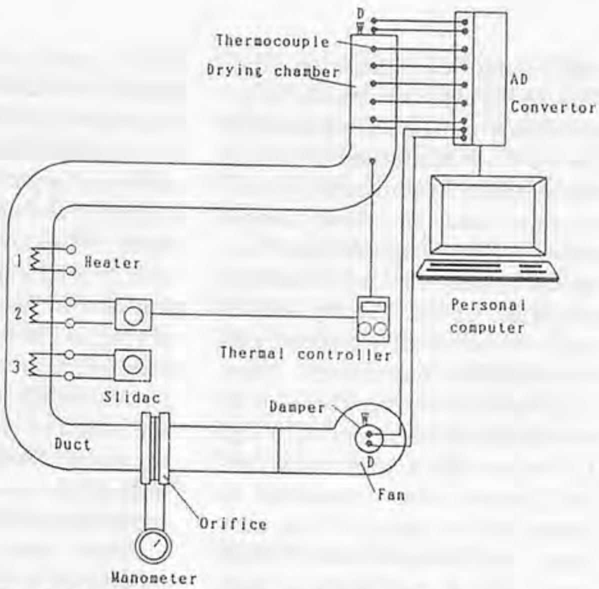

Fig. 2 Diagram of an experimental deep bed dryer.

problems created by the green revolution and new crop production technology, e.g., rainy season harvest of high yield varieties of most grains or multiple cropping in main rice producing areas of the world. For this reason it is obvious that wide-spread introduction of grain driers cannot be avoided in the near future in many countries, but to be sucessful the drier would have to be economical in its capital and running costs. To achieve this goal, driers must be designed taking full consideration of costs, local environmental conditions, and skill and materials available.

In this study a basic digital simulation approach was adopted to predict grain drying under different sets of drying conditions. Drying is a continuous porcess with changes in moisture content, air, grain temperature and humidity of the air all occurring simultaneously, these changes vary by different drying methods and different locations in the drying bed. The objective of this study is to develop a computer simulation procedure whereby grain drying prediction could be made with infinitely many sets of drying conditions. The simulation model incorporates many of the factors that affect grain drying and is capable of determining the effect of many drying parameters of the drying result. The effect of drying parameters on the crack ratio of the dried rain was also investigated in order to determine the most appropriate drying technique for grain.

Equipment and Simulation Procedure

The drying system for the cross-flow drier consist of an air blower, calibrated orifice for air flow measurement, psychrometer, drying chamber, thermal control unit, heater, bucket elevator assisted by screw conveyer, screw and rotary feeders and a tempering tank as shown in **Fig. 1**.

The drying chamber has length of 0.91m, width of 0.15m, and depth of 0.11m, an area of $0.1366m^2$ and a total volume of $0.01502m^2$. The column of grain in the drying chamber was considered to consist of 19 layers of rice each being 6 mm deep. A series of hole was drilled and thermocouple probes inserted in the configuration shown in **Fig. 1**. To monitor the temperature of the grain layers and the air temperature at different locations in the drier, thermocouple probes were connected to an interface analog digital converter (Green kit 77A model) then to a personal computer for data collection and analysis. A 100-kg sample of freshly harvested medium grain rough rice of Japonica specie at known moisture content stored at 5°C was used for each of the trials. Diferent trials were run in this manner for periods of time ranging from 8 to 32h during the summer month of August and the autumn month of October. The temperature of each location in the drier was recorded every 5 min by the data acquisition unit, and 10-g sample of rice were also withdrawn every hour for moisture determination by the use of single grain moisture meter. At the end of each trial, the dried rough rice was withdrawn and samples were analyzed for cracking by inspecting them under fluorescent light. The number of cracked grains per given sample was then used to calculate the crack ratio for the particular batch of grain.

An experimental drier shown in **Fig. 2** was built and used for the static deep bed simulation trial. The drying system consisted of an air blower unit, calibrated orifice for air flow measurement, a drying chamber, heating section, temperature and air flow control section. These sections were connected by polyvinyl pipes with the exception of the heating section which is made of insulated galvanized sheet metal duct. Thermocouple probes were inserted in the configuration as shown in **Fig. 2**. The column of grain in the drying chamber was considered to consist of five layers of rice each being 5 cm deep. A series of holes were drilled at mid-points of each layer and thermocouple probes inserted. To monitor the average temperature of these individual layers, thermocouple probes were connected to a digital analog converter then to a personal computer for data collection and analysis. The drying chamber was filled with the same type of rough rice as was used for the cross-flow drier trial. Different trials were run for temperature ranging from 40 to 60°C and air flow 0.1 to 0.45 m^3/min. The temperature of each layer was recorded automatically every 5 min. At the end of each trial the moisture content of the entire layer was determined by the over method for 24 h at 135°C (JSAM).

Simulation Method and Theory

Fig. 3 shows a simplified flow chart of the simulation of a cross-flow drier. A method of computer simulation of the performance of two kinds of drier for grain is proposed. It is based on the thin layer drying rate of the grain, heat and mass balance in the drier and the rate of heat transfer from the heated air to the grain. Many researchers in the past have used thin layer equation to predict drying of grain in deep bed. For this study, thin layer drying equation was used as a basis for the simulation of grain drying system. The drying of grain in deep bed can be taken as the sum of several thin layers. Thin layer drying equation can be used in predicting the drying rate of grains as affected by drying air temperature, air flow rate, initial moisture content, relative humidity and also to predict the length of time required to dry the grain to require moisture level, both in the static and continuous cross-flow driers.

Thin layer drying equation based on a single-term solution of the diffusion equation in spherical coordinates is a very common way of calculating the drying rates of grains, (Brooker et al., 1981). This single term solution is a good approximation of the diffusion series as it converges rapidly. It is mathematically represented as:

$$M_r = (6/n^2)e^{-kt} \qquad (1)$$

where M_r = moisture ratio = $(M - M_e)/(M_i - M_e)$

M_e = equilibrium moisture content, % (d.b.)

M_i = initial moisture content, % (d.b.)

M = moisture content at time t, % (d.b.)

k = drying constant, s^{-1}

This single-term solution is analogous to Newton's law of cooling (Brooker et al., 1981), where it is assumed that the rate of moisture removal is proportional to the difference between the kernel moisture and its equilibrium moisture content. Expressed mathematically, it is:

$$dM/dt = -k(M - M_e) \qquad (2)$$

Integration gives

$$M_r = (M - M_e)/(M_i - M_e) = A{*}e^{-kt} \qquad (3)$$

Drying Rate Equation

The drying rate of most agricultural products can be simple related to the moisture content, temperature of the material, ambient temperature and humidity of the drying air. One such relationship which is often found to hold for agricultural materials is equation (2) written in the form

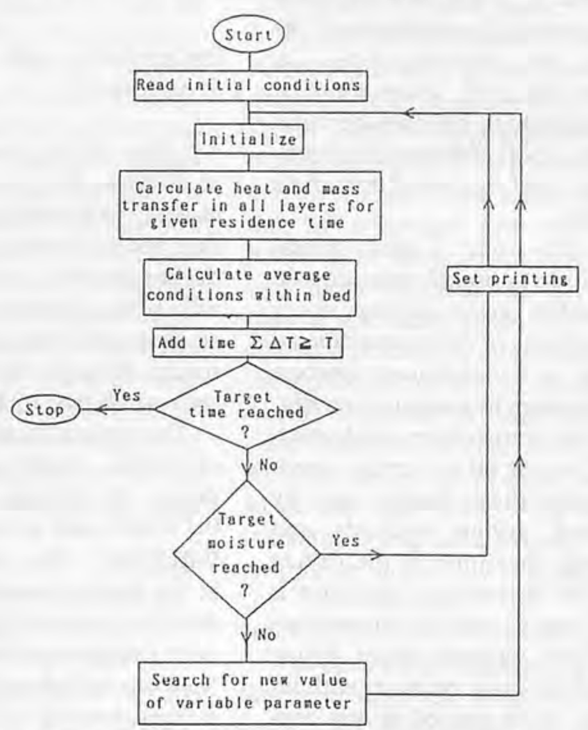

Fig. 3 Simplified flow chart of the simulation of a cross-flow dryer.

$-dM/dt = k(M - M_e)$. After an interval of time dt the moisture content of the layer has changed from M to $M_i = M + dM$, whence

$dM = -k(M - M_e)dt$

where M is assumed to be constant for the purpose of calculating dM, if the average moisture content over the time interval is taken as $(M - dM)/2$ then euqation (2) becomes:

$$dM = -k(M - M_e)dt/(1 + .5k*dt) \quad (4)$$

Equation (4) describes the rate of drying of and contains two coefficients k, and M_e which in their turns, are functions of drying air temperature and humidity.

The values of k and M_e used for this study were those of Abe and Ofoche:

$K_1 = 2.236 * 10^{\wedge}2 * \exp(-1.597*10^{\wedge}3/T + 273.16)$

$K_2 = 3.229 * 10^{\wedge}3 * \exp(-2.888*10^{\wedge}3/T + 273.16)$

$\ln Rh = a*e^{bMe} * \ln PWSTA + c*e^{dMe}$

where $a = 0.68$ $b = -0.114$

$c = -7.63$ $d = -0.144$

for paddy, these values have been found to give good agreement results with those obtained by drying paddy in a cross-flow drier and static deep bed drier.

Mass Balance Equation

Taking a mass balance for a layer over a time interval dt in terms of the moisture;

Moisture lost by material = moisture gained by air

$$AR*dz*PD*dM = AR*G*dt(X^i_e - X_e) \quad (5)$$

whence $dx_e = X^i_e - X_e = PD*dz*dM/G*dt \quad (6)$

Heat Balance Equation

Taking a heat balance equation for the layer over the time interval dt, with 0°C as the reference temperature for enthalpy:

Energy lost by air = Energy gained by material.

$$AR*G*dt(h = h^i) = AR*PD*dz((Cpd + M^i*Cpw)(TG^i - TG) + dM*Cpw*TG^i) \quad (7)$$

The specific enthalpy of the air-vapour mixture is expressed as:

$$h = ha + Xa*h_v = Cp_e*Ta + X_e(Cpw_v*Ta + Lpw_v) \quad (8)$$

where h_v is the value of the specific enthalpy of the vapour and Lpw_v the steamtable value of the specific enthalpy of the vapour, at temperature Ta, h_v approximates 2501000 + 1820Ta joules per kg of water where Ta is expressed in C, then equation (8) becomes:

$$h = 1005*Ta + X_a(1820*Ta + 2501000) \quad (9)$$

If $F = -dx*PD/G*dt$

Then,

$$Ta^i(1005 + 1820*X_a^i) = Ta(1005 + 1820*X_e) + TG^i(F(Cpd + M^i*Cpw)) + TG(-F(Cpd + (M^i - dM)Cpw)) + 25010000(X_a - X_a - X_a^i) \quad (10)$$

Heat Transfer Equation

The heat transfer equation describes what happens to the heat that is transferred from the air to the material. This heat both raises the temperature of the grain and the water remaining in it and evaporates the water that is transferred and raises its temperature to Ta^i, this is represented by:

$$HCAS*AR*dz*dt((Ta + Ta^i)/2 - (TG + TG^i)/2) = AR*dz*PD*((Cpd + M^i*Cpw)(TG^i - TG) + (-dM)(h_v^i - Cpw*TG)) \quad (11)$$

The enthalpy of water vapour at temperature Ta^i is given by

$$h_{vi} = 1820*Ta^i + L_{pwv} \quad (12)$$

Substituting and letting

$D^i = 2*PD/HCAS*dt$

$$(1 + 1820*D^i*dM)TG^i = (D^i(Cpd + M^i*Cpw) + 1)TG^i + D^i(Cpd + (M^i - dM)Cpw)TG - D^i*dM*L_{pwv} - Ta^i \quad (13)$$

Use of the Equations

Consider the drying of a thin layer of material, if it is sufficiently thin, the properties of the material can be regarded as constant within the layer. Consider 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. The change in the moisture content over the time interval dt, can be calculated from Eqn. (4).

The change in the humidity of the air after it has been passing through the layers for a time dt, can be calculated from Eqn (6).

The change in temperature of the air and of the material can be calculated from the two simultaneous Eqn (10) and (13).

Simulation Procedure

Static Bed Drier

In order to predict the one dimensional through drying of a deep bed of grain accurately, it is necessary to divide it into layers that are sufficiently thin for the application of the thin layer equations, i.e., the properties of the material must be constant, or nearly so within the layer. The time interval must be sufficiently short for air condition to be constant, at the inlet to and at the exit from the layer. Both the number of layers and number of time intervals, however, must be kept as small as possible, so that the calculation time may be of a reasonable length. Having selected values of dt and dz and determined the physical properties of the material, the calculation are done as follows: The temperature and humidity of the air entering the first layer at time $t = 0$ are those of the drying air, Ta and X_a. The temperature and moisture content of the material in the first layer are initially TG and M_i, respectively, in all layers. The change in moisture content in the first layer during the first time interval dt is calculated from Eqn (4) values of K and M_a are calculated for Ta and X_a. The changes in humidity of the air is calculated from Eqn (6). The resulting humidity is that of the air leaving the first layer and entering the second layer. The changes in the temperature of the air and the material are calculated from Eqn (10) and (13). The resulting air temperature is that of the air entering the second layer.

The calculations are repeated for all the other layers in the bed, until the last layer is reached. If a relative humidity higher than the equilibrium value corresponding to the moisture content and temperature of the material 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 a lower temperature, its relative humidity will rise as it is cooled at constant absolute humidity. When 100% relative humidity is reached the mass transfer process is reversed, water being deposited on the material with release of latent heat, and the temperature of the material is raised. The state of that bed at 1 dt is now known. The calculation process is repeated starting again with air having properties Ta and X_a at entry to the modified first layer. Each time the calculations are carried out the properties of the air and the material are altered, and drying of the bed is thus simulated. The calculations can be stopped when any desired condition, for example, average bed moisture content or total drying time is reached.

Simulation of Cross-flow Drier

The calculation procedure for static bed drier can easily be modified to predict the moisture content and temperature change which occur when the material is moving through the drier as well as the air. Computer simulation of a cross-flow drier based on the thin layer theory can be done considering a cross-flow drier as analogous to a static bed drier, the bed profile at a given distance along the trough or down, being the same as that in a corresponding static bed drier after the time interval necessary for the bed to move that distance. Calculations are made after suitable chosen time interval of equivalent throughput, mean moisture content and grain temperature across the bed at the conclusion of the last set of heat and mass transferred calculations and mean exhaust air temperature, humidity and enthalpy at exit from the N^{th} layer over each of the I interations from the commencement of drying. For given air flow rate and air and grain inlet condition, N*I set of heat and mass calculations (where I is the total number of time interations) will cover all required throughput rates. calculations of means can also be made to represent driers of various bed thicknesses less than that represented by N layers, so that the same N*I basic calculations can in effect give information for several different driers, for instance, to determine optimum bed thickness in a design problem.

The validity of the simulation has been checked by comparing the predicted values with those measured in static bed and cross-flow driers.

Results and Discussion

Fig. 4 shows the tempering tank temperature profiles at given drying air temperatures during the months of August and October. The tempering tank of a continuous flow drier serves as a cooling area, grains from the drying section passed to the cooler under a restriction which mixed the grain to some exent. Deterioration in storage of many products, including cereal, is a function of excessive temperature as well as moisture content. The tempering tank temperature is, therefore, an important factor to consider in making efforts to improve the performance of the drier and quality of the dried grain. **Fig. 3** shows the plot of tempering tank temperature at different drying air temperatures as a function of drying time.

The temperature in the tempering tank ranged from 28 to 38°C and the drying time ranged from 10 to 22 h during the summer month of August. During the autumn month of October the tempering tank temperature ranged from 12° to 37°C and the drying time ranged from 5 to 22 h. High moisture grain kept at temperature above 25°C start to deteriorate af-

Test No.	August	D31	D32	D33	D34	D35	D36	D37	D38
	October	D41	D42	D43	D44	D45	D46	D47	D48
Air temp. ℃		47.5	47.5	n.h	n.h	47.5	55.0	55.0	55.0
Air rate m3/s		0.03	0.05	0.03	0.05	0.03	0.015	0.03	0.05
Defumidifier		on	on	on	off	off	off	off	off

* n.h:non heat

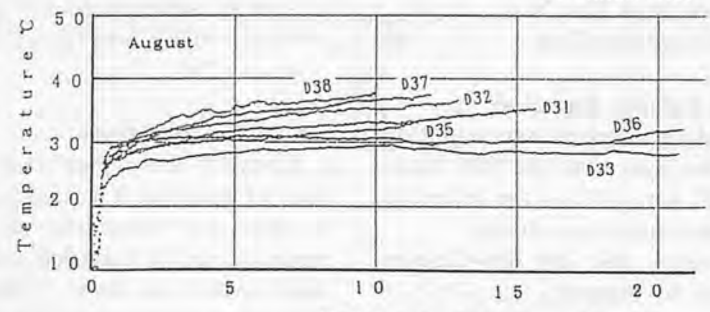

Fig. 4 Temperature profile in the tempering tank.

ter a short period. Loss in quality and viability of rice grain dried with heated air ranging between 40° to 60°C and tempered at 30°C for a total time period exceeding 13 h has been reported (Abe T., et al, 1990). It is, therfore, important for designers of grain driers to seek a way of reducing the temperature in the tempering tank to about 25°C which has so far given the best result in terms of quality and viability of the dried grains.

Validation of the Simulations

Before a model can be used it must be validated by comparison with experimental results. Generally, a model should give reasonable predictions of grain moisture content and temperature profiles for a range of drying air. The validity of the simulation has been checked by comparing predicted values with those measured in static bed and cross-flow driers.

Fig. 5 shows the agreement between the simulated air temperature, experimental air temperature, simulated moisture content values and the experimental moisture content values.

Table 1 gives the root mean square erros calculated for moisture content (% w.b.) for each of the drying tests and predicted values using values of rough rice thin layer drying constant developed by the author, Kameoka and Hosokawa et al. Ideally one would like to obtain a complete agreement with the experimental results but such a result would be very difficult to obtain in this type of experiment.

Table 1 Experimental Moisture Content and Simulated Values

Test condition	Ta = 48.87°C, Xa = 0.0112kg/kg′, Air flow rate = 0.031m³/sec						
Time (min)	Actual M.c %w.b	Predicted using Author's K value	Predicted using Kameoka's K value	Predicted using Hosokawa & Motohashi's K value	Difference from M		
					Author's	Kameoka's	Hosokawa & Motohashi's
0	22.6	22.60	22.60	22.60	0.00	0.00	0.00
60	22.1	22.00	21.94	22.13	0.10	0.16	−0.03
120	20.8	20.97	20.88	21.30	−0.17	−0.08	−0.50
180	19.8	19.87	19.80	20.41	−0.09	−0.02	−0.63
240	18.7	18.77	18.75	19.51	−0.11	−0.09	−0.85
300	17.8	17.70	17.73	18.63	0.10	0.07	−0.83
360	16.8	16.67	16.76	17.74	0.13	0.04	−0.94
480	15.9	15.62	15.83	16.88	0.28	0.07	−0.98
520	14.7	14.63	14.96	16.03	0.07	−0.26	−1.33
Root mean square errors					0.0450	0.0253	0.2195

Note: Room mean square errors were computed by (1) summing the squares of the deviations between experimental and simulated values; (2) dividing by the number of observation; and (3) taking the square root.

Air temperature—The values of simulated air temperatures and experimental as shown in **Fig. 5** indicate that all three equations generally produced a small difference between experimental and predicted values of temperature. This is a good indicator that the simulation model is capable of predicting the performance of cross-flow drier, over a wide range of conditions, with reasonable accuracy.

Air humidity—Effort was made to compare measured air absolute humidity at the inlet to the drier and at the outlet from the drier, with simulated values of air absolute humidity. In genral, the measured values during drying experiments with a cross-flow drier show a great difference from the simulated values. One source of error is the mixing of air coming out from the drying chamber with air from the tempering tank before getting to the exit where the hydroscopic property is measured.

Fig. 6 shows the plot of measured absolute humidity of the air and simulated values for a static deep bed drier when the drying air

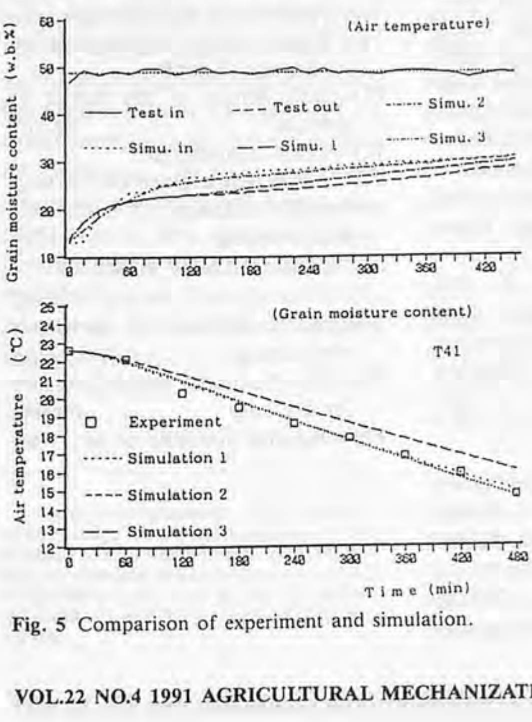

Fig. 5 Comparison of experiment and simulation.

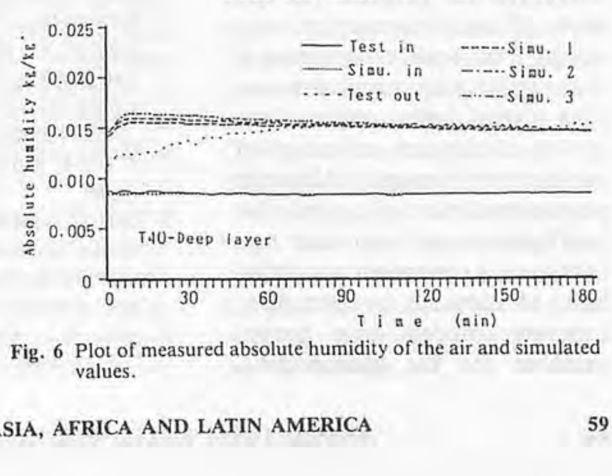

Fig. 6 Plot of measured absolute humidity of the air and simulated values.

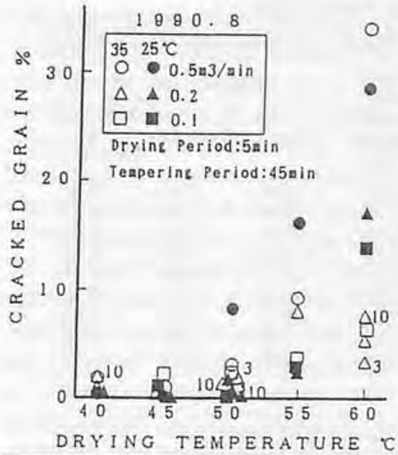

Fig. 7 Effect of drying parameters on grain crack ratio.

temperature was 40°C. It can be observed that the measured and simulated values are in close agreement except during the initial period of the experiment.

Fig. 7 shows the effect of drying parameter such as air flow rate, drying air temperature, tempring temperature and exposure period on crack ratio, when the paddy was dried from 26.4% moisture content wet basis to 15% moisture content. Drying air temperature, tempering temperature and air flow rate are the most important factors that effect the cracking of paddy during drying. Tempering temperature of 25°C is considered better than 35°C.

Conclusions

Computer simulation of grain driers proved valuable for the study of their management and design. Full scale experiments is costly and it is not practical to consider several drying seasons for a variety of climates and range of management strategies. Although precise prediction of a particular configuration of grain drier performance is dependent on availability of adequate the layer data, computer models have proved valuable for the management, performance test evaluation, and design of high temperature grain driers.

It is possible to study many aspects of conventional drying problems that would be difficult if not impossible to obtain in the laboratory or in field application by the use of computer models.

It is important to keep the temperature in the tempering tank at around 25°C as this would enhance the head rice recovery and the quality of the dried grain.

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Nomenclature	*Dimensions*
AR = Area of layer	m^2
dt = The element of time adopted for calculation	sec
dz = Thickness of elemental layer of material	m
M = Moisture content dry basis	
M^i = Moisture content dry basis after drying	
dM = Change in moisture content dry basis, in an element of time	
G = Mass flow rate of drying air per unit area of bed	$kg/m^2/sec$
Cp_a = Specific heat of dry air	J/kg
K = Drying rate parameter	sec^{-1}
F = An operator defined in the text	
X_a = Mass of water associated with unit mass of air, before drying	
X_e^i = Mass of water associated with unit mass of air, after drying	
dX_a = Change in air humidity in an element of time	
HCAS = Valumetric heat transfer coefficient	$J/m^3/sec/°K$
Cpw = Specific heat of water liquid	J/kg/°K
Cpw_v = Specific heat of water vapour	J/kg/°K
D = An operator defined in the text	
Ta = Temperature of drying air	°C
Ta^i = Temperature of drying air after an element of time	
PD = Bulk density of dry matter in material	kg/m^3
TG = Grain temperature	
TG^i = Grain temperature after drying	
h = Specific enthalpy of moist air before drying	J/kg
h_a = Specific enthalpy of dry air	J/kg
h^i = Specific enthalpy of moist air after drying	J/kg
PWSTA = Saturated vapour pressure for dry bulb	mmHg
PH = Relative humidity of air	

Drying Fruits and Vegetables with Solar Energy in Egypt

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Abstract

A solar tunnel dryer was tested in Egypt for drying grapes, okra, tomatoes, potatoes, onions, basil and wild marjoram. The capacity of the dryer ranged between 100 and 200 kg of fresh material per day. The consumption of electric energy for operating the dryer in most cases was about 0.1 kWh per kg of dried material. An organoleptic evaluation gave results between good and excellent for all dried samples with the exception of tomatoes. The total microbial count of the different crops, except onions and tomatoes, was diminished by factor 10 and more by solar drying. The reconstitution properties of the dried samples were sufficient.

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Introduction and Objectives

In Egypt natural sun drying is one of the most common ways to conserve agricultural products. Cereals, legumes, fruits, vegetables and spices are spread on the ground to be dried by sun and wind. During drying these products are neither protected against dust and rain nor against rodents, birds and insects. Poor quality due to contamination with partly pathogenic microorganisms and high losses caused by uneven or incomplete dehydration are the characteristics of natural sun drying. To meet the food requirement of the growing population and, moreover, to provide high quality products for export it is necessary to develop suitable methods of drying. An economical use of oil or gas-heated dryers as common in industrial countries is restricted to cooperatives or large plantations because of the high initial costs and the need for expensive fossil energy. Therefore, solar drying facilities are reasonable alternative for smallholders, especially under the favourable meteorological conditions of Egypt.

Under the terms of agreement between the government of the Federal Republic of Germany and the government of the Arab Republic of Egypt on cooperation in the field of scientific research and technological development a joint project between the Horticulture Research Institute, Agriculture Research Center and the Institute for Agricultural Engineering of Hohenheim University was started on scientific and technological cooperation in the field of solar crop drying. Within this framework a low-cost solar dryer with integrated collector being developed at the Institute of Agricultural Engineering of Hohenhim University was mounted at the experimental site of the Directorate of Agriculture in Medinet El-Fayoum (1, 2).

The purpose of the investigations was to demonstrate the benefits of solar drying under Egyptian climatic, social and economical conditions. Specific objecives include:

—Determination of drying time and energy consumption for different products under varying weather conditions;
—Evaluation of product quality in terms of colour, taste, reconstitution properties, microbial count and life expectancy; and
—Economic evaluation of the solar drying system.

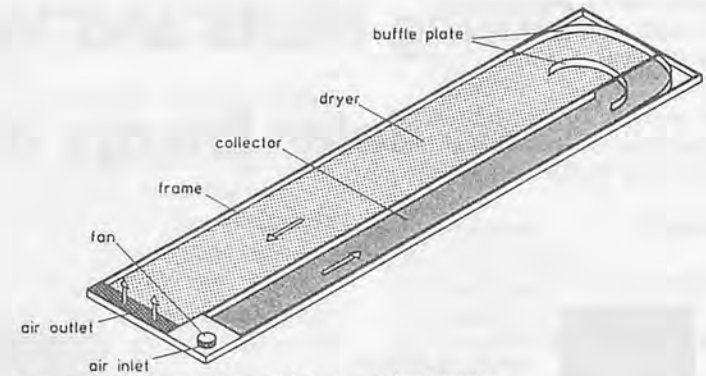

Fig. 1 Solar tunnel dryer with integrated collector.

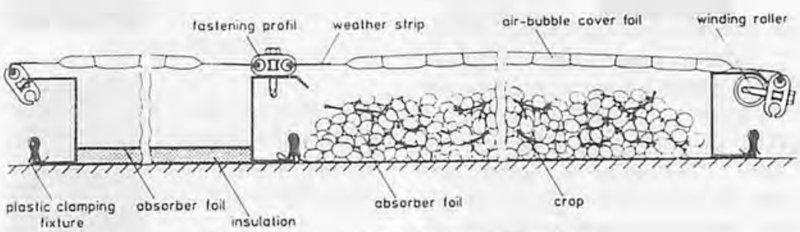

Fig. 2 Cross section of collector and dryer.

Equipment

The above mentioned solar dryer with integrated collector consists of a small radial flow fan driven by a 150-Watt AC-motor and a solar collector leading into a tunnel dryer being arranged in parallel. To reduce the air resistance and, therefore, the power consumption of the fan to a minimum, the air was directed over a thin layer of crop spread on the bottom of the tunnel dryer instead of being forced through a bulk.

The frame of the collector and tunnel dryer, 20 m long and 6 cm high is fixed on the ground (**Fig. 1**). The collector is 1 m wide, the tunnel dryer 2 m. Both components are covered by a transparent PE-EVA air-bubble foil. Due to high UV-stabilization and impact strength, the life time of the foil is about five years. The air-bubble foil was provided with a weather strip fastening profile, which is welded along both sides of the length. The foil was fastened on the frame by pulling the weather strip into a PVC-profile which was fixed on top of the side walls. On the bottom of the collector a black polyester fabric is layed out as absorber. To reduce back side heat losses insulation material was installed underneath the absorber (**Fig. 2**).

Ambient air is sucked by the fan and forced through the collector, heated by conversion of solar radiation, reversed by 180 degrees and directed over the crop spread in the tunnel dryer. According to the absorption properties of the crop, solar radiation is also converted in the tunnel dryer. Yet, the heat loss caused by evaporation of water during drying is compensated by this additional energy gain, resulting in almost uniform temperature along the tunnel. To open the dryer for filling or control the cover foil can easily be wound on a tube by one person.

Procedure

Scientists at the Horticulture Research Institute carried out drying experiments for grapes, okra, tomatoes, potatoes, onions, basil and wild marjoram. To remove the peel of onions as well as the skin of potatoes an electric peeling machine was used. The tomatoes, potatoes and onions were sliced with the use of an electric slicing machine. Grapes were immersed in an emulsion of potassium carbonate, olive oil, palmetic acid and water before spreading on the drying tunnel in order to increase the water permeability of the waxen coat (3, 4). The okra, wild marjoram and basil were dryed without pre-treatment.

Results and Discussion

The load of the dryer was fitted to the properties of the different crops. It ranged between 150 kg for sliced crops like tomatoes, potatoes and onions and 1000 kg for grapes, being dried as whole (**Table 1**). With the exception of tomatoes, having a high initial moisture content of 93% w.b., the moisture content of the other examined crops was around 80%. The final moisture content varied between 6 and 9% w.b. The load of the dryer as well as the pretreatment and the physical properties of the crop take influence on the drying time, which ranged from 29 hours for wild marjoram and basil to 130 hours for grapes.

The capacity of the dryer, calculated as quotient out of the load of fresh material and the drying

Table 1 Load, Drying Time, Capacity and Energy Consumption of Solar Tunnel Dryer for Various Crops

Item* Unit	m_1 kg	U_1 %	m_2 kg	U_2 %	t h	c kg/d	E kWh	e kWh/kg
Grapes	1000	80	220	9	30	185	19.5	0.089
Okra	500	81	101	6	56	214	8.4	0.083
Tomatoes	150	93	11	7	73	49	11.0	1.000
Potatoes	150	77	37	7	32	113	4.8	0.130
Onions	150	84	26	6	34	106	5.1	0.196
Basil	200	80	43	6	29	166	4.4	0.102

*m_1 load of fresh material per batch, m_2 mass of dried material, U_1 initial moisture content w.b., U_2 final moisture content w.b., t drying time, c capacity, E total energy consumption, e specific energy consumption on the base of dried material.

time, was highest for okra with 214 kg/d. Due to the high initial moisture content of tomatoes the capacity of 49 kg/d was remarkably low. In accordance with the drying time the total electric energy consumption of the fan amounted 19.5 kWh for grapes and 4.4 kWh for basil. The specific consumption of electric energy, calculated as kWh per kg of dried material, ranged from 0.083 kWh/kg for okra to 0.196 kWh/kg for onions. For tomatoes as an exception a specific energy consumption of 1.0 kWh/kg was calculated.

In comparison to natural sun drying the drying time for grapes could be reduced from 7.5 days to 5.5 days by solar drying, i.e., the drying time has been reduced to 70%. Drying okra, basil and wild marjoram the drying time could be reduced to 40%. A comparison for tomatoes, potatoes and onions is not possible because natural sun drying of these crops is not common in Egypt (Fig. 3).

To assess the quality of dried products, especially that of vegetables, the extent of reconstitution was used as measure. To determine the reconstitution the dried product was soaked in distilled water at 25°C for 40 minutes (5). The results of the reconstitution test is shown in **Fig. 4**. Onions and potatoes approximately reached initial moisture content whereas okra and tomatoes only reached 75% of this value.

A further parameter of quality is the total microbial count, i.e., the number of microorganisms per gram of sample as given in **Fig. 5** (6). In all cases, except for onions and tomatoes, the microbial count was diminished by factor 10 and more due to solar drying instead of being increased as known from natural sun drying. For tomatoes, the number of microorganisms was not affected by drying. In the case of onions, the count of bacteria was nearly 20 times higher in the dried product than in the fresh sample. This may be due to the distinctive constituents being destructive of the viability of bacteria, which are active only in the fresh product.

The assessment of the product quality was completed by an organoleptic evaluation. Samples of the solar dried crop were rehydrated using distilled water at 80°C for 10 minutes and 20 minutes for tomatoes, respectively, a panel of 10 members judged the colour, aroma, flavour and texture of four samples on a 1-10 hedonic scale. The samples being packaged in

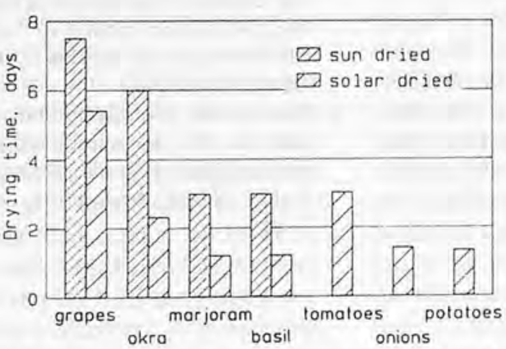

Fig. 3 Drying time of different solar and sun dried products.

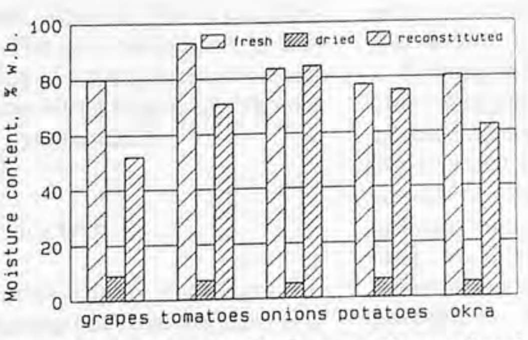

Fig. 4 Moisture content of fresh, dried and reconstituted products.

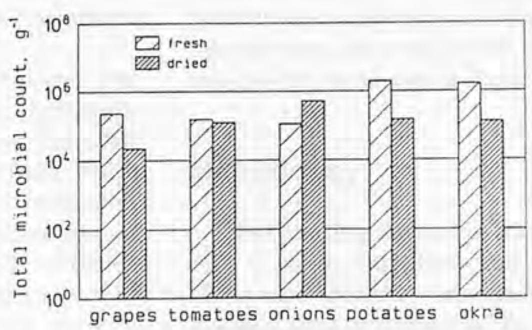

Fig. 5 Total microbial count of fresh and dried products.

polyethylene bags and stored under room conditions were examined monthly for one year. The scores are listed in **Table 2**. The majority of the examined crops kept its grade between "excellent" and "good" for a year. However the tomato slices lost their grade "good" after four months because of spoilage of colour as well as deterioration in aroma, flavour and texture caused by chemical composition reactions.

Table 2 Organoleptic Evaluation of Dried Products

	Time of storage, in months											
	1	2	3	4	5	6	7	8	9	10	11	12*
Raisins	10	10	10	10	9	9	9	9	9	8	8	8
Okra	10	10	10	10	10	10	10	10	10	10	9	9
Tomatoes	8	7	7	7	6	6	5	5	5	2	2	2
Potatoes	10	9	9	9	9	8	8	8	8	8	8	8
Onions	9	9	9	9	9	9	9	9	9	8	8	8
Basil	10	10	10	10	10	10	10	10	10	10	10	10
Marjoram	10	10	10	10	10	10	10	10	10	10	10	9

*1-2 very poor; 3-4 poor; 7-8 good; and 9-10 Excellent

Conclusions

In summary, the solar tunnel dryer with integrated collector showed, compared to natural sun drying, the following advantages:

—The simple construction of the solar dryer enables manufacturing either by small-scale industries or by farmers using locally available materials.

—The drying time was reduced significantly resulting in a higher product quality in terms of colour and reconstitution properties.

—During drying the crop was protected against rain, dust, insects and other animals leading to a reduction of microbial count as well as to a reduction of mass losses.

—A cost-benefit analysis showed that the pay-back period ranges from 1 to 2 years taking into account the reduction of mass losses and the improvement of product quality.

Although the electricity costs for operating the fan are negligible in comparison to the additional earnings, the use of the solar dryer is limited to electrified areas. The low power consumption of the fan offers the possibility to use solar cells as energy source in further investigations.

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A Portable Field Recording Device for Draft Measurement

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Abstract

A very simple, fairly accurate and portable load measuring/recording kit was developed to fulfil the need of researchers on farm implements in developing countries. The unit involves a two-stage strain amplifier and an audio recorder converted to a field recorder with a simple timer-based frequency modulator to record analog load signals to the audio tape and filter based FM demodulator to play back the analog output thus recorded. A programme for 8085 microprocessor was developed to digitalize the recorded data and to compute the average of load signal at 100 m.sec intervals. The unit yielded precise pull reading of a plough tested in comparison with a non-recording pull-type dynamometer.

Introduction

Agricultural engineers in developing countries are more and more concerned about developing implements which are energy economic in their use. This demand for measurement of parameters concerning the forces acting on the agricultual implements. Already available instruments are custom- built for industrial uses and no instrument could be found in developing countries to suit the need of agricultural engineers who want to measure, for instance, the pull of an animal-drawn plough. The available field loggers are either too bulky to be portable or need AC power source for their operation. Keeping in view the basic need of the researcher, an attempt was made to develop a simple device to suit an ordinary tape recorder to record the measured parameter.

Further, it is often found that only non-recording hydraulic dynamometers are used for draft measurements in the field, which cannot be read with reasonable accuracy due to fluctuation of load. To alleviate this difficulty the recorded signal corresponding parameter was fed into a microprocessor to find the accurate average value.

Review of Literature

Clyde[1] measured the forces acting on a tillage tool using six dynamometers which are basically of the hydraulic diaphragm types whose readings were photographically recorded. Neuhoff[2] developed a strain gauge instrument to show how a strain analyser and recorder could be used to record the sum of two or more forces simultaneously by connecting members electrically instead of measuring forces separately. Zoerb[3] developed a low cost strain gauge type dynamometer which measured the draft component of an angled, pull automatically and by measuring the speed on a tail wheel calculated the horse power directly.

Methodology

The recording device has a load cell, a strain amplifier, and an audio tape recorder converted to record signals, as shown in **Fig. 1**.

The load cell is a standard strain gauge-type tension device, the specifications of which are:

Capacity : 100 kg tension
Safe overload : 150% rated load
Transducer supply: 5V DC
Terminal resistance: 200 ohms
Gauge factor : 2
Linearity : ± 1% FSD
Working temperature : 0 to 70°C

The designed strain amplifier has simple circuitry involving 741 Op.amp ICS (Fig. 2). The circuit

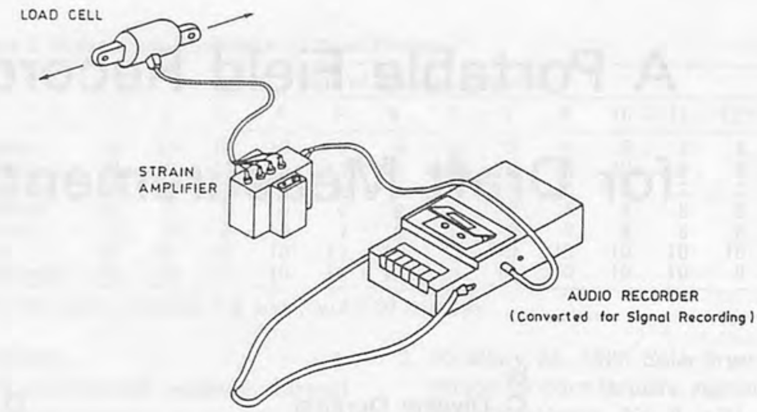


Fig. 1 Recording instrumentation.

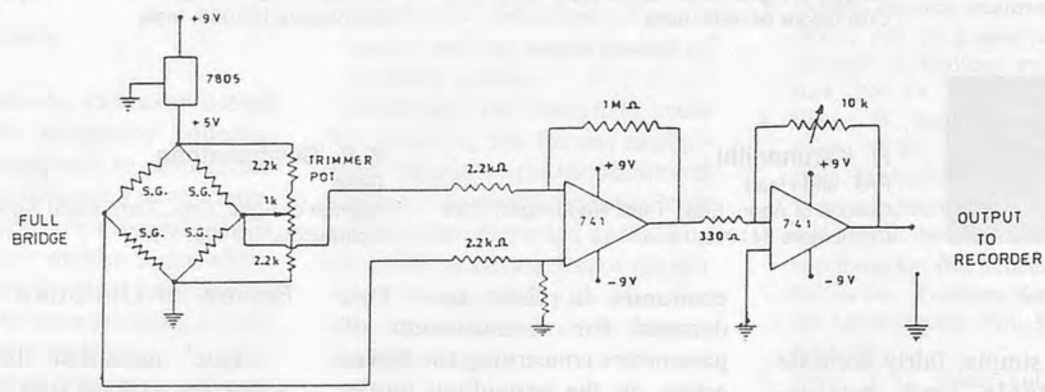
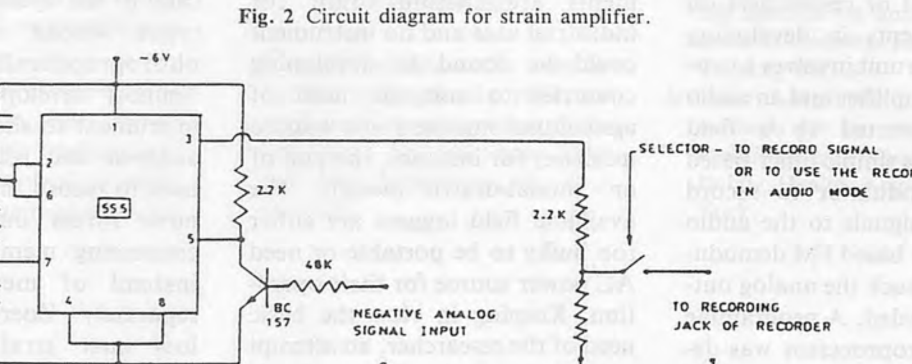


Fig. 2 Circuit diagram for strain amplifier.



CIRCUIT FOR RECORDING

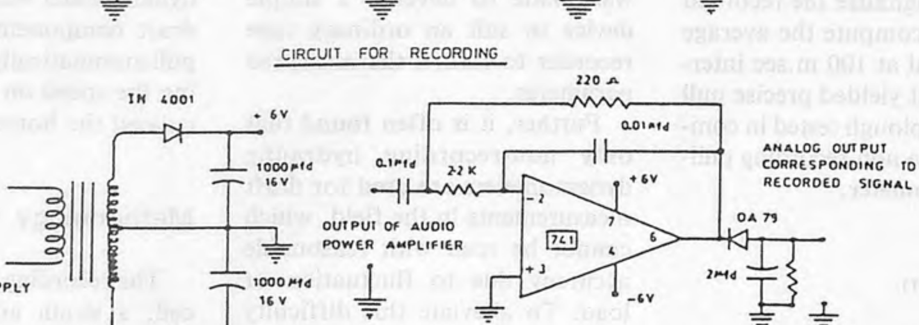


Fig. 3 Circuit for playback.

has two stages of amplifiers that are designed to give the final output of 4V corresponding to 100 kg maximum load on the load cell. Offset null networks have also been incorporated to minimize errors. Gain adjustment to calibrate the instrument is provided to accommodate the use of the recording device.

The strain gauge load cell could be connected to the four terminals or the bridge points of the circuit and shunt balancing network given to balance the instrument for zero output at no load. The PCB bearing the circuit is accommodated in a small acrylic package with compartments for two 9V transistor batteries. The batteries provide a ± 9V dual supply to operate the 741 operational amplifier. An output jack is provided on the unit to tap the output of the recorder. Four bridge terminals are also provided to connect the load cell in.

The audio tape recorder conversion circuit has a timer built around IC 555 acting as a high frequency signal generator of Audio range (**Fig. 3**). The analog signal to be recorded which is of very much lower frequency to audio level is applied to the base of a BC 157 transistor network connected to the 'control' input of the '555'. The output from the signal generator is controlled /frequency modulated by the analog signal fed in. This output by suitable impedance matching resistance network is given to the recording jack terminals through a single pole switch. The FM signal of audio range is recorded on the cassette tape. The switch can be opened if necessary to convert back the recorder to an ordinary audio recorder.

During playback the recorded output is taken from the audio power amplifier of the recorder itself with a high gain. This signal is taken to a 741 operational amplifier built as a band pass filter. The filter is so designed to have a shart response curve and is utilized as a non-linear frequency discriminator. So according to the frequency of the reproduced signal from the cassette, the gain of hte filter varies non-linearly. Corresponding signal is obtained from the output which is demodulated by a single detector diode and smoothened out by capacitors. The final output though not linearly related to the signal that is recorded, the output can be easily analysed through a calibration chart (**Fig. 5**). The power supply for the circuit is derived from the recorder itself.

The strain amplifier and the audio tape recorder together proved to be a very light portable unit weighing only 10 N, which can be carried around in the field, especially in a ploughed area.

The tension load cell is hitched to the bullock-drawn plough similar to an ordinary pull type dynamometer. A shielded cable from the load cell provides connection to the strain indicator /recorder carried by the operator at the back.

The recorder on play back yields an analog output corresponding to the load exerted on the cell. In the laboratory the output was connected to a SDA 85 microprocessor kit through an A/D convertor interface. The kit is a 8085 microprocessor based

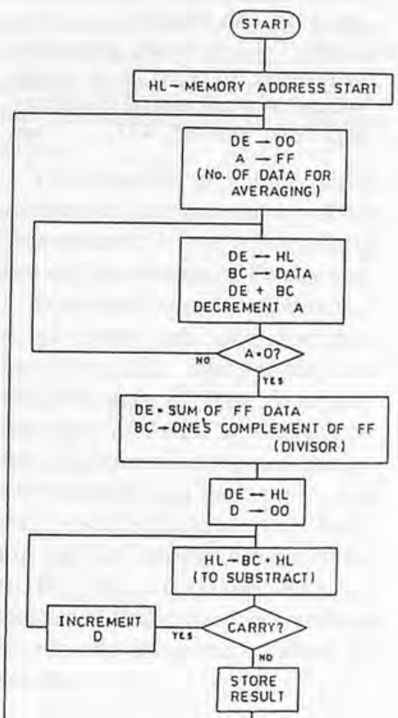


Fig. 4 Flow chart for averaging software.

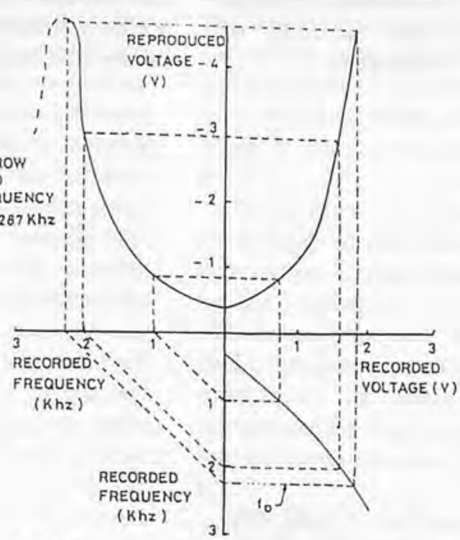


Fig. 5 Calibration chart of the recording-reproducing circuit.

unit provided with 8255 PPI ports. The analog output is hence converted by the interface programmed to store the digital equivalents sampled at an interval of 100 m.sec which is well below the settling time of the comparator and DAC 08 provided in the interface.

The digitalized data is thus recorded in allocated memory spaces of the microprocessor's RAM. A programme was developed to determine the average of the data thus stored to a precise level.

The flow diagram of the programme is presented in **Fig. 4**. The programme accommodates for computing the average of FF numbers of data only and store the result in a specified memory location. This process is done sequentially for data sets of FF numbers until the data stored is depleted to the end.

Three:

POP H	:	Revive IIL, the memory address processed
DCX H	:	Last - Decrement address
MOV M, D	:	Store quotient in last address of FF set of data
INX H	:	Increment Address
MOV A, H	:	Test for end of dataspace
CPI ICH	:	
JNZ : REP	:	Yes, Repeat, No, Reset

RST.5

The calibration chart shown in **Fig. 5**, is the superimposition of the recorded voltage vs frequency relationship on the reproduced voltage vs frequency relationship of the device, as depicted.

The equipment was used in an experiment to measure the draft of an animal-drawn improved iron plough. The recorded signals when reproduced and loaded into the microprocessor kit was in the range vs 8F H to B3 H corresponding to 549 N to 686 N of pull. The computed average was A7 H equipment to 640 N pull.

An ordinary hydraulic dynamometer was also used in the above trials. The hydraulic dynamometer's indicator was fluttering between 60 and 70% F.S.D of 981 N, and was not easily readable.

As the recorder and load cell strain amplifier were both portable, the signal was fed through a very short shielded cable. This yielded a noise-free recording and reproduction.

The heart of the reproducing mechanism is a band pass filter and the entry of 100 W and flutter frequencies at a lower level are hence inhibited.

Conclusion

A highly portable device weighing only 10N, was developed for recording fluctuating signals from transducer used in application such as draft measurement of an implement in the field.

A microprocessor software was developed utilizing an 8085 kit to sample and average the recorded data over specified intervals of time using suitable interface.

The output from the field recorder could also be directly coupled to a chart recorder or analog meter for direct recording or measurement.

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Effect of Method of Handling Tractor Fuel: Iraqui Countryside Experience

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Abstract

It is necessary that diesel fuel entering a diesel fuel injection equipment should be clean, free of dirt, and remaining within its standard specifications. Most of the changes on these specifications take place after receiving fuel from the refinery, i.e., during its handling. The contaminators include sediment of sand or rust as well as water. The study includes also the change in fuel flash point which has relationship to fuel storage safety.

This research depended on the increase of contaminators above the standard (Iraqui refinery fuel) and on the change in flash point.

Samples of diesel fuel were collected from eight administrative districts (AD) representing the middle region of Iraq. The samples were analysed in the laboratory to determine the sediments, water content and flash point. The results were analyzed statistically and showed no significant effect on the basic fuel specifications because of the method of handling despite the incorrect method of storage.

Introduction

Fuel is transported from oil refineries to filling stations all over the country in tankers. Farmers are supplied with it from these stations.

The Iraqi farmer store the fuel in barrels of different shapes and sizes although the majority of them are standard barrels of 45 British gallons (204 litres) usually used for oils. After filling these barrels at the filling stations they are transported by wagons drawn by tractors; if it is a singular barrel it may be put on the bar joining the two lower link arms of the hydraulic system in the tractor. The tractor or wagon drawn by the tractor move on sandy roads to the farms. The barrels are dropped on the ground in the open where they are exposed to direct sunlight and to the various weather conditions. The floor is usually contaminated with dripping fuel when pouring the fuel from the barrels into containers used to fill the tractor tank. These containers are usually topless cans supplied with wooden handle nailed to the top side of the can. Cleaned paint cans of one gallon are also used **(Fig. 1)**. Fuel is poured from these containers into the tractor tank with the help of metal or plastic funnels. These containers are usually left on the ground, fuel barrels or on cement slabs, all of which are exposed to the various contaminating conditions.

By comparing these conditions to which the fuel is exposed during storage and handling with the recommendations issued by the formal organizations or scientific authorities, it seems that the tractor fuel is exposed to too many contaminators and at high rates.

Barger E.L. et al state for the flash point that "It is, however, of importance in connection with legal requirements and safety precautions involved in fuel handling and storage and is normally specified to meet insurance and fire regulations".[1]

Flash point is defined as "the minimum temperature to which the fuel is heated to form a mixture of its vapor with air which is inflammable for a moment only, or a mixture which gives a flash when a fire is brought close to it".[2]

The general recommendations concerning keeping fuel from contamination during storage issued by the Ministry of Oil state that "the most important reasons for fuel contamination and the deterioration of its qualities during storage are the mixture with water and existence of sand and dust in it".[3]

The Shell Petroleam Co. Ltd. states the importance of correct fuel handling and storage and their

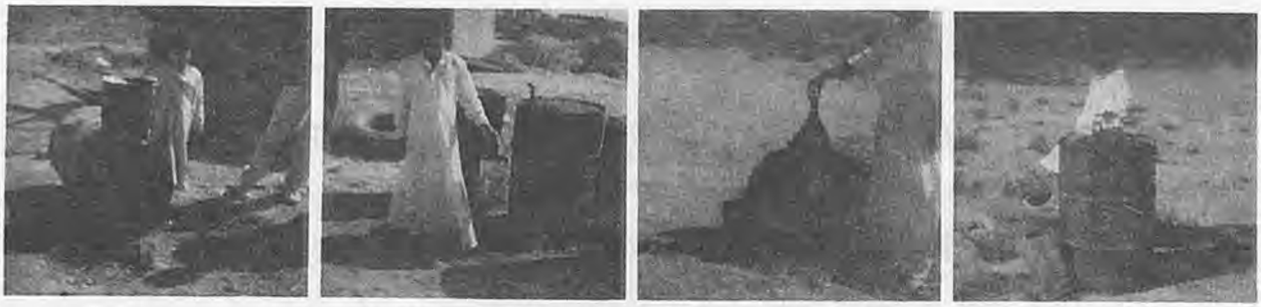

Fig. 1 Storage and handling equipment on different farms.

effect on the diesel engine performance as follows:

"Large, slow-running engines of the kind mainly used for the propulsion of motor ship can digest a wide range of fuel, but small, high-speed engines such as are generally used in diesel tractors are more selective. In these engines, running at speed of the order 1500 r.p.m., the time available for complete combustion is very short".

"The fuel injection equipment of a diesel engine is manufactured to exceeding fine limits and it is vital to keep the fuel clean. Dirty fuel can, in fact, do far more harm in a diesel engine than in a spark ignition engine. To safeguard against contamination, fuel specifications usually include limits for sediment, ash and water content. When transporting the fuel great care is taken by the oil company to avoid contamination; unfortunately, all too often, contamination occurs after delivery, through careless handling by the operator".

"Quite small particles are likely to score injection equipment and obstruct injection orifices. Water is, of course, exeptionable, especially in that wet fuel is likely to cause blocking of fuel filters".

"Tanks and drums containing fuel or lubricating oil should be shaded from direct sunlight because wide temperature variations increase the chance of water condensation within the container".

The Shell Co. emphasized that fuel contamination usually comes after receiving it from the filling stations, that is, during handling.

Materials and Methods

Random diesel fuel samples were collected in 300 cc glass bottles during the months of June, July and August 1989 from the eight Middle Region Governorates, i.e., Diala, Babylon, Kerbala, Anbar, Wasit, Najaf, Muthanna and Qadissiya. Collecting the samples was done in the following way:

During tours through farm roads in each AD we asked farmers about any farmer owning a tractor in the area; upon direction to the farmer a sample is taken from his tractor tank and another sample from the farm filling tank, then the farmer is asked about the filling station from which he got his fuel supply. A sample is taken also from the filling station.

From each AD 8 samples were taken from 8 tractors and 8 samples from filling tanks and 3-4 samples from filling station as some farmers get their supply from the same filling station. A total of 152 samples were collected from the eight AD. Upon completion, another tour was taken to collect samples from the same tractors, filling tanks and filling stations so that there were two different samples at least collected at two different times for statistical analysis. Thus the number of samples reached 304. The Oil Research Centre asked for I.D. 42 000 ($134 400) as fees for analysing these samples which forced us to reduce the number of samples by mixing at random every two or three samples representing tractors, filling tanks and filling stations. This resulted in 3 repeated samples representing tractor tanks, the same for filling tanks and 2 repeated samples representing filling station in each governorate. Thus the total number was reduced to 128 samples which wre analysed in the laboratory.

Results and Discussion

The laboratory results of analysis were arranged in tables containing flash point (C°), sediments (%) and water content (%). These results were analysed statistically to determine the mean of flash point (°C) with the standard deviation (±). The mean of sediments (%) and the mean of water content (%) for each sample representing tractor tank (T.T.), filling tank (F.T.) and filling station (F.S.) in the eight AD covered by the survey. The mean for flash point, sediments and water content was found for all governorates as shown in **Table 1**.

Upon comparison of the above results with the physical and chemical properties (standard disel fuel specification) listed in **Table 2** (above) for two kinds of diesel fuel (one of them is used in high-speed engines such as used in diesel tractors and the other slow-running engines as used in marine engines) we conclude the following:

Conclusions

1. Since the flash point is considered as an indicator of fuel safety during storage (whenever the flash point is higher the fuel is

safer during storage period) the fuel storage safety in all ADs is within safe indicators which should be 650°C as a minimum despite our belief that it is very bad and very dangerous.

If we want to compare flash points in farm storage in the ADs we find that the best one is in Babylon (71.17 ± 1.040°C) while the worst one is in Najaf (70.83 ± 3.185°C).

2. The percentage of sediments allowed internationally should not exceed 0.10%. Upon scrutinizing in **Table 1**, we find that the percentage of sediments in all ADs is much less than the allowed percentage. The statistical analysis indicates that there are no sediments in all filling stations in all ADs. These sediments vary in percentage in tractor tank and farm filling tanks from one AD to another despite it's being less than the allowed limit. The best ADs in having the least sediments in farm filling tanks were Anbar and Qadissia (zero) and the highest was (0.014%). The highest sediment content in tractor tanks was in Qadissia; this means that filling the containers used in filling tractor tanks from farm filling tanks are more contaminated in this AD.

3. The primary belief resulting from the way the field filling tanks are laid on the ground without shade indicating an increase in the percentage of water content in these tanks because there is a great chance of water condensation resulting from humid air entering these tanks. Both laboratory analysis and statistical analysis showed no sign of water in the fuel. The reason for that, in our belief, is that the samples were collected in the hot months of summer and under direct sunlight accompanied by an increase in the barrel temperature resulting in evaporation of humidity, if indeed condensation developed during the night on the inside wall of the barrel.

Table 1 Statistical Analysis of Diesel Fuel Samples Collected from Tractor Fuel Tank, Farm Filling Tank and Filling Station

AD	Type of sample	Flash point (°C)	Sediment (%)	Water (%)
Diala	T.T.	69.92 ± 0.577(*)	0.007	0
	F.T.	71.42 ± 2.810	0.011	0
	F.S.	71.63 ± 0.530	0	0
Babylon	T.T.	70.75 ± 1.500	0.002	0
	F.T	71.17 ± 1.040	0.0015	0
	F.S.	73.00 ± 0	0	0
Kerbala	T.T.	70.75 ± 1.090	0.008	0
	F.T.	70.75 ± 1.090	0.002	0
	F.S.	70.00 ± 0	0	0
Anbar	T.T.	70.92 ± 1.627	0	0
	F.T.	70.92 ± 1.702	0	0
	F.S.	71.63 ± 0.530	0	0
Wasit	T.T.	70.50 ± 1.323	0.014	0
	F.T.	70.00 ± 1.090	0.014	0
	F.S.	71.50 ± 0.707	0	0
Najaf	T.T.	71.00 ± 2.839	0.014	0
	F.T.	70.83 ± 3.185	0.0095	0
	F.S.	71.63 ± 0.530	0	0
Muthanna	T.T.	70.67 ± 1.258	0	0
	F.T.	70.42 ± 1.127	0	0
	F.S.	70.25 ± 0.354	0	0
Qadissia	T.T.	70.83 ± 0.946	0.018	0
	F.T.	70.33 ± 0.520	0.006	0
	F.S.	70.50 ± 0.707	0	0
Mean of all administrative mit (AD) or Governorates	T.T.	70.67 ± 0.338	0.008	0
	F.T.	70.80 ± 0.565	0.0055	0
	F.S.	71.27 ± 0.997	0	0

(*) First number refers to the mean and the second one, preceded by (±), refers to the standard deviation.

Table 2 Physical and Chemical Properties of Diesel Fuel(*)

Property	Values for high-speed engines		Values for slow-running engines	
Flash point (Pensky-Martens)	Minimum	65°C	Minimum	65°C
Sediment (%)	Maximum	0.10	Maximum	0.10
Water content (%)	Maximum	0.15	Maximum	0.20

(*) Reference No. 2

4. Despite the positive results of the way fuel is stored on the farms, it is not free of criticism shown in the previous figure. The barrels were put in clay sites the base of which is fuel with sand. The equipment used to fill the tractor tanks (cans, gallons, funnels, etc.) are put on the ground directly in the open where it is exposed to contamination. We suggest that the simplest fuel storage methods are to be followed, i.e., barrels are put on concrete slab rather than the ground, that they are covered with a shade, and that the tractor tank filling equipment are kept in closets or places far from dust and contaminants. In this respect we recommend reference Nos. 4 and 5.

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Feasibility of Fuel Ethanol from Cane Molasses in Pakistan

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Abstract

Fuel ethanol from its characteristics is an ideal substitute for gasoline. Its 20%-blend with gasoline can be used in conventional engines without any modification with additional benefits of reduced pollution. The fuel ethanol production technology is simple, proven and locally available. In Pakistan, under present circumstances, molasses is the most suitable raw material for fuel ethanol production. The surplus molasses in the country is sufficient to produce 100 to 105 million litres of fuel ethanol annually.

Introduction

Pakistan is one of the major sugarcane producing countries in the world. The average sugarcane production has been 31.2 million tons per annum during the last 10 years. The sugar mills crush around 50 percent of the production. About 650 thousand tons of molasse is produced annually as a by-product. More than 70% of it is exported at a very low price.

Presently the country is heavily dependent on oil imports to meet energy requirements. More than 50% of foreign exchange earnings are spent on the import of fossil fuels. Molasses can be converted into fuel ethanol (99-100% ethyl alcohol), mixed 20 percent with gasoline and used in spark ignition engines without modification. This can help reduce the dependency on imported fuels.

Methodology

There are 45 sugar mills and eight industrial ethanol producing distilleries in Pakistan. A survery of 10 sugar mills and four distilleries was conducted. The mills and distilleries were randomly selected. Data regarding the production and utilization of molasses/ethanol and cost of various raw/finished materials were collected. The structural questionnaire/informal interview system was adopted. The standard methods of Net Present Value (NPV), Pay-Back Period and Internal Rate of Return (IRR) were applied for economic analysis of fuel ethanol production.

Results and Discussion

Characteristics of Fuel Ethanol

Ethanol (C_2H_5OH) is an organic chemical. It is completely soluble in gasoline, diesel or fuel oil, provided no water is present in the system. It mixes and burns well with gasoline in internal combustion engines (1). The fuel ethanol as such cannot be used as diesel substitute because of difference in their combustion properties.

Molasses As Raw Material For Fuel Ethanol Production

Due to simple conversion process, molasses has been the most common raw material for ethanol production all over the world. The ethanol production is higher in the case of molasses due to high sugar content. It contains between 50-55 percent fermentable sugar and yields about 280 litres of ethanol per ton of molasses (1). **Table 1** presents an estimate of the composition of molasses (2). In Pakistan, under present circumstances molasses is the most suitable raw material for fuel ethanol production due to easy availability, simple conversion process, high alcohol content and low production cost.

Molasses Production

In Pakistan every ton of cane sugar produced gives around 1/2 ton of molasses which is usually 4-5 percent of the total cane crushed (3). **Table 2** shows that

Table 1 Composition of Molasses

Constituent	Normal % Range
Water	17-25
Sucrose	25-40
Invert sugar	8-20
Other carbohydrate	2-5
Ash (carbonate)	7-15
Nitrogen compounds	3.3-6.5

Source: Alternate Uses of Sugarcane and its products (1979).

Table 2 Sugar and Molasses Production (Unit: Million ton)

Item	1983-84	1984-85	1985-86	1986-87	1987-88
Cane crushed	13.5	14.7	12.1	14.5	20.3
Cane sugar	1.14	1.31	11.02	12.56	17.44
Cane molasses	0.619	0.661	0.546	0.666	1.020
Molasses % cane	4.58	4.5	4.526	4.60	5.02

Source: Pakistan Society of Sugar Technologists (1988).

the cane sugar production which was 1.13 million tons during 1983 has gone up to 1.75 million tons in 1988. Accordingly, molasses production has also increased from 0.76 million tons to one million tons. In the next three years eight more sugar mills are expected to be operational. This will take the installed capacity to 2.2 million tons annually (4). As such the annual molasses production will be more than one million tons in the coming years.

Present Molasses Utilization

At present a very small quantity of molasses (less than 30% of the total production) is consumed in the country. Almost 90% of it is converted into industrial alcohol (95% ethanol) by eight distilleries operating in the country **(Table 3)**. About 53% of the industrial alcohol is exported. A very small quantity of molasses is also used for the production of animal feed and bakers yeast, etc. However, the surplus molasses is sufficient to produce 100 to 105 million litres of fuel ethanol.

Production Technology for Fuel Ethanol

Table 3 Molasses Consumption and Alcohol Production in Pakistan Distilleries.

Distillery	Molasses consumption (tons/year)	Alcohol production (tons/year)
Premire, Mardan	21 000	3 700
Frontier, Takhtbhai	55 000	1 000
Khazana, Peshawar	7 000	2 100
Habib, Nawab Shah	40 000	8 600
Hye Sons, Khanpur	12 000	2 200
Hussain, Jaranwala	14 000	2 500
Crecent, Faisalabad	14 000	2 500
Ravi Rayon, Lahore	30 000	6 400
Total	143 500	29 300

The technology to produce ethanol from molasses through fermentation is simple, proven and locally available. Presently eight distilleries are producing industrial ethanol (95%) from this process **(Table 3)**. The production process of ethanol from molasses is shown in **Fig. 1**. The process consists of dilution of molasses, acidification and addition of a selected strain of yeast. The fermentation proceeds for 2 to 3 days at 20° to 40°C to produce an alcohol concentration of 8-10 percent. This mixture is distilled to give a 50-60% alcohol as tillage. The alcohol stream is redistilled to give a 95% product. A final codistillation with benzene or another azeotrophic agent can yield absolute (100%) ethanol (5). During the ethanol production process CO_2 and fodder yeast are produced as by-products.

Use of Fuel Ethanol As Engine Fuel

Fuel ethanol from its characteristics, particularly by its high octane (above 89) and low cetane number (below 10) is, as a unit fuel, an ideal substitute for gasoline (5). In principle all engines with internal combustion can use ethanol as fuel. The more adaptations are made in the engine the more effective is its utilization. Fuel ethanol can be used as follows:

1. As "gasohol" in which case anhydrous ethanol is mixed with gasoline up to 20% ratio in conventional engines with carburetor and in higher ratios after modifications.
2. For direct burning for heating.

The fuel ethanol mixture (gasohol) can be used in gasoline engines without any engine problem. The research and practical ex-

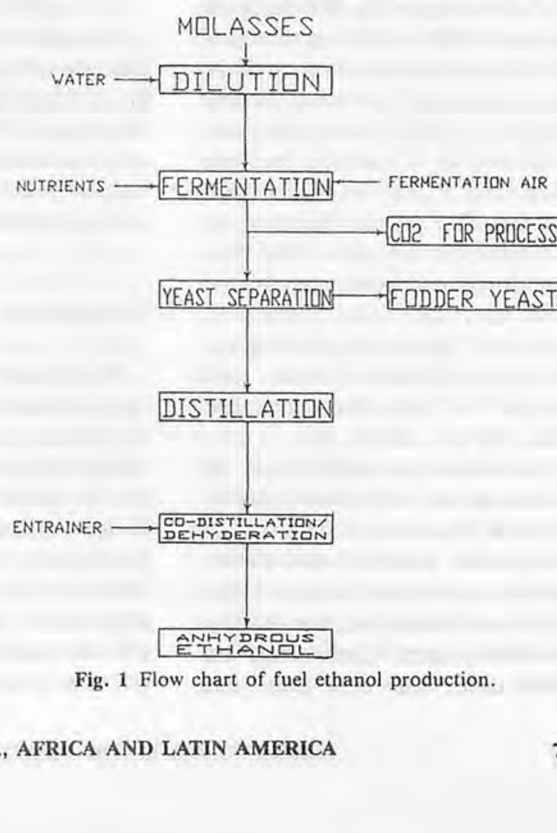

Fig. 1 Flow chart of fuel ethanol production.

Annexure 1 Economic Analysis of Fuel Ethanol Production

A. General		
1. Capacity of the plant	65 000 tons molasses per annum	
2. Expected fuel ethanol production	60 000 litres per day or 18 mil. litres/annum	
3. Expected life of the plant	15 years	
4. No. of operating days	300 per year	
5. Construction period	24 months	
6. Interest/average discount rate	12%	
B. Capital Cost		(000 US$)
1. Machinery, equipment, transportation		4 640
2. Land, civil construction and erection of mechanical equipment		855
3. Supervision of erection + management cost		945
4. Pre-Operational cost including commissioning, start-up + training.		620
Total:		7 060
5. Interest during construction period		1 200
Total investment until start-up		8 260

C. Aanunl Cost of Fuel Ethanol Production	(000 US$)
1. Cost of molasses (65 000 tons œ US$ 25 per ton)	1 625
2. Salaries + wages + administrative cost	270
3. Chemicals (725 tons)	110
4. Electricity (6 480 MW @ US$ 0.05 per kWh)	234
5. Fuel oil (360 tons @ US$ 100 per ton)	36
6. Maintenance and Spare parts	185
7. Transportation charges of chemicals and fuel oil (1 085 tons @ US$ 7.5 per ton)	8
8. Depreciation of total investment for 15 years	550
9. Interest on fixed investment (12%)	495
10. General over head charges (1% of operating cost of point 3-7)	6
Total:	3 519
D. Analysis	(Amount in 000 US$)
- Total annual income from fuel ethanol production (assuming present petrol price of US$ 0.36 per litre)	6 435
- Production cost per year	3 519
- Excise duty at US$ 0.06 per litre	1 125
- Net gain per year	1 791

perience in many countries such as Brazil show that existing gasoline engines do not require any modification to run on gasohol (20% anhydrous ethanol blends). For using ethanol in higher proportions modifications of the engine have to be made in carburation and compression ratio. Anhydrous alcohol is the most suitable for utilization as pure fuel because it needs high air-fuel ratio and lower compression ratio.

Economics of Fuel Ethanol Production

According to the World Bank report (1980) ethanol production in a medium cost country could be economical at the existing petroleum prices (1980) in case the economic value of molasses was less than US$ 60 per ton at the plant (1). Gasoline price in Pakistan has increased by 45% since 1980. The molasses export price is around 40 US$/ per ton. The break-even point for the manufacturing capacity is 20,000 litres/day and number of operating days less than 180 per annum (1).

An economic analysis of an optimum size fuel ethanol plant is given in Annexure I. The cost of equipment, materials and chemicals are estimated. In case of electricity consumption, the existing electricity tarrif (industrial) has been used. The sale price and excise duty have been fixed according to the gasoline price and duty prevailing in the country. For simplicity of calculations it has been assumed that the plant starts 100% production from first year and the profit remains constant throughout its life. The results of various methods applied for economic analysis are as follows:

A. Net Present Value (NPV)
= 3069.1 thousand US$ at 12% discount rate.

B. Pay-Back Period
= 7.5 years.

C. Internal Rate of Return (IRR)
= 12%

The above data reveals that the business of fuel ethanol is profitable in Pakistan under present circumstances. The proper disposal of by-products such as CO_2 and fodder yeast can make it even more profitable.

Conclusion

Fuel ethanol has been proven to be a commercial automobile fuel. The existing conventional engines do not require any modification to run on gasohol (upto 20% anhydrous ethanol blends). The production technology of fuel ethanol from molasses is simple, proven and well known. The surplus molasses not consumed in the country is sufficient to produce 100 to 105 million litres of fuel ethanol annually. In Pakistan under present conditions the business of fuel ethanol production can be profitable.

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Optimum Utilization of Animal Dungs as Domestic Fuel: A Case Study

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Abstract

The study was conducted in Hisar district to assess the effective utilization of animal dung for household cooking. The study reveals that the families possessing 3-4 animals can very well obtain the benefit of installing a biogas plant of 2 m^3 capacity. Besides, getting biogas as fuel for cooking purpose, better environment and good quality of manure having higher percentage of nutrients are obtained as compared to those families using cow dung as domestic fuel in the form of dung cakes. About 3.14 times the heat of dung cakes is available when the same dung is used in biogas plant for gas production. But, those families possessing 1-2 head of animals cannot benefit from a biogas plants. This is because to install a biogas plant it needs a minimum of 3-4 animal to meet the requirement of dung for the plant. Families possessing more niamals can install a large size biogas plant which not only meet their cooking requirements but also provide extra fuel for their engines or lighting purpose.

Introduction

The coventional biomass fuels like firewood, crop residues and animal dung are used in traditional mud stoves. The animal dung is used in the form of dung cakes. The calorific values of wood, crop residues and dung cakes are 20MJ/kg, 16-18 MF/kg and 8.76 MJ/kg, respectively. According to an NCAER survey the firewood, dung cakes and crop residues account for 56%, 20% and 19%, respectively, as cooking fuels in the rural areas. The areas under forests and wood is not adequate to meet the timber and cooking fuel requirements of the country. Firewood is scarce and costly. Similarly, crop residues generally used are cotton, rape seed and Arhar stalks. According to Girja Saran (1984) only the farmer having more than 45-ha land holding can produce surplus crop residues to fulfil the cooking requirements of his family. But the farmers in this category are very limited in number. Therefore, to narrow the gap between the demand and supply of energy for cooking purpose, the only fuel left is animal dung.

There is no second opinion that the animal dung has multidimensional utilities. It is utilized as fuel, building material and farm yard compost since ages. Presently in village it is mostly used as domestic fuel and provides about 74% of the total energy consumed for cooking and only very limited quantity of dung is utilized for farm yard manure and that, too, in rainy seasons. The use of dung cakes as fuel not only pollute the environment but lot of nutrients are also lost. On the other hand, if same dung is used for making farm yard manure, a lot of nutrients in the form N,P and Kare available to the crops. The use of dung in biogas plants can further enhance the energy availability in addition to good quality of FYM. Considering the present energy crisis, and in order to bridge the gap beween the demand and supply, proper and alternative use of cattle dung is very vital.

Therefore, keeping this objective in view, a study was conducted in Hisar district to assess the availability/potential of animal dung, mode of its use by the farmers and to suggest suitable/viable alternate ways of its utilization to increase thermal efficiencies of the existing cooking techniques.

Materials and Methods

The study on extent of dung utilization in various villages of Hisar district was undertaken and 50 families from all the four villages situated around Hisar, namely; Anipura, Badon, Neolikalan and Deven were selected in different categories of the farming families. The respondents were randomly selected using random sample technique. A standard questionnaire was prepared and a sample survey in each village was carried out in order to determine the number of animals possessed by the family, dung produced and its mode of use, i.e., preparation of dung cakes for domestic fuel, biogas or farm yard manure. Statistical data on ownership of biogas plant was also gathered from the selected families. The data was analysed by classifying the respondents according to number of animals owned and size of the biogas plants available for them. The yearly total dung produced in each family was computed and its heat value was calculated, when the dung is used as dung cakes or as biogas.

The theoretical considerations for the calculation/analysis of data are as follows:

1. Average quantity of dung produced /animal (kg/year) = 4 105
2. Average dry dung produced/animal (kg/year) = 660
3. Gas production/kg of fresh dung at 20-25°C (m^3) = 0.0414-0.066
4. Heat value of dung cakes (MJ/kg) = 8.918
5. Heat value of biogas (MJ/m^3) = 18.841
6. Heat utilization efficiency of traditional chulhas (%) = 9-10
7. Heat utilization efficiency of biogas (%) = 45-50
8. Gas required for cooking/day/adult (m^3) = 0.35-0.45
9. Annual compost production/animal (kg) = 860
10. Composition of compost (kg/1 000 kg) when fresh dung used for N = 5.0, P_2O_5 = 2.0, K_2O = 5.0
11. FYM production/animal when dung is passed through biogas plant (kg) = 1 032.0
12. Composition of FYM (kg/1 000 kg) when fresh dung is passed through biogas plant N = 16.0, P_2O_5 = 15.0, K_2O = 10.0

The additional benefit of utilization of dung as biogas and production of FYM by the biogas plant was also analysed. On the basis of this study the recommendations for different sizes of biogas plants according to number of animals possessed by the farmers were given.

Results and Discussion

The respondents of the survey were classified according to landholding, number of animals owned, family type, educational status, family size, occupation and age group (**Table 1**). Majority of the respondents were from farming group (94%) of which 43% had landholding size of 4 hectares each, followed by the group of 2-4 ha of holding (26%) each. Landless families represented 6% of the total.

Thirty-four percent of the respondents had 3 to 4 head of

Table 1 Distribution of Respondents in Sample Survey

Particulars	No. of Respondents	Percentage of Total
Family Size		
Small (1-4 members)	21	42.0
Medium (5-8 members)	21	42.0
Large (More than 8)	8	16.0
Age group (years)		
20-30	28	56.0
30-40	11	22.0
Above 40	11	22.0
Educational status		
Illiterate	36	72.0
Primary	6	12.0
Middle	3	6.0
Above middle	5	10.0
Land holding (acres)		
Landless	3	6.0
Less than 2.5	4	8.0
2.5-5	9	18.0
5-10	13	26.0
Above 10	21	42.0
Animals owned (No. of head)		
Less than 3	10	20.0
3-4	17	34.0
5-7	10	20.0
8-10	9	18.0
More than 10	4	8.0
No. of biogas plants owned	0	0.0
Family occupation		
Farming	42	84.0
Non-farming	8	16.0

animals each whereas, 20% had having 5-7, 18% 7-10 and 8% had more than 10 head of animals (**Table 1**). The households were categorized into different family sizes, i.e., small (0-4), medium (5-8) and large (above 8). The respondents were mostly from medium (42%) and small (42%) family size. The sample contained about 48% and 52% as nuclear and joint families, respectively. Most of the respondents were illiterate and only a very small percentage (16%) had middle or lower educational qualifications.

Table 2 shows the comparative results of different modes of animal dung utilization. It is clear from the table that when the dung is used in biogas plants the calorific value of biogas produced is 18.841 MJ/m^3 with thermal utilization efficiency as high as 45-50%. Whereas it is only 8.918 MJ/kg with 9-10% thermal efficiency when used as dung cakes. Moreover, no FYM is available and all nutrients are lost due to burning of cakes. An additional advantage of using dung in biogas plant is clean environment and availability of good quality manure. When dung is used for compost formation then only 860 kg of compost/animal/year is available. The compost contains only 4.2 kg, 1.68 kg and 4.10 kg/1 000 kg of compost N,P,K, respectively. Contrary to this when dung is passed through biogas plant 1 040 kg/animal/year, FYM is obtained which is in agreement with Joshi et al (1985) according to whom the amount of manure obtained through biogas plant is about 46% more than the traditional methods. Also, the quantities of N,P,K available in the FYM of biogas plant are higher than in compost prepared directly. The FYM contains 16.00 kg, 15.60 kg and 10.00 kg/1 000 kg, N,P,K, respectively.

Table 2 Comparative Results of Fresh Dung in Various Models of Utilization

Particulars	Mode of dung utilization		
	Dung cakes	Compost	Biogas
Calorific value	8.918 MJ/kg	—	18.841 MJ/m^3
Heat utilization efficiency (%)	9.0-10.0	—	45.0-50.0
Quantity of FYN/compost produced (kg/adult animal/year)	—	860.0	1 040.0
Nutrient composition of compost/FYM (kg/1 000 kg of FYM)	—		
N	—	4.20	16.12
P_2O_5	—	1.68	15.61
K_2O	—	4.10	10.07

Table 3 shows the dung availability, heat equivalent of dung cakes and biogas and FYM production according varying numbers of animals possessed by the families. It is clear from the table that the average quantity of fresh dung produced by 3-4 head of animals is 11 143.24 kg/year but the dry dung production is only 1 825.66 kg/annum. The heat value of this dry dung when used in the form of dung cakes is very low (8.841 MJ/kg). Moreover, the thermal efficiency of traditional chulha is only 9-10%. The dung cakes not only generate smoke during burning but considerable amount of nutrients are also wasted. According to Singh et al (1977) the loss due to burning of dung cakes amounts to 0.8 million tonnes of nitrogen, 0.91 million tonnes of phosphorus and 0.961 million tonnes of potash in terms of soil nutrients. The smoke generated during burning also pollutes the home environment. However, when the same dung is used in biogas plant, 602.85 m^3 of biogas is produced per annum. The heat equivalent of biogas is 4 500 kcal/m^3 of gas. The total heat equivalent of biogas is 11.359 × 10^{3}MJ. The heat utilization efficiency of biogas varies from 45-50%. Thus heat actually avail-

Table 3 Dung Availability, Heat Equivalent, FYM Production from Animals Owned by Farmers

Item	Number of head of animals				
	1-2	3-4	5-7	8-10	Above 10
Quality of fresh dung produced (kg/year)	4 197.5	11 143.24	19 856.0	30 193.60	44 256.25
Quantity of dry dung cakes produced (kg/year)	687.70	1 825.66	3 253.1	4 946.79	7 250.75
Heat equivalent of dung cakes (MJ/year)	6.134×10^3	16.285×10^3	29.016×10^3	44.13×10^3	64.65×10^3
Heat available from dung cakes for production work (MJ/year)	6.134×10^2	16.285×10^3	29.016×10^2	44.13×10^2	64.65×10^2
Total biogas product (M^3/year)	227.085	602.85	1 074.21	1 633.47	2 394.26
Total heat equivalent of biogas (MJ/year)	4.279×10^3	11.359×10^3	20.24×10^3	30.779×10^3	45.09×10^3
Heat available from biogas for productive work (MJ/year)	1.926×10^3	5.112×10^3	9.107×10^3	13.859×10^3	20.294×10^3
Additional heat available when dung is used in biogas plants	3.139 Times the heat of dung cakes	3.140 Times the heat of dung cakes	3.139 Times the heat of dung cakes	3.140 Times the heat of dung cakes	3.139 Times the heat of dung cakes
Quantity of compost available (kg/year)	840.0	2 230.5	3 971.2	6 038.72	8 851.25
FYM available through biogas plant (kg/year)	1 007.40	2 676.60	4 765.44	7 246.46	10 621.5
Nutrients available from compost (kg)					
N	4.20	11.15	19.86	30.19	42.26
P_2O_5	1.68	4.46	7.94	12.08	17.70
K_2O	4.10	11.15	19.86	30.19	42.26
Nutrients available from FYM of biogas plant (kg)					
N	16.12	42.83	76.25	115.94	169.94
P_2O_5	15.61	41.88	73.86	112.32	164.63
K_2O	10.07	26.07	47.65	72.46	106.21

able for productive work from 3-4 head of animals is 5.112×10^3 MJ which is 3.140 times than the heat of dung cakes for cooking purpose. Similarly, the heat availability from 5-7, 8-10 and above 10 head of animals are 9.107×10^3 MJ, 3.589×10^3 MJ, 420.294 MJ/year, respectively, as compared to dung cakes (29.016×10^2 MJ) 44.13×10^2 MJ & 64.65×10^2 MJ/year, respectively.

Annual compost production from 3-4 head animals is 2230.5 kg but when same dung is passed through the biogas plant, the quantity of FYM produced is 2 676.6 kg. Joshi et al (1985) reported that the quantity of manure obtained through biogas plant is about 46% more than the traditional methods 2 676.60 kg/annum the composition of FYM in terms of nitrogen, phosphorus and potash is 16, 15.5 and 10.0 kg/1 000 kg of FYM, respectively, as compared to compost (5.0, 2.0, and 5.0 kg/1 000 kg, respectively). It has been observed that compost/FYM when properly prepared can give increased yields ranging from 10-60% in addition to restoring soil fertility (ICAR, 1971).

Figure 1 shows the different sizes of biogas plants according to number of head of animals, i.e., 2, 4, 6, 8 m^3 capacities for 3-4, 5-7, 8-10 and above 10 animals, respectively. The average gas production from the plants were 1.65, 2.94, 4.475 and 6.56 m^3/day, respectively. However, for proper gas production from biogas plants the dung slurry having concentration of 6-9% total solids in desired to be fed to the inlet tank of the digester (Joshi et al, 1985). A farmer having 3-4 dairy cows can install a biogas plant of 2 m^3 capacity which will be sufficient for cooking for 5 persons daily and 2 676.6 kg of good quality of FYM is available to the owner in addition to clean environment. Similarly, farmers with 5-6 head of animals can install a biogas plant of 4 m^3 capacity which is sufficient for cooking for 9 persons. Farmers owning 10 or more head of animals can go in for a biogas plant of 8 m^3 capacity. This can meet the requirements of 20 persons daily and 10 621.5 kg of good quality FYM/year is available from this plant. It is also clear from **Table 3** that families possessing 1-2 head of animals cannot meet the requirements of a minimum size biogas plant. Therefore, they should use the animal dung for making compost rather than using it as domestic fuel in the form of dung cakes.

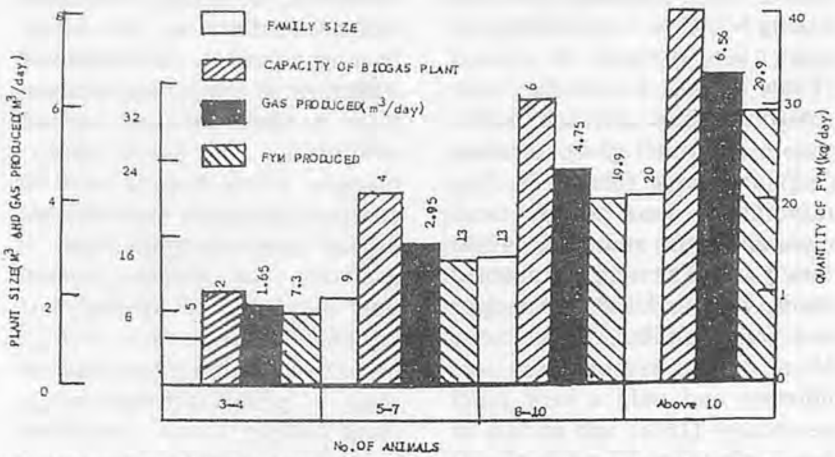

Fig. 1 Selection of bio-gas plant on the basis of number of animals and family size.

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Frictional and Packing Behavior of Green Coffee Beans

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Abstract

The variation in the packing and frictional characteristics of green coffee beans was determined as a function of overburden pressure, moisture content and sliding velocity. These parameters were allowed to vary over a range of 0 to 103 kPa, 10 to 30% wet basis (w.b.) moisture content and 0.8 to 3.0 m/h, respçectively. Throughout these tests the initial bulk density decreased with an increase in moisture content, while the bulk density increased with an increase in vertical pressure. The coefficient of friction increased with increases in both moisture content and sliding velocity while it decreased with an increase in overburden presssure. The internal angle of friction and angle of repose varied with changes in moisture content.

Introduction

In Kenya, during the last 20 years, the area under coffee cultivation has doubled, with a total production of approximately 129 000 t of coffee harvested during 1983/1984 (Couldrey, 1985). Increased production of coffee along with the storage of coffee now becoming much more common in bulk rather than in bags had led to the development of processing and storage facilities (Ghosh, 1966; Sharma et al, 1983; Clarke et al, 1985; and Hkanya, 1987). With the adoption of this type storage technique, knowledge about the physical properties of the stored material under different conditions will be needed to produce reliable engineering designs. The study reported herein was carried out to determine: the effects of vertical pressure and moisture content on bulk density; effect of lateral pressure, moisture content and sliding velocity on the dynamic wall friction coefficient; and the effect of moisture content on the internal angle of friction and angle of repose of green coffee beans.

Background

As the degree of mechanization in the handling of agricultural products increases, it places greater emphasis on the knowledge of physical properties of handled materials and their effects on material flow, bin-wall pressures and the design of handling equipment. There is much information available on the physical properties of many common grains and seeds which have been stored in bulk for many years. However, there is only limited information available on many specialized agricultural products which are grown in the tropical and subtropical regions of the world. This information is required by engineers in the design of storage structures and material handling equipment.

In tests conducted with parchment coffee, Ghosh (1968) reported that the static coefficient of friction was dependent on both the type of sliding surface and moisture content of the product. He found that the coefficient of friction decreased rapidly over a range of moisture contents between 48 and 56% w.b.*. However, further reduction in moisture content had very little effect on the coefficient of friction. A plot of the coefficient of friction as a function of moisture content depicts two distinct regions on the curve. At moisture contents above 56%, the curve was approximately vertical, while at moisture contents between 12 and 48% moisture content, the coefficient of friction curve was almost horizontal. Between 48 and 56% moisture content a transitional stage occurred connecting the two parts of the curve. Ghosh reported static coefficients of friction of 1.75 and 0.30 for parchment coffee at moisture contents

*All moisture contents reported in this paper are given on a wet vasis unless otherwise specified.

of 54 and 12%, respectively.

In tests performed with parchment coffee, Eschewald and Hall (1961) determined that, for fully wet and dry stages, the static coefficient of friction was 1.15 and 0.34, respectively. Tests were conducted using a tilting table apparatus similar to that used by Ghosh.

Experimental Procedure

Green Arabica coffee beans from Colombia, marketed under the trade name of Expresso, were used exclusively in this study. This type coffee corresponds to the most common type species grown in Kenya. The coffee used in this study had undergone several processes such as dehulling and grading and would more likely be stored in bulk at the milling and export level. Prior to testing, lots of 14 kg were prepared having moisture contents of approximately 10, 15, 20, 25 and 30% moisture content. The moisture content of each lot was altered either by a drying or rewetting process. Moisture contents of a given lot were determined using a gravity convection oven and a heating period of 72 h at 103°C (ASAE, 1987)

Bulk Density

Changes in bulk density, as it varied with moisture content and overburden pressure, were measured using an apparatus shown in **Fig. 1**. A similar type apparatus was used by Thompson and Ross (1983) in experiments with soft red winter wheat. The apparatus used a flexible pressure diaphragm and a dial guage. The flexible diaphragm exerted a known pressure on the coffee mass, which was located inside a steel box 30 cm by 30 cm square and 10 cm tall, causing it to compact and thereby caused changes in its bulk density. The pressure applied to the coffee was used to simulate the stress condition, called the overburden pressure, created in a bulk bin at a point by various depths of overbearing material. The change in height of the sample was measured by a dial gauge operated by a brass rod which was inserted through a hole in the middle of the top plate of the apparatus, the bottom end of the rod rested on a small thin metal plate placed on top of the diaphragm. The metal plate was used so that the rod would not puncture the diaphragm, and also to provide a consistent level surface upon which to start the test. The top end of the rod actuated the dial gage as the coffee compressed. The changes in height are related to the

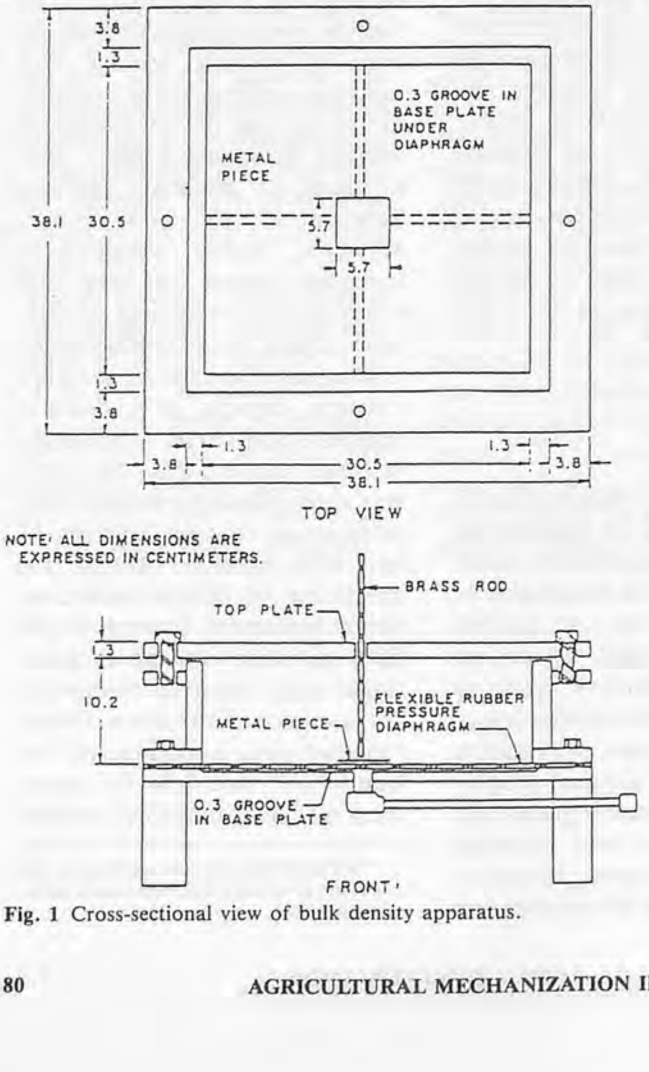


Fig. 1 Cross-sectional view of bulk density apparatus.

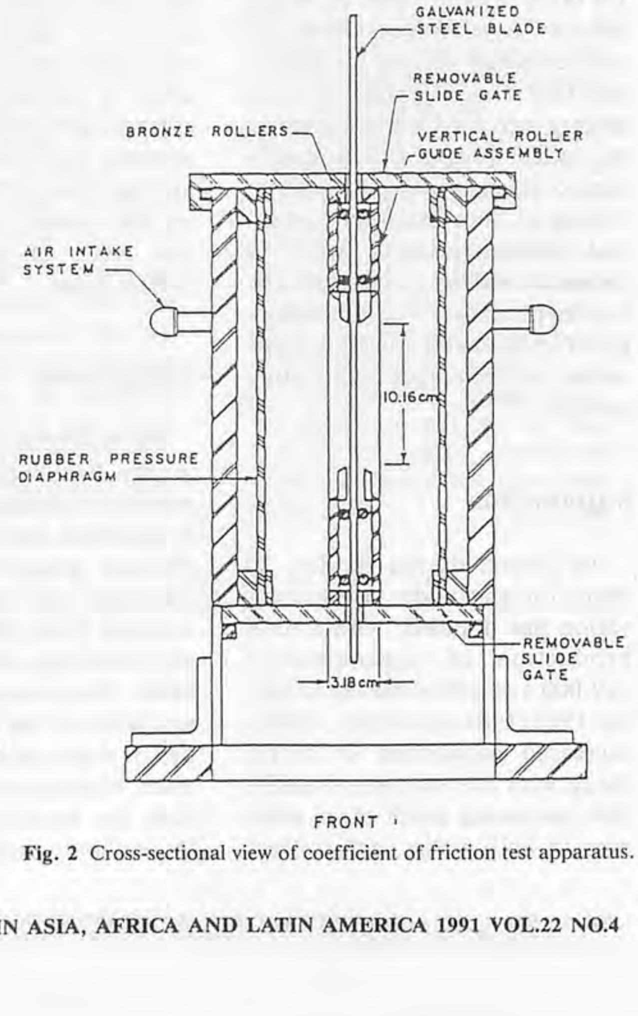


Fig. 2 Cross-sectional view of coefficient of friction test apparatus.

changes in bulk density of the material within the test apparatus. Internal pressure was simulated by applying air pressure under the rubber diaphragm. The air pressure was monitored by a mercury manometer. Tests were conducted over a range of pressure from 0 to 103 kPa which would represent the stress condition found in most farm and commercial storage structures. The bulk density, D_i, resulting from compression is estimated by:

$$D_i = D_0 \frac{h}{h - \delta h} \qquad (1)$$

where,

D_i = bulk density, kg/m^3

D_0 = bulk density at 0 kPa, kg/m^3

h = initial test height of the specimen equal to 0.102 m

δh = change in height of the specimen as a function of overburden pressure, m

A filling box was used to load the coffee into the bulk density apparatus, because of the dependence of the initial value of bulk density on the method of filling. D_0 was determined by dividing the *in-situ* weight immediately after filling by the initial volume of the apparatus.

Dynamic Wall Friction Coefficient

The dynamic coefficient of friction as a function of lateral pressure, moisture content and sliding velocity was measured using an apparatus shown in **Fig. 2.** A similar type apparatus was used by Thompson and Ross (1983) in experiments with soft red winter wheat. The apparatus consists of two flexible pressure diaphragms located in opposite walls of the apparatus. The diaphragms are used to exert a known lateral force on the test sample within the apparatus and simulate the lateral forces which occur on the sidewalls of a bulk storage bin. Lateral pressures ranging from 0 to 103 kPa were simulated by means of compressed air. The metal blade, which was used to simulate the walls of a smooth walled storage bin, was attached to a universal test machine which was used to measure the force required to pull the blade through the coffee sample at a given pressure and speed. Velocities of 0.8, 1.5, and 3.0 m/h were used to simulate discharge conditions as they might occur in commercial storage structures. These velocities correspond to those found in most farm and commercial storage structures during plug flow conditions. The metal blades used in these tests were made of galvanized steel with dimensions of 32 mm wide, 2 mm thick, and approximately 0.45 m long. The forces exerted on the blade by the coffee beans acted over a length of approximately 0.10 m in the middle of the test apparatus. This technique was used to eliminate any edge effects of the forces applied by the apparatus and to accommodate the vertical roller guide assembly. The dynamic sidewall friction coefficient was calculated with the equation:

$$u = \frac{F}{2P\ (LW + kLt)} \qquad (2)$$

where,

F = force required to pull the blade through the grain mass, N

P = pressure exerted on the coffee sample by the diaphragms, kPa

L = length of the test section = 0.10 m

W = width of the test blade = 32 mm

t = thickness of the test blade = 2 mm

k = horizontal to vertical pressure ratio

A spring loaded box was used in filling the apparatus to provide uniform initial conditions. Four replications were carried out for each moisture content level and velocity.

Internal Angle of Friction and Angle of Repose

Triaxial tests were conducted to determine the internal angle of friction using a T-114 single triaxial assembly (Soiltest Inc.). The triaxial device was equipped with a chamber which could accommodate specimens 71 mm in diameter. The magnitude of the internal angle of friction was measured at confining pressures between 55 and 103 kPa. Compressive loads were applied to the specimen at a constant loading rate through the gearing system and the proving ring assembly of the triaxial tester. The criteria of failure was when three or more consecutive readings of the proving ring showed a decreasing or a constant reading (Head, 1981). The internal angle of friction was calculated as:

$$\sin\phi = \frac{(P1/P3) - 1}{(P1/P3) + 1} \qquad (3)$$

where,

ϕ = internal angle of friction

P1 = major principal intergranular stress, kPa

P3 = minor principal intergranular stress, kPa

Four replications were carried out for each moisture content. To ensure a uniform density of a test specimen, the beans were dropped into the specimen membrane from a fixed hight of 0.25 m.

The filling angle of repose was determined using a principle similar to that employed by Fowler and Wayatt (1960). The angle formed by the surface of a conical pile of beans dropped from a funnel of fixed height of 0.25 m was considered to be the filling angle of repose.

Results

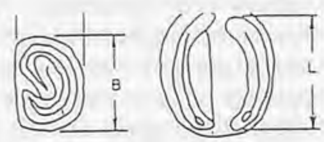
Size Characteristics

Measurements of 250 individual beans were made to determine the average dimensions of coffee beans used during testing (**Table 1**). No statistical correlation was found between various dimensions. It was determined that the average dimensions of these beans were 10 mm long, 3.9 mm in diameter, and 6.9 mm in breadth. It was observed that over 90% of the coffee beans were retained on sieve with openings of 4.75 mm. This limited size distribution is believed to have been caused by earlier grading prior to marketing of these coffee beans. At 10% m.c. the percentage of voids was determined using an air comparison Pycnometer and it was found that the average portion of the volume occupied by the voids was approximately 51%.

Table 1 Average Dimensions of Coffee Beans Used During Testing

Dimension		Mean	Std. dev.	Minimum	Maximum
Length	(L)	10.0	1.0	7.6	12.7
Diameter	(D)	3.9	0.4	2.8	5.8
Breadth	(B)	6.90	6.0	3.9	7.8

All dimensions in the above table are listed in units of mm.



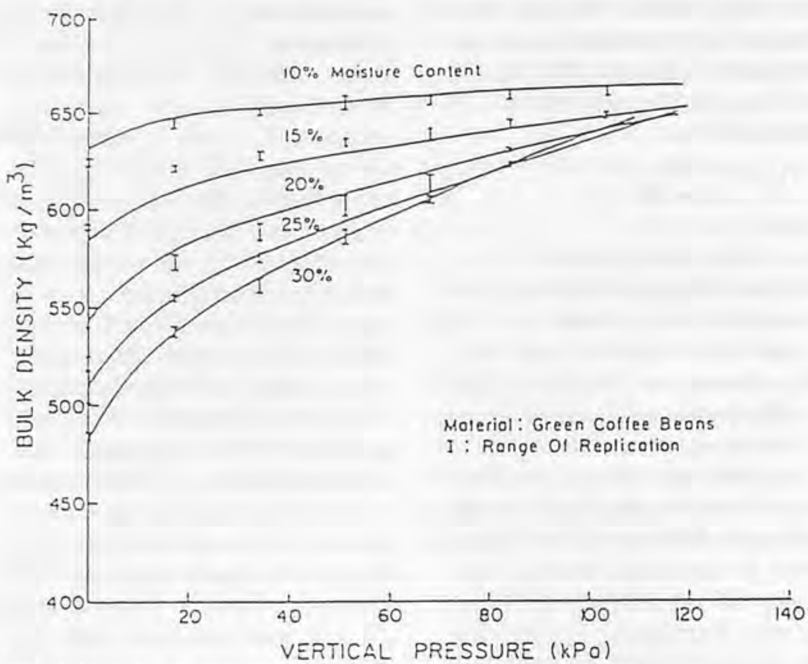


Fig. 3 Variation in bulk density as a function of vertical pressure.

Bulk Density

The variation in bulk density as a function of vertical pressure is shown in **Fig. 3**. It was determined that the bulk density of green coffee beans was significantly influenced by both vertical pressures and moisture content. The bulk density increased as the vertical pressure increased. It is believed that the increase in bulk density was a result of two different phenomena: 1) a decrease in voids caused by rearrangement of the particles; and 2) increased intergranular stresses between the particles. Between 0 and 16.9 kPa it is believed that the predominant change in bulk density is caused by a rearrangement of the test particles within the apparatus which resulted in a decrease in void space between the particles. At 10% m.c., it was observed that 57% of the total change in bulk density occurred between 0 and 16.9 kPa while at 30% m.c. 32% of the total change in bulk density occurred between 0 and 16.9 kPa. At 10% m.c., the coffee beans were hard and inelastic in behavior and it is believed that much of the change that could occur would be caused by rearrangement of the particles within the apparatus while at 30% m.c. the particles are more elastic and pliable and, therefore, it is believed that the changes in bulk density are a result of both rearrangement and increased intergranular stress condition. A discontinuity appears to occur between 15 and 20% m.c. which separates the two states (elastic and inelastic). Over the given range of pressures a 9.6% change in bulk density occurred at 15% m.c., while at 20% m.c., a 19.2% change was observed to occur. Many of the same type observations noted for green coffee beans were similar to those observed for soft red winter wheat as observed by Thompson and Ross (1983).

A model describing the variation in bulk density as a function of vertical pressure and moisture content can be expressed as:

$$\text{Bulk Density} = 743.9 - 12.55\,(\text{M.C.}) - 0.32\,(\text{P}) + 0.13\,(\text{M.C.})^2 + 0.58\,(\text{M.C.} \times \text{P})^{1/3} + 0.04(\text{M.C.} \times \text{P}) \quad (4)$$

where,

M.C. = moisture content of the coffee beans, % w.b.

P = applied pressures, kPa.

The above model had a correlation coefficient of 0.986.

Wall Friction Coefficient

Tests were conducted to determine the effects of initial surface conditions on the dynamic coefficient of friction of coffee beans on galvanized steel. Over the given range or lateral pressures it was noted that as the number of runs increased there was a decrease in the wall friction coefficient. A run can be described as one complete movement of the test blade through the material. For green coffee beans, the average dynamic wall friction coefficient varied from 0.135 to 0.112 for the 1st and 7th run, respectively, at 34.5 kPa. Similar variations in the coefficient of friction were observed by Thomson et al. (1989) for soft red winter wheat in which the dynamic wall friction coefficient varied from 0.180 to 0.152 for similar test conditions. it was determined that long chained alcohols located on the endosperm of the wheat grains were being coated on the galvanized steel.

The effects on the dynamic coefficient of friction of green coffee beans were determined for of m.c. of 10, 15, 20, 25, and 30% m.c. Typical plots of the dynamic friction coefficient as a function of pressure at sliding velocities of 0.8 and 3.0 m/h are shown in **Fig. 4** and **5**. It was observed that over the range of lateral pressures the dynamic wall friction coefficient increased as the moisture content increased. It was noted that at 10% m.c. some beans retained their silvery membranes while at higher moisture contents, the silvery membrane was missing. While little is known about the behavior of this membrane on the coefficient of friction, it was believed that the membrane may cause the dynamic coefficient of friction to decrease. As the beans were rewetted for tests at higher moisture contents the silvery membrane was separated from the beans. Therefore, it did not affect the measured values at m.c. other than those at 10% m.c.

The effect of sliding velocity on the dynamic coefficient of friction of green coffee beans was determined for 0.8, 1.5 and 3.0 m/h. It was noted during the experiments that the dynamic coefficient of friction increased as the velocity increased over the given range

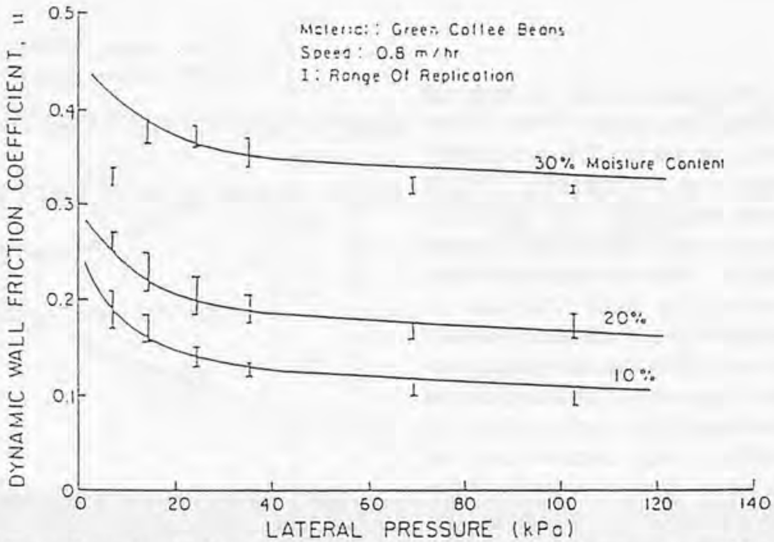


Fig. 4 Dynamic wall friction coefficient as a function of lateral pressure at a sliding velocity of 0.8 m/h.

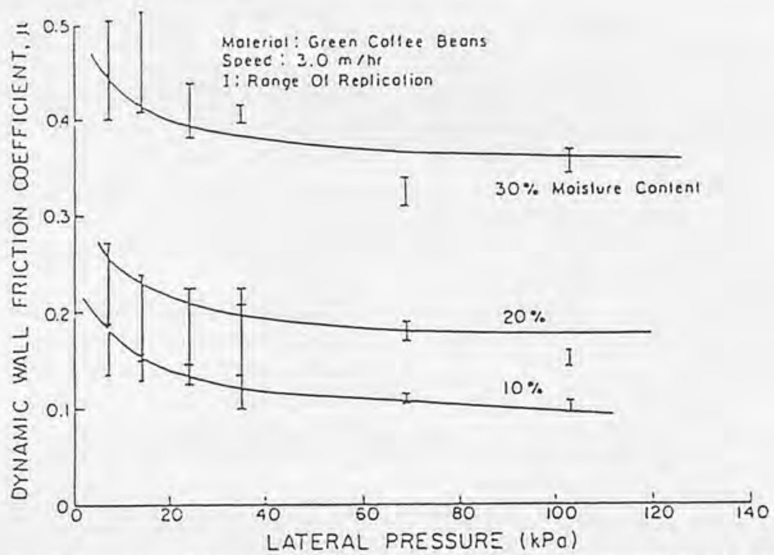


Fig. 5 Dynamic wall friction coefficient as a function of lateral pressure at a sliding velocity of 3.0 m/h.

of pressures. At 20% m.c., it was noted that the average dynamic coefficient of friction increased from 0.259 to 0.322 at 6.9 kPa for sliding velocities of 0.8 and 3.0 m/h, respectively. While at 30% m.c. the dynamic coefficient of friction increased from 0.329 to 0.424 for similar test conditions.

Ghosh (1966) and Eshewald and Hall (1961) used a tilting table apparatus to measure the COF of coffee beans on various materials. Ghosh noted that the static COF varied from 0.3 to 1.75 for 12 and 54% m.c., respectively. While these values are much higher than those achieved in this study, it was believed that the differences between Ghosh and Eshewald and Hall and the results in this study were caused by differences in type of coffee, testing techniques and the static and dynamic COF. The coffee used by Ghosh was collected directly out of a processing plant and did not have its hulls removed prior to testing. Coffee in this state would normally be stored at the processing plant level in bags. The coffee used in this study had undergone several processes such as dehulling and grading, and would more likely be stored in bulk at both the milling and export levels. Likewise, the static COF is normally larger than the dynamic COF for most materials. In addition, Moysey and Fast (1989) suggest that for flax and wheat and some surfaces the static COF can be affected by the adhesion between the materials and the sliding surface if the depth of materials on the tilting table is too shallow. In addition, they caution that the results of these type tests are too subjective with respect to operator technique and interpretation of the results and thus the test method is flawed.

A model describing the variation in the dynamic COF as a function of m.c., lateral pressure, and sliding velocity was developed using stepwise regression and can be expressed as:

$$COF = 0.25 - 0.012V - 0.033P - 3.39 \times 10^{-6}(M.C.)^5 + 0.27(p)^{-1/3} + 0.0059(M.C. \times V^{0.2}) + 7.7 \times 10^{-7}(M.C. \times \ln(M.C.)) + 2.44 \times 10^{-5}(M.C.)^4 \quad (5)$$

where,

P = applied pressure, kPa
M.C. = moisture content of the coffee beans, % w.b.
V = sliding velocity, m/h

The above model had a correlation coefficient of 0.905.

Internal Angle of Friction

The variation in the internal angle of friction of green coffee beans as a function of moisture content was determined for moisture contents of 10, 15, 20 and 25% m.c. Tests were not performed at 30% m.c. because of problems associated with storage of the sample at that high m.c. A plot of the deviator stress as a function of % axial strain for 10 and 20% m.c. coffee is shown in **Fig. 6**. At low moisture contents, a well defined peak deviator stress was observed. The deviator stress is defined as the difference betwen the axial and lateral stress during triaxial testing. However, at higher moisture contents there was not a well defined peak value. During testing, the peak deviator stress was chosen as that stress where the stress-strain curve deviates from a straight line.

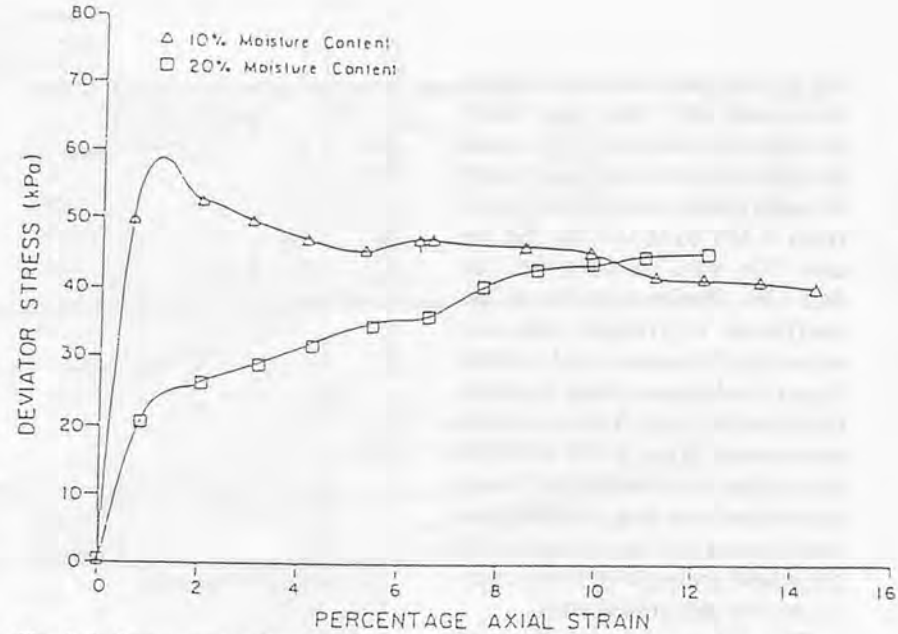


Fig. 6 Deviator stress as a function of percent axial strain at 10 and 20% moisture content.

For green coffee beans, the internal angle of friction was determined to vary between 12.8 and 14.4 degrees and for m.c., between 10 and 25%. **(Table 2)**. This corresponds to a lateral to vertical pressure ratios of 0.64 to 0.60 for these moisture contents calculated from:

$$k = \frac{1 - \sin\phi}{1 + \sin\phi} \quad (6)$$

For these moisture contents, it appeared that the internal angle of friction increased for an increase in moisture content of the sample.

Angle of Repose

The filling angle of repose was significantly affected by moisture content of the samples **(Table 3)**. The filling angle of repose was determined to be between 32 and 40 degrees for all tests **(Table 3)**. These measured results correspond in a similar manner to the angle of repose of other granular products in that Lorenzen (See Pierce and Bodman, 1987) determined that the internal angle of repose of corn varied from 27 to 38 degrees at moisture contents between 7.5 and 23% m.c. and from 29.5 to 41° for wheat at moisture contents of 7.9 to 19.5% m.c.

Summary

The purpose of this paper was to investigate some of the physical properties of green coffee beans. The bulk density, dynamic coefficient of friction, angle of repose, and the internal angle of repose were measured as a function of moisture content, pressure, and sliding velocity.

It was determined that the bulk density was significantly affected by the vertical pressures and moisture content and that the bulk density increased as the vertical pressure increased.

The dynamic coefficient of friction was affected by lateral pressure, moisture content, and sliding velocity of the material on galvanized steel. The dynamic COF decreased for an increase in lateral pressure, increased for an increase in moisture content, and increased for an increase in sliding velocity. It was also determined

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Table 2a Effect of Moisture Content on Internal Angle of Friction

Moisture content (% wb)	Internal angle of friction			
	Mean	Std. dev.	Min.	Max.
10	12.8 A	0.12	12.7	12.9
15	13.7 B C	0.36	13.3	14.1
20	13.6 B	0.19	13.3	13.7
25	14.4 C	0.89	13.3	15.2

*The internal angles of friction in the above table are listed in degrees.
**Means with the same letter are not considered significantly different at the 0.05 level according to Duncan's multiple range test.

Table 3 Effect of Moisture Content on the Angle of Repose

Moisture content (% wb)	Angle of repose			
	Mean	Std. dev.	Min.	Max.
10	37.1 A B	1.09	35	39
15	34.3 C	1.61	32	37
20	36.0 B	1.54	33	38
25	37.3 A	1.30	35	39
30	37.1 A B	1.78	34	40

*The angles of repose in the above table are listed in degrees.
**Means with the same letter are not considered significantly different at the 0.05 level according to Duncan's multiple range test.

Table 2b Effect of Moisture Content on the Pressure Ratio (k)

Moisture content (% wb)	Internal angle of friction			
	Mean	Std. dev.	Min.	Max.
10	0.638*	0.0028	0.635	0.640
15	0.617	0.0079	0.608	0.625
20	0.620	0.0044	0.616	0.625
25	0.601	0.0190	0.584	0.625

$* \; k = \frac{1 - \sin\phi}{1 + \sin\phi}$

that the initial surface conditions on which the material was tested influenced the dynamic COF. As coffee was passed over the sliding surface the COF decreased with the number of runs. In addition, the variation in the COF between samples decreased with the number of runs. This is similar to that determined by Thompson et al (1988) for soft red winter wheat and is believed caused by coating of materials on the sliding surface by the coffee.

The filling angle of repose for green coffee beans varied with moisture content. It was noted that, for the moisture contents tested, the mean filling angle of repose varied from approximately 34 to 37°.

By triaxial testing the internal angle of repose was not significantly affected by moisture content of the sample. The internal angle of friction varied from 12.8 to 14.4° which produced a lateral to vertical pressure ratio in the granular material of between 0.64 and 0.60 for the conditions tested.

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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. The requests from the readers for publishing the whole contents among the articles introduced here may also be sent to editorial staff. Regarding the article of many requests, the publication of whole contents will be reconsidered.

102

Effect of Harvesting Stage on Yield Potential of Soybean: B.B. Saxena, Scientist S-2; T.P. Ojha, Director, Central Institute of Agric. Engg., Nabi Bagh, Berasia Road, Bhopal 462018, India. A.C. Datta, Asst. Prof., Agric. Engg. Dept., IIT, Kharagpur, Midnapore (WB), India.

Harvesting crops at proper time ensures minimum damage and loss of grain during threshing and other losses in subsequent handling and processing operations. Soybean, a rich source of protein is more susceptible to shattering even when the crop is grown under optimum agronomical conditions. this loss becomes excessive if rain drops during harvest. An optimum period of harvest for two local soybean varieties, namely; JS-2 (early maturing and more susceptible to shattering) and JS-7244 (high yielding and less susceptible to shattering) of Madhya Pradesh were selected and trials were conducted in two consecutive years. The data on grain yield, moisture content of grain and shattering losses of soybean showed highly significant variation when harvested during 70-102 for JS-2 variety and 85-118 days after sowing (DAS) for JS-7244 variety. An optimum period of harvesting was statistically determined and was found to be 85 to 93 DAS for JS-2 and JS-7244 varieties during the above periods ranged between 12.9 to 14.8 q/ha and 13.8 to 14.8 q/ha, respectively. Shattering losses were 0.42 q/ha on 93 DAS and increased thereafter and showed 7.8 q/ha on 102 DAS for JS-2 variety. The shattering loss was relatively less for JS-7244 than JS-2 and started 112 DAS onwards.

104

Developments in Food Regume Processing Machinery in India: S.D. Kulkarni, Scientist S-3, SPU, Central Inst. of Agric Engg., Nabi Bagh, Berasia Road, Bhopal 462018, M.P., India.

The research efforts for reducing the food legume milling losses (10-15%) by machine design and optimization of parameters are reviewed. Among the machines reported the abrasive rice polisher, URD sheller and culinder-concave dehuller can find commercial acceptability with specified machine parameters. The effect of grain parameters and environmental factors on pulse processing has not received due attention. Some small capacity units developed for dehusking and splitting of food legumes have been identified for use in developing countries. The details of various aspects are reported along with the suggestions for further work.

105

Effect of Lug Parameters on Tractive Performance of Tractor Tyres in Situ: N.K. Bansal, Asst. Prof., Dept. of Agric. Engg., H.A.U., Hisar; R.B. Ram, Prof., Dept. of Farm Machinery, Rajendra Agric. Univ., Bihar, India.

Lugs on traction wheel contribute a lot to its field performance. No consistent results have been obtained until the present study. An attempt was made to observe the effect of lug spacing and lug height on tractive performance of tyre in three different soil conditions, namely; sandy, vegetative covered (grass) loam, and clay loam soils. It was found that wide spaced lug tyre performed better in sandy soil and vegetative covered soil. The performance of high lug tyre was better in clay loam soil whereas its effect on the performance was negligible in sandy soil.

108

Heated Air Drying of Paddy under Controlled Conditions: M.N. Islam, Assoc. Prof., Dept. of Farm Power and Machinery, Bangladesh Agric. Univ., Mymensingh, Bangladesh; V.K. Jindal, Prof., Division of Food and Agric. Engineering, Asian Institute of Technology, Bangkok, Thailand

The thin-layer drying characteristics of paddy were investigated in a laboratory dryer with provision to control air temperature and relative humidity in the range of 27-44°C and 25-95%, respectively. Air temperature, relative humidity and initial moisture content of the paddy were found to influence the experimental drying rates remarkably.

The semi-logarithmic plots of moisture ratio versus drying time showed non-linear relationships in all drying tests. A second-degree polynomial best represented the drying of paddy in the range of experimental conditions. The parameters of the developed thin-layer drying equation were related to the air temperature, relative humidity and initial moisture content of paddy by step-wise multiple regression analysis. The correlation coefficients for such relationships were low, indicating that the developed thin-layer drying equation may not be used as a crude approximation of experimental drying results. Therefore, a more complex model for deep bed drying of paddy is required.

109

Effect of Moisture on Rough Rice Storage in Open and Closed Containers: Md. Wali Ullah, Assoc. Prof.; Md. Shahjahan Mia, Md. Zahidur Rahman and Md. Mostafa Kamal, Graduating Students, Dept. of Farm Structures, Bangladesh Agric. Univ., Mymensingh, Bangladesh.

Paddy or Rough rice of Pajam variety was stored at different moisture contents, (13.8%, 18.5%, 22.22%, 25.91% and 29.6%) in order to investigate the effect of moisture content on changes during storage in small containers. Two kinds of storage conditions were maintained, namely; open condition and closed condition. The parameters studied were viability, changes in moisture content and dry matter loss at intervals of 3 weeks for a storage period of 15 weeks.

For all samples, grains with high moisture content were affected by fungi in closed containers only which may be due to the air space (about 10%) left inside the top of the containers. The viability of paddy in open containers was higher than in closed containers for all levels of moisture content. Temperature was an influential factor contributing to higher germination. Grain moisture content did not vary widely for both the types of containers. Maximum dry matter loss was 0.33% at moisture content of 29.83% in closed containers.

114

Paddy Grain Hardness and its Interaction with Proximate Composition and Milling Characteristics: P.K. Omre, Research Associate; Maharaj Narain, Prof., Centre of Advanced Studies in Post Harvest Technology, G.B. Pant Univ. of Agriculture and Tecnology, Pantnagar 263145, India.

Grain hardness and milling characteristics of six Indian paddy varieties (Jaya, Cauvery, Ratna, Saket-4, Type-3 and N-12) were determined at 10, 13 and 16% (d.b.) moisture and three cross head speeds (15, 25 and 35 mm/min).

The grain hardness varied significantly from variety to variety and kernel to kernel. It ranged from 116.5 N to 207.6 N for paddy and 51.4 N to 91.6 N for brown rice. The hardness of brown rice was 40-50% of the paddy grain. In most paddy varieties, hardness decreased with an increase in moisture content. Multiple regression equations for head yield and grain hardness as a function of grain thickness, crude protein and crude fibre were also developed.

117

Study on Comprehensive Effects of Different Soil Preparation Processes in Fine Paddy Field: Zhang Libing, Assoc. Prof,; Wu Shisi, Prof., Agric. Engg. Dept., Zhejiang Agric. Univ., Hangzhou, Zhejiang, China.

Rotary tillage has been one of the main preparation processes in paddy field in South China since rotary cultivator was developed as a kind of machinery to take the place of tractor-ploughing. But, to the problem whether adopting rotary tillage for a long period will make soil of tilled layer hardened and impervious, there is not a reasonable conclusion.

The study was conducted from 1983 to 1987 at farmers' fields in Jiaxin which lies in the main rice growing tract of China. Comparative performance of plough and rotary cultivator used independently to prepare fine paddy soil was evaluated.

During the test for five years, under the condition of field dry naturally, two kinds of preparation processes, rotary tillage and ploughing, made no significant difference in the effects on physical-chemical properties of soil and yield. Under the condition of field irrigated with 3-4 cm water layer before being prepared, ploughing made better effects on the physical-chemical properties of soil and obtained higher yield than rotary tillage does. The range of difference in yield was 3.4% equally. The conclusion was obtained that it is more reasonable to say that under different conditions, rotary tillage will result in different effects on soil physical and chemical properties. The key is to choose different preparation process under different condition for preparation process. ■■

NEWS

International Congress on Agricultural Engineering
October 12-14, 1992
Beijing, China

Sponsored by: Chinese Society of Agricultural Engineering, American Society of Agricultural Engineering, Japanese Society of Agricultural Machinery, Asian Institute of Technology (Thailand), Silsoe College (U.K.), Moscow Institute of Agricultural Engineers (USSR), Sino - Foreign Socio - Cultural Exchange Association, China — EC Centre for Agricultural Technology, Beijing Agricultural Engineering University.

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Submission of Manuscripts
April 30, 1992

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Agraria International
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Vienna, Austria

Agricultural Show Window for Europe's Eastern Countries: Austria's agriculture has recently invested some 1.5 billion USD in agricultural machinery and plants and therefore represents a remarkable market with a corresponding growth potential. The oncoming involvement with the EC, for which the course is expected to be set in 1992, will undoubtedly also produce new impulses for this area.

In addition the AGRARIA INTERNATIONAL is, however, intended as a competent agricultural show window for the Danube region and the Eastern European neighbouring countries. Here there exists, as a matter of fact, an immense backlog and consequently a considerable need for information on topical technical knowledge and know-how, on efficient cultivation, harvesting and processing methods and on high-performance working materials, machinery and tools.

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Development of the Portable Votex Ricefan Thresher

(Netherlands)

By A.A. Wanders

In Africa and Latin-America, mostly medium to large-size conventional types of threshers have been introduced in the past: threshers equipped with conventional shaker and sieve mechanisms, weighing 500 up to 2000 kg and driven by engines of 5 to 20 kW.

Operation and management of such expensive and heavy equipment, however, is not within the reach of the predominantly small farmers, nor is the exploitation on a private customhiring basis in most countries economically or politically justified. Consequently, these types of threshers are essentially being operated and maintained/repaired by parastatal organizations, charging the small farmers in a specific project area threshing fees of 10 to 12% of the threshed product.

Because of technical and organizational constraints, this approach proved to be less successful in a number of projects, both for the small farmers (delay of threshing, lack of participation, high costs) as well as for the parastatal organizations (uneconomical exploitation, high overhead costs). More recently smaller and cheaper threshers from Asia are being introduced or tested in Africa as well: either the "hold-on feeding threshers" or the above mentioned axial-flow threshers. So far these threshers proved to be less successful, this was due to poor adaptability to the local crop conditions and to shortcomings in terms of operational reliability and durability.

Based on above considerations and experience, there is an obvious need for an improved rice threshing technology, with particular reference to smallholder wetland rice regions in Africa and Latin-America. This is needed, either for supplementing the available manual labour to reduce the post-harvest losses and to increase the speed and timeliness of the threshing operation; or as a viable alternative for the conventional medium to large-size threshers to increase the timeliness of threshing, to reduce the losses and threshing costs and especially to increase the farmer's participation and responsibility in the whole post-harvest process.

The present report describes the process of development and the technical characteristics of the VOTEX Ricefan thresher, highlighting the design criteria which form the basis of this new threshing technology.

Size: 29.5 × 21 cm, pp.16, soft cover.

For distribution contact:
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1990 Directory of Country Environmental Studies

(U.S.A.)

Editors: D.S. Baker, D.B. Tunstall

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BACK ISSUES

(Vol. 21 No. 1, Winter 1990 ~)

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INSTRUCTIONS TO AMA CONTRIBUTORS

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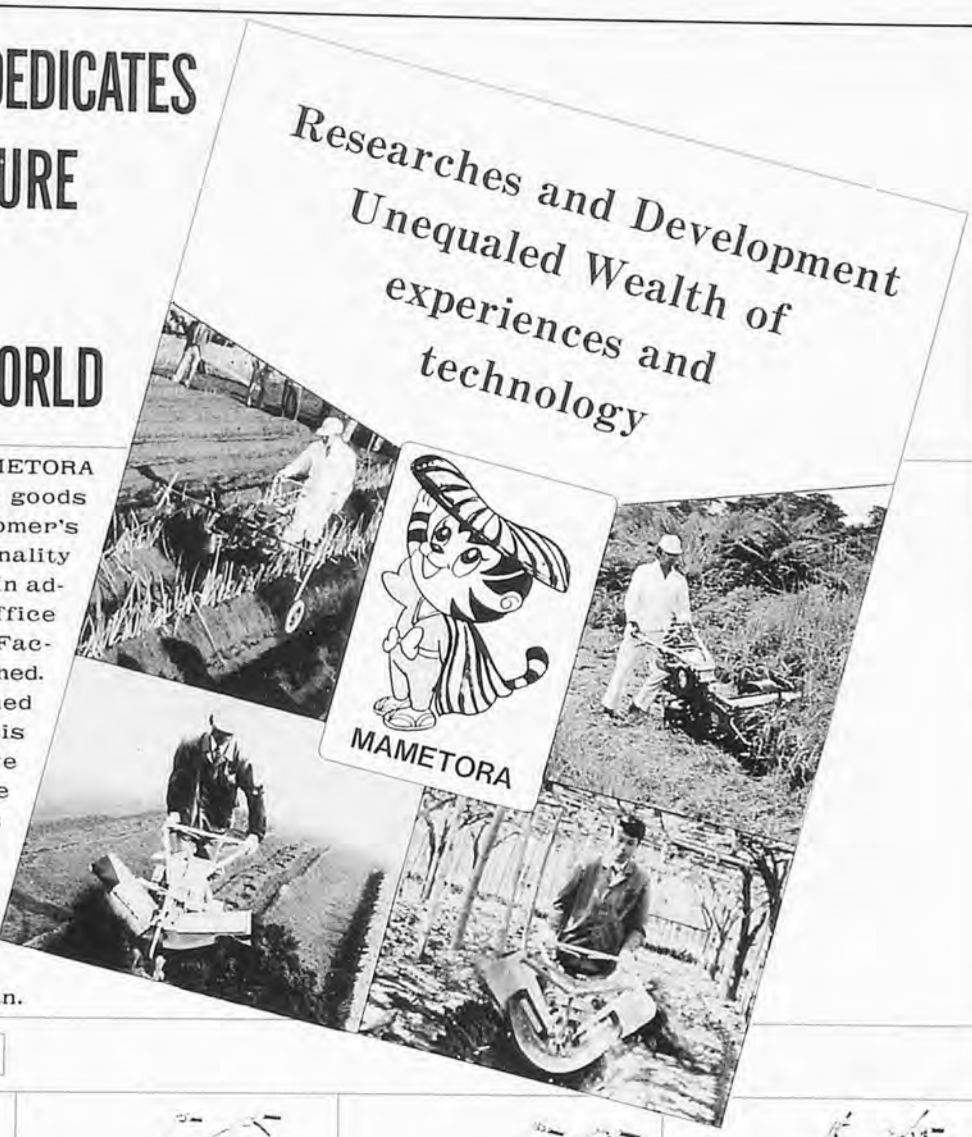

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